Wang, S., Schenk, M., Guo, H., & Viquerat, A. (2020). Tip Force and Pressure Distribution Analysis of a Deployable Boom During Blossoming. *International Journal of Solids and Structures*, 193-194, 141-151. <https://doi.org/10.1016/j.ijsolstr.2020.01.026>

Peer reviewed version

License (if available):
CC BY-NC-ND

Link to published version (if available):
[10.1016/j.ijsolstr.2020.01.026](https://doi.org/10.1016/j.ijsolstr.2020.01.026)

[Link to publication record in Explore Bristol Research](#)
PDF-document

This is the author accepted manuscript (AAM). The final published version (version of record) is available online via Elsevier at <https://www.sciencedirect.com/science/article/pii/S0020768320300329> . Please refer to any applicable terms of use of the publisher.

University of Bristol - Explore Bristol Research

General rights

This document is made available in accordance with publisher policies. Please cite only the published version using the reference above. Full terms of use are available:
<http://www.bristol.ac.uk/red/research-policy/pure/user-guides/ebr-terms/>

Tip Force and Pressure Distribution Analysis of a Deployable Boom During Blossoming

Sicong Wang^{a,*}, Mark Schenk^b, Hongwei Guo^a, Andrew Viquerat^c

^a*Harbin Institute of Technology, Harbin, China, 150001*

^b*University of Bristol, Bristol, England, BS8 1QU*

^c*University of Surrey, Guildford, Surrey, England, GU2 7XH*

Abstract

Deployable coilable booms have many advantages for use in space, but these kinds of structures sometimes experience a deployment failure mode called ‘blossoming’. Blossoming of a coiled boom occurs when the boom stops deploying, and instead unwinds and expands within the deployer. This can occur even in the presence of sprung rollers used to constrain the coil. In the blossoming process, friction between the layers of the coil plays an important role that has only been briefly considered in previous work. In order to be able to model and predict the onset of this phenomenon more precisely, the pressure distribution between adjacent layers of the coil must be known. This paper establishes a numerical model to investigate the pressure distribution within a coiled open-section tape spring boom, then combines this result with theoretical analysis to produce an estimate of the maximum tip force that a deploying boom can withstand before the onset of blossoming. The effect of the roller springs’ stiffness and the boom friction coefficient are also taken into account in the simulation. The results of the theoretical analysis and numerical simulation are compared with previous experimental results to provide some practical verification.

Keywords:

Deployable boom; blossoming; tape spring

*Corresponding author

URL: wangsicong890101@126.com (Sicong Wang)

1. Introduction and literature review

Deployable coilable booms like that shown in Figure 1(a) have a variety of potential applications in space structures such as solar panels, membrane antennas and solar sails. This is due to their high packaging efficiency, low mass, simple deployment mechanism, and scalability. An early design was the called Storable Tubular Extendible Member (STEM) which was invented by Klein for the first Canadian satellite Alouette in 1960s to act as a compact and lightweight antenna [1]. After this concept was proposed, a variety of designs was developed to improve the boom stiffness and make it more stable and reliable during the deployment process. A closed-section member with a ‘lenticular’ cross-section was put forward by the Marquardt Corporation [2]. These lenticular booms had high torsional rigidity when deployed due to their closed cross-section, but their double wall thickness led to larger diameter coils than STEM booms when stowed [3]. Triangular Rollable And Collapsible (TRAC) booms were another kind of deployable boom, first investigated by the Air Force Research Laboratory (AFRL). By comparison, TRAC booms had higher bending stiffness than STEM or lenticular booms for the same packaged height [4], however, their torsional stiffness was quite low [5].

Deployable coilable booms cannot only be manufactured from isotropic materials such as copper-beryllium or steel, but also from composite materials such as carbon and glass fiber [6]. Bi-stable variants of these composite open-section coilable booms have also been developed, allowing lighter hold-down mechanisms to be used to restrain the booms in their coiled state [7, 8, 9, 10]. The in-space applications of deployable booms have increased in recent years, including on the Hubble space telescope, NanoSail-2D, LightSail and InflateSail [11, 12, 13, 14].

Despite the advantages that coilable flexible thin-walled booms have, they can exhibit a potentially catastrophic failure mode when deploying. A simple way of deploying a coiled boom is to drive the hub around which the boom(s) are coiled (see Figure 1(b)) with a motor, producing a controlled deployment

Figure 1: Single and multi tape spring coils

sequence in which the outermost layer of the coil continuously transitions to its straight state and exits the deployer housing. Blossoming is a failure mode in which a coiled boom stops deploying and instead unwinds within the deployer while the central hub turns. In this case, if the hub continues turning, the deployer will be jammed and the boom may eventually be irreparably damaged. One of the causes of blossoming is an excessive compressive axial load (or ‘tip force’) being applied to the already deployed portion of the boom. Such a tip force might be generated if a boom experiences some resistance while uncoiling, such as might happen when a boom is used to unwrap and tension the thin membrane of a solar or drag sail. A key question in the design phase is what magnitude of tip force can be sustained before the onset of blossoming. For this purpose, Hoskin developed an energy method model of a coiled tape spring copper-beryllium (CuBe) boom and acquired the maximum tip force, self-blossoming torque and self-deployment torque when the boom was deploying [15]. Hoskin also found the *friction torque*, which is a resistance torque caused by friction between adjacent layers of a coil in the blossoming process. Friction torque had a large effect on the maximum tip force, so Hoskin ana-

lyzed the relative motion between adjacent layers when blossoming occurred, and predicted the friction torque [16, 17]. However, in Hoskin’s friction analysis, the pressure distribution between the layers of the coil was assumed to be either zero, or uniform in nature when forces were applied to the outside of the coil. The cross-sectional shape of coiled tape springs was also assumed to be perfectly flat, which was not the case for most tape spring geometries [18]. These phenomena will affect the pressure distribution between layers of a coil. Because of this, one of the main aims of this paper is to investigate the pressure distribution that develops between the layers of a coil, both with and without the presence of external *compression rollers* to constrain the expansion of the coil. The blossoming principles explored in this paper are applicable to a variety of different kinds of boom cross-section, however, the main study made here is of STEM type open-section tape spring booms.

Section 2 presents the different sources of torque considered in the ensuing analysis, and their balance relationship in the boom blossoming process. Section 3 presents a finite element model of a single boom in Abaqus, including the pressure distribution between adjacent layers of a coil, and the resulting the maximum tip force that can be sustained. A semi-empirical prediction of the total pressure is presented in this section. In Section 4, the effect of compression roller spring stiffness and boom friction coefficient is obtained via the analytical method and a numerical model. Section 5 presents the results of an experiment to verify the simulation model of Section 3. Section 6 will conclude the paper.

2. Blossoming analysis

A typical deployment process of a boom is shown in Figure 2(a). In the figure, the boom root is fully fixed to the hub and the boom tip goes outward when the hub turns. Compression rollers around the outside of the coil prevent gaps appearing between the layers of the coil when deploying. The number of compression rollers used will affect the pressure distribution between the layers of the coil, however, it is common for four rollers to be used in space sail

Figure 2: Boom deployment diagram

deployment mechanisms [13, 14]. Blossoming will occur when the load on the boom tip is too high (shown in Figure 2(b)), or the compression rollers exert too much traction on the outermost layer of the coil. In this case, if the hub continues turning, the deployer will be jammed and the boom will be damaged.

Four different sources of torque will be considered in the following analysis. They are friction torque T_f , boom energy torque T_b , compression spring torque T_s and hub torque T_h . A moment balance relationship will be constructed between these torques. For ease of illustration, the main parameters are shown in Figure 3. These parameters are: hub radius R , transverse radius r_y , cross-section path length b , and coiling radius r_x (simplified to the hub radius R in the following analysis). The coil is formed by bending the tape spring boom in an equal sense direction in following analysis (illustrated in Figure 4).

The model being developed here is based on the assumption that the number of turns, or layers, within the coil is relatively small. By making this assumption, it is possible to treat the layers of the coil as all having the same radius of curvature. With a greater number of turns the coil would start to become thicker, and the difference in radius between the innermost and outermost layers would become more significant. In practical designs, 10-15 turns are typical while InflateSail had 14 turns [19]. In this case a model based on an Archimedes spiral would need to be developed.

T_f is a torque caused by friction between layers of the coil, and works to

Figure 3: Main parameters of the coiled boom and the relative motion of different layers in the blossoming process

Figure 4: Bending styles of a boom

prevent blossoming. The pressure (P) on the coil comes from two sources: the compression rollers (P_r) and the boom itself (P_b). P_r comes from the compression rollers pressing on the outermost layer of the coil, but roller forces tend not to transmit through all the coil layers [20, 21]. Apart from P_r , there is still some pressure between adjacent layers. As the layers are not wholly flat, especially near the edges [18], these boundary regions often form points of contact between adjacent layers of a coil.

As compression rollers are generally able to rotate, friction between the rollers and the outermost layer of the coil will be ignored in this analysis. From previous work, the radii of different layers of a coil will increase simultaneously during the blossoming process. That is to say there is no relative motion between the innermost layer and the hub until this process begins, and a gap appears around the hub. Therefore, the friction between the boom and the hub will also be neglected. In the blossoming process, no gap appears between adjacent layers. Defining a torque making the boom deploy normally as positive, T_f (positive) can be expressed as:

$$T_f = C_f \frac{1}{n_l} RN \quad (1)$$

where C_f is the coefficient of friction between the layers of the coil, n_l is the number of layers/turns in the coil ($n_l \geq 2$), and N is the total pressure on the coil:

$$N = A \sum_{i=1}^{n_l-1} (P_{ri} + P_{bi}) \quad (2)$$

in which A is the total area of contact between adjacent layers of the coil. Since different layers move at different speeds (shown in Figure 3), the relative motion between adjacent layers is $1/n_l$ of the hub speed [16]. P_{ri} and P_{bi} are pressures caused by the compression rollers and the boom itself between coil layers i and $(i+1)$ in Figure 3 respectively.

T_b is a torque from the boom strain energy (U_{coil}). The strain energy in the coil can be released either by the normal deployment process (self-deployment

torque T_{sd} , positive), or by blossoming (self-blossoming torque T_{sb} , negative). T_b can be expressed as:

$$T_b = T_{sd} + T_{sb} \quad (3)$$

T_{sd} and T_{sb} in Equation 3 can be calculated by an energy method. According to Timoshenko's and Calladine's theories [22, 23] and Hoskin's work [16], strain energy in a bent tape spring section (equal sense) can be expressed as:

$$U = \frac{1}{2}DA (\Delta\kappa_x^2 + \Delta\kappa_y^2 - 2\nu\Delta\kappa_x\Delta\kappa_y) \quad (4)$$

in which

$$A = bL = bR\theta \quad (5)$$

$$D = \frac{Et^3}{12(1-\nu^2)} \quad (6)$$

where U is the coil strain energy, ν is boom material's Poisson's ratio, L is the total length of boom in the coil, θ is the total angle swept out by the turns in the coil, E is the material elasticity modulus, t is boom thickness, and D is the flexural rigidity of the thin walled structure. The two curvatures $\Delta\kappa_x$ and $\Delta\kappa_y$ are:

$$(\Delta\kappa_x, \Delta\kappa_y) = \left(\frac{1}{R}, \frac{1}{r_y} \right) \quad (7)$$

In normal deploying process, the coiled length ($L = R\theta$) decreases while the radius of each layer (simplified to R) is assumed to have a constant value. Therefore,

$$\begin{aligned} T_{sd} &= \frac{dU_{coil}}{d\theta} \\ &= \frac{d}{d\theta} \left(\frac{DbL}{2} \left(\frac{1}{R^2} + \frac{1}{r_y^2} - \frac{2\nu}{Rr_y} \right) \right) \\ &= \frac{d}{d\theta} \left(\frac{DbR\theta}{2} \left(\frac{1}{R^2} + \frac{1}{r_y^2} - \frac{2\nu}{Rr_y} \right) \right) \\ &= \frac{Db}{2} \left(\frac{1}{R} + \frac{R}{r_y^2} - \frac{2\nu}{r_y} \right) \end{aligned} \quad (8)$$

In the blossoming process, the radius of each layer (simplified to $R = \frac{L}{\theta}$) increases, while the coiled length (L) stays constant. Therefore, T_{sb} can be

calculated as follows:

$$\begin{aligned}
T_{sb} &= -\frac{dU_{coil}}{d\theta} \\
&= -\frac{d}{d\theta} \left(\frac{DbL}{2} \left(\frac{1}{r_o^2} + \frac{1}{r_y^2} - \frac{2\nu}{Rr_y} \right) \right) \\
&= -\frac{d}{d\theta} \left(\frac{DbL}{2} \left(\frac{\theta^2}{L^2} + \frac{1}{r_y^2} - \frac{2\nu\theta}{Lr_y} \right) \right) \\
&= -Db \left(\frac{\theta}{L} - \frac{\nu}{r_y} \right) \\
&= -Db \left(\frac{1}{R} - \frac{\nu}{r_y} \right)
\end{aligned} \tag{9}$$

The torques in Equations 8 and 9 are both derived from the boom strain energy which can be released in two different ways: one is to extend the tip outwards with a corresponding change in hub angle (normal deployment), and the other one is to increase the coil radius with a fixed tip (blossoming). Each of these energy release methods has its corresponding torque equation. Combining Equation 8 and 9 into Equation 3, T_b can be known as:

$$T_b = \frac{DbR}{2} \left(\frac{1}{r_y^2} - \frac{1}{R^2} \right) \tag{10}$$

In the blossoming process, the roller springs are gradually compressed by the increasing coil radius (shown in Figure 5(a)). In other words, roller forces can generate a reactive torque (compression spring torque T_s , positive) to the hub to restrain the coil from blossoming (shown in Figure 5(b)). In the blossoming process, some extra energy (U_{extra}) is introduced into the compression springs. Therefore, T_s can be obtained from U_{extra} as:

$$\begin{aligned}
T_s &= \frac{dU_{extra}}{d\gamma} \\
&= n_r \frac{d}{d\gamma} \left(\frac{1}{2} k (r_{pre} + \Delta r)^2 - \frac{1}{2} k r_{pre}^2 \right) \\
&= n_r \frac{d}{d\gamma} \left(F_{pre} \Delta r + \frac{1}{2} k \Delta r^2 \right) \\
&= n_r (F_{pre} + k \Delta r) \frac{d\Delta r}{d\gamma}
\end{aligned} \tag{11}$$

Figure 5: Compression spring torque diagram

where γ is the hub rotation in the blossoming process (see Figure 6), n_r is the number of rollers, r_{pre} is the spring pre-compression length, F_{pre} is the roller pre-load, Δr is the roller displacement, and k is the spring stiffness.

The term Δr in Equation 11 can be obtained as follows. In the blossoming process, the coiled length (L) is constant. Therefore,

$$L = R\theta = r_b(\theta - \gamma) \quad (12)$$

where r_b is the outermost radius of the coil when blossomed (shown in Figure 6). And then,

$$\Delta r = r_b - R = \frac{R\gamma}{\theta - \gamma} \quad (13)$$

The derivative of Δr respect to γ is:

$$\frac{d\Delta r}{d\gamma} = \frac{R\theta}{(\theta - \gamma)^2} \quad (14)$$

When blossoming just begins to happen, γ in Equation 14 approaches zero. Therefore Equation 14 can be simplified as:

$$\frac{d\Delta r}{d\gamma} = \frac{R}{\theta} \quad (15)$$

Figure 6: Key parameters of the blossoming process

Combining Equations 13 and 15 into Equation 11, T_s can be obtained as:

$$T_s = n_r \left(F_{\text{pre}} + k \frac{R\gamma}{\theta - \gamma} \right) \frac{R}{\theta} \quad (16)$$

Since $\gamma \rightarrow 0$, T_s can be seen as:

$$T_s = n_r F_{\text{pre}} \frac{R}{\theta} \quad (17)$$

T_h is a torque on the hub provided by a deployment motor. Regarding the deployment system as a whole, a balance relationship between T_h (which is a torque to promote blossoming, negative) and the boom tip force F_t can be obtained as follows:

$$T_h = -F_t \cdot R \quad (18)$$

When these four sources of torque above are in balance, there is a risk that blossoming will occur. The balance equation for the torques is:

$$T_f + T_b + T_s + T_h = 0 \quad (19)$$

Combining Equation 19 into Equation 18, a maximum allowable boom tip force expression can be obtained as follows:

$$\begin{aligned} F_t &= -\frac{T_h}{R} \\ &= \frac{T_f + T_b + T_s}{R} \end{aligned} \quad (20)$$

Figure 7: Roller pressure distribution diagram (illustrative distribution only)

In Equation 20, only T_f cannot be obtained directly using a theoretical method because the precise pressure distribution on the boom is quite difficult to estimate. In previous friction analysis by Hoskin [16], roller force was assumed to produce a uniform pressure distribution on different layers, and inter-layer pressure when no roller force was introduced was considered to be zero. That is to say this problem was simplified as $P_{ri} = F_r n_r / A$ and $P_{bi} = 0$ respectively, where $F_r = F_{pre} + k\Delta r$ is the force on each roller. However, in reality the total pressure between the layers integrates to a much lower force than that obtained by summing the forces applied by the compression rollers, which is elaborated in Section 3.2. This is especially true at the inner layers of the coil (an illustrative diagram is shown in Figure 7). Note that the the pressure between the layers of the coil is also not zero in the case that all roller forces are removed.

3. Numerical analysis

3.1. Model development

A finite element model was established in Abaqus (shown in Figure 8.) A plane symmetry condition is used and only half of the model is shown. Since the boom thickness was much smaller than its radius, S4R shell elements and quadrilateral meshes were appropriate. A mesh refinement study indicated that using ten elements across each half section was sufficient to provide an appropriate balance between accuracy and computational expense, since a model with

Figure 8: Color-map of blossoming simulation

twenty elements across each half section was established and very similar results produced. As two sides of the boom needed to contact each other, only the general contact type can be used.

The boom geometric parameters used in the following analysis are chosen to mimic those used in the InflateSail [14] drag deorbiting sail. The parameters are shown in Figure 3. The numerical values are given in Table 1. To capture both the onset and post-blossoming behavior, the hub was rotated for four full turns in simulation to produce a coil with four layers. Unlike the carbon fiber booms used in the drag sails mentioned above, the boom simulated in this analysis is isotropic (CuBe). The rollers and the hub in simulation were modelled by S4R shell elements as rigid bodies. The rollers were modeled as cylinders with a mesh size half that of the boom. The roller radius was found to have little impact on the behavior of the blossoming process, hence the size selected was chosen to be appropriate for the size of the deployer mechanism. The hub had the same mesh size as the boom and the hub radius should be as small as possible to minimize the coiled volume needed, while the coil inner layer radius must be larger than the boom natural radius or a high blossoming torque can appear [16].

Table 1: Table of boom properties

b (mm)	t (mm)	r_y (mm)	R (mm)	E (GPa)
31.8	0.13	12.5	15	205
ν	n_l	F_{pre} (N)	k (N/m)	C_f
0.3	4	6	10	0.25

The first goal in the finite element analysis was to determine the pressure distribution between the layers of the coiled boom. The finite element results are used later to produce a semi-empirical equation for the total pressure N between the layers of the coil. There were three steps in this analysis in which the boom was coiled and constrained using compression roller forces. To make the boom coil smoothly, implicit analysis was used. Friction between the coil layers was neglected in this first coiling stage of the analysis.

Step 1 of the coiling process was to flat one end of the boom and attach it to the hub (and wholly fixed).

Step 2 involved a small amount of tension being applied to the tip of the free end of the boom. The boom was then coiled by turning the hub.

Step 3 involved the tensioning tip force being removed, and forces from the compression springs being introduced on the rollers around the outside of the coil. Pressure distributions resulting from the application of different roller forces could then be obtained.

Once the coiled boom model had been generated and the pressure distribution obtained, the same coiled boom model could then be used to simulate the blossoming process. This simulation can be used as a basis for comparison between the theoretical results and the results of an earlier experiment presented in Section 5. The blossoming simulation was achieved by introducing a coefficient of friction between the layers of the coil, and applying a driving torque to the hub. As complex deformation occurs in this part of the simulation, explicit analysis was used to prevent convergence problems. The results from the pres-

sure analysis in the initial coiling analysis were imported as initial conditions. In this analysis, the hub was rotated with a uniform velocity with the boom free end fully fixed. The purpose of the analysis was to find the tip force when the hub just left the 0° position ($\gamma \rightarrow 0$ in Equation 11), while, in the simulation process, the tip force was recorded and averaged while the hub turned from 0° (not included) to 10° , which was regarded as “the onset of blossoming”. A hub rotational velocity of $50^\circ/\text{s}$ was used with a duration of 0.2 s. As the boom was fully balanced after the tension was removed in Step 3 and the friction was not introduced until the blossoming analysis, the small tension imposed in Step 2 did not impact the initial condition of the pressure distribution and the blossoming analysis.

3.2. Pressure distribution

A color-map of the pressure between the layers of the coil derived during the simulation in Abaqus is shown in Figure 9. The pressure distribution with and without roller force is shown in Figure 10. From the two figures it can be seen that the effect of the compression rollers on the pressure distribution between the layers of the coil remains quite local to the point of contact. In Figure 10, Figure 10(a) is the pressure caused by the boom itself P_b (see Equation 1), while Figure 10(b) is the total pressure ($P_r + P_b$). Note that the pressure is mainly concentrated at the boom edge because of the boundary effect that appears when the tape spring boom is flattened. A significant peak appears in each plot at around 200 mm length on account of the compression from the boom transition zone.

The result of integrating the pressure distribution both across the boom width, and along its length is shown in Figure 11. Comparing Figure 11(a) with Figure 11(b), it can be seen that it is between the outer layers of the coil that the effects of applying compression rollers is most marked. Figure 11(c) and 11(d) also show that the roller force is transmitted primarily between the outer edges of the boom.

The result of integrating the pressure over the total area between different

Figure 9: Color-map of normal pressure distribution from simulation (6 N on each roller)

layers of the coil (and across the whole boom) with different roller forces is shown in Figure 12(a). The tip force based on the pressure results in Figure 12(a) and the pressure approximation in [16] (and the simulation tip force from Section 3.3) are shown in Figure 12(b).

It can be seen from Figure 12(a) that the pressure integrals of each layer and the total coil are almost linear. Hence, the total integral can be expressed by a linear function of F_r

$$N(F_r) = pF_r + q \quad (21)$$

where p and q are unknown coefficients.

To find the relationship between N and the boom parameters, the coil total pressures under different parameters acquired by the numerical model are shown in Figure 13, in which F_r changes from 0 N to 6 N (typical values used in experiment and practical use). Four parameters which can wholly describe the boom property were investigated: the number of layers/turns n_l , the transverse radius r_y , the cross-section path length b and the flexural rigidity D . From Figure 13(a), N increases with n_l since more area is under the action of the rollers when more turns are coiled. Meanwhile, as the pressure acting on the inner layer is lower than the outer one, the increment of N is lower than that of n_l . In Figure 13(b), N reduces with increasing r_y as a lower cross-section curvature generates a smaller boundary effect. The gap between different r_y is

(a) Pressure distribution without roller force

(b) Pressure distribution with roller force (6 N on each roller)

Figure 10: Contact pressure distribution between adjacent layers of a coil (one side of symmetry)

(a) Pressure integral across the boom width without roller force

(b) Pressure integral across the boom width with roller force (6 N on each roller)

(c) Pressure integral along the boom length without roller force

(d) Pressure integral along the boom length with roller force (6 N on each roller)

Figure 11: Contact pressure integrals along sections (one side of symmetry)

(a) Pressure integral between coil layers with different roller force

(b) Tip force results comparison

Figure 12: Pressure integral and tip force results

Figure 13: Total pressure parametric analysis

nearly constant, so it can be assumed that r_y has no effect on the slope p of Equation 21. Figure 13(c) and 13(d) show that a more flexible coil acquires a lower pressure. Based on these results and the analysis above, a semi-empirical formula for N derived using a least-squares fit can be found as:

$$N(n_l, F_r, b, D, r_y) \approx 2(n_l - 1)^{0.7} \left(F_r + \frac{1.2bD}{r_y^2} \right) \quad (22)$$

Together with Equations 22 and 1, it is possible to calculate the friction torque T_f , while the boom energy torque T_b and the compression spring torque T_s can be found using Equations 10 and 17. Combining the results above into Equation 20, the boom tip force F_t can be acquired. The tip force of the boom

(a) Tip force with four layers/turns. (b) Tip force with four layers/turns.
 $r_y = 12.5$ mm, $b = 31.8$ mm, $D = 0.0412$ Nm $r_y = 15$ mm, $b = 15.9$ mm, $D = 0.0206$ Nm

Figure 14: Tip force results comparison

with the parameters in Table 1 is shown in Figure 14(a). To further verify the method in this paper, a new boom with the parameters $n_l = 4$, $r_y = 15$ mm, $b = 15.9$ mm, $D = 0.0206$ Nm is analyzed, and the tip force is shown in Figure 14(b). Comparisons with the results acquired by the method in [16] are also shown in Figure 14.

3.3. Blossoming simulation

To verify the results acquired in Section 3.2, an estimate of the tip force is obtained by direct simulation using the finite element model described in Section 3.1. The results are shown in Figure 14. Note that these results will be further verified by experiment in Section 5. To compare with the work in [16], the friction coefficient C_f and spring stiffness k are taken to be 0.25 and 10 N/m respectively in the simulation. From Figure 14, it can be seen that new analysis results in this paper are much closer to the simulation results than those obtained by the method in [16] alone, especially when higher roller force and more layers are used.

4. Parametric study

Using the analytical method from Section 2 and the semi-empirical model from Section 3, a parametric study to explore the maximum tip force a boom can

sustain before the onset of blossoming was carried out across a range of different friction coefficient C_f and spring stiffness k values (shown in Figure 15). The tip force obtained by direct simulation is also provided for a comparison in Figure 15.

From Figure 15(a), it can clearly be observed that spring stiffness has little effect on friction torque. While this result is initially surprising, it stems from the fact that the radial displacement of the outer layer of the coil is quite small during the initial stages of blossoming (because of a small γ in Equation 13 when blossoming just occurs). This small radial displacement means that the springs attached to the compression rollers are unable to greatly affect the overall energy balance in the system as they are compressed, i.e. a small Δr in Equation 11. Instead, the term that has the greatest effect on the maximum tip force is the spring pre-load (F_{pre}) (rather than the spring stiffness, k). The spring pre-load influences the tip force via the compression spring torque (T_s) in Equation 17, and the friction torque (T_f) by increasing P_{ri} in Equation 1. However, high spring stiffness can be quite influential in keeping the boom deploying normally because it allows the rollers to maintain a high pre-load on the outer layer of the coil, especially when the radius is quite small near the end of the deployment. Because k has such a small impact on T_f , the x axis in Figure 15(a) is presented on a logarithmic scale. The parameters used in generating the figure are friction coefficient $C_f = 0.25$, and pre-load $F_{\text{pre}} = 6$ N.

Figure 15(b) confirms that the maximum sustainable tip force rises with increasing friction coefficient. The boom can deploy normally only if load on the tip is below the boundary line. The parameters used in generating the figure are pre-load $F_{\text{pre}} = 6$ N and spring stiffness $k = 10$ N/m.

The increase in maximum tip force with coefficient of friction shown in Figure 15(b) is approximately linear. This characteristic is in accordance with the behavior of C_f in Equation 1. However, the gradient of the relationship between maximum tip force and coefficient of friction differs depending on the means of calculation. It appears that the ability of Equation 20 to produce accurate results for high coefficients of friction is uncertain, and this will require further

Figure 15: The maximum tip force in different friction coefficient and spring stiffness

investigation in future work.

5. Experiment

To verify the accuracy of the numerical model presented in Section 3, previously performed experiments [16] were re-produced in the simulation. At the beginning of the experiment, the friction coefficient of the boom material was measured as $C_f = 0.25$. The blossoming experiment method used in [16] is shown in Figure 16. The boom coil was constrained by roller forces from compression arms actuated by torsion springs, while the hub was driven by an electric motor at a uniform velocity. The boom tip was fully fixed by a force sensor which was able to measure forces when the hub was turning. The initial number of turns was five. The hub was turned in 36° increments, for a total of 360° (leaving four turns at the end), while the tip forces at each increment were measured. To match the conditions of the experiment, the number of turns in the simulated coil was increased from four (as in Table 1), to five.

Using the numerical model in Section 3, the same situations in the experiment were reproduced in the model. As the simulation results fluctuated greatly when roller force was high (higher friction tends to make the hub jump during the blossoming process), only results for roller forces in the range 1 N to 4 N are given. Both experimental and simulation results are shown in Figure 17 and 18. In Figure 17, tip forces can be seen to gradually increase as the hub turns, and the experiment results do tend to be lower than the simulation ones because the presence of friction resistance in the roller bearings, and the vibration caused by the turning of the electric motor (overcoming some of the static friction between the coil layers) in the blossoming experiment can lead to an earlier onset of blossoming.

Figure 16: Blossoming experiment diagram

Figure 17: Experiment and simulation results comparison

Figure 18: Experiment and simulation figures comparison

6. Conclusion and discussion

Blossoming is a failure mode that can occur in the deployment process of tightly coiled deployable booms. To be able to design a deployer system that is capable of functioning reliably, it is necessary to be able to predict when this failure mode is likely to occur, and to ensure that the geometric and operational parameters of the deployer are such that blossoming is avoided. This paper analyzed four sources of torque and their balance relationship in the boom blossoming process. From this, an estimate of the maximum tip force that the boom can carry during deployment can be estimated. One of these torques, friction torque, was difficult to acquire by analysis because of the complexity of the geometry involved, so this paper established a numerical model to solve the problem.

The numerical analysis presented allowed a pressure distribution between the layers of the coiled boom to be found. It was found that the force exerted

by compression rollers on the outermost layer of the coil is only transmitted through a small number of layers, and is concentrated in a small region in each roller's vicinity. The boom edges carried a greater portion of the roller forces than other parts of the boom section from layer to layer because of the presence of a boundary effect. It was also found that some pressure between the layers of the coil is present even in the absence of compression rollers. Combining a theoretical analysis and the results of the simulation pressure investigation, maximum tip forces under different roller forces were found. The tip forces were also exported from the simulation model to check the analytical results.

Afterwards, a parametric study was conducted to understand the effects of spring stiffness and friction coefficient on blossoming. It was found that spring stiffness had little effect on the allowable tip force because the spring compression displacement was quite small in the initial stages of blossoming. The friction coefficient had an apparent linear effect on the tip force. This characteristic was in accordance with theoretical analysis, except that the gradient of the relationship was smaller in the results of the simulation than the analytical model. Further investigation into this should make use of a larger number of turns in the coil, likely necessitating the treatment of the coil as an Archimedes spiral, rather than as a set of concentric circular rings.

Finally, existing experimental results were used to verify the fidelity of the numerical model. It can be seen that the simulation results are close to those generated in the experiments.

The results in this paper will help to more precisely determine the maximum tip force a deployable boom can afford before blossoming occurs. The paper has also shown the importance of including the role of friction between layers of a coil in accurately predicting the conditions under which blossoming appears.

An example is given to show the applicability of the work presented in this paper to a typical deployment mechanism. A single boom deployer, with dimensions $b = 20$ mm, $t = 0.1$ mm, $r_y = 10$ mm, $R = 15$ mm, $E = 150$ GPa, $\nu = 0.35$, $N_l = 4$, $C_f = 0.25$, whose geometry is shown in Figure 19(a), is given, and the load on the boom tip is set to be no more than 2 N. Single

Figure 19: Practical design application ($b = 20$ mm, $t = 0.1$ mm, $r_y = 10$ mm, $R = 15$ mm, $E = 150$ GPa, $\nu = 0.35$, $N_l = 4$, $C_f = 0.25$)

boom deployers for space applications are now quite common, with a number of companies offering commercial products. The dimensions in this example are chosen to be representative of such systems. Practical considerations such as motor vibration and friction in roller bearings, will impact the boom working condition, so a safety margin (assumed to be 50%) should be considered. A plot showing the boom tip force for different roller compression values can be seen in Figure 19(b). From the figure, the compression force on each roller F_r should be no less than 3 N.

7. Acknowledgements

The authors would like to acknowledge that the research presented in this paper was carried out with the aid of National Natural Science Foundation of China (Grant No. 51835002).

References

- [1] R. I. Bourgeois-Doyle, George J. Klein: The Great Inventor: NRC Press Biography Series, No. 2, NRC Research Press, 2004.
- [2] C. Staugaitis, State-of-the-art materials and design for spacecraft booms,

- in: JPL Technical Memorandum, Jet Propulsion Laboratory, California Institute of Technology, 1970, pp. 43–45.
- [3] F. Roybal, J. Banik, T. Murphey, Development of an elastically deployable boom for tensioned planar structures, in: 48th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference, 2007, p. 1838.
 - [4] J. Banik, T. Murphey, Performance validation of the Triangular Rollable And Collapsible mast, in: 24th Annual AIAA/USU Conference on Small Satellites, 2010.
 - [5] C. Leclerc, S. Pellegrino, Ultra-thin composite deployable booms, in: Proceedings of IASS Annual Symposia, no. 13, International Association for Shell and Spatial Structures (IASS), 2017.
 - [6] T. Murphey, J. Welsh, Composite materials in deployable space structures, in: 16th International Conference on Composite Materials, 2007.
 - [7] S. D. Guest, S. Pellegrino, Analytical models for bistable cylindrical shells, Proceedings of the Royal Society: Mathematical, Physical & Engineering Sciences 462 (2006) 839–854.
 - [8] E. Kebabdz, S. Guest, S. Pellegrino, Bistable prestressed shell structures, International Journal of Solids and Structures 41 (11-12) (2004) 2801–2820.
 - [9] K. Iqbal, S. Pellegrino, Bi-stable composite shells, in: 41st Structures, Structural Dynamics, and Materials Conference and Exhibit, American Institute of Aeronautics and Astronautics, 2000. doi:10.2514/6.2000-1385.
 - [10] A. Daton-Lovett, Extendible member, US Patent US6217975B1 (2001).
 - [11] M. Lake, W. Francis, K. Craven, T. Murphey, Robust, highly scalable solar array system, in: 3rd AIAA Spacecraft Structures Conference, 2016, p. 1951.

- [12] C. Katan, L. D., NASA’s next solar sail: lessons learned from NanoSail-D2, in: 26th Annual AIAA/USU Conference on Small Satellites: Enhancing Global Awareness through Small Satellites, 2012.
- [13] R. Ridenoure, R. Munakata, A. Diaz, S. Wong, P. B., D. Stetson, D. Spencer, J. Foley, LightSail program status: one down, one to go, in: 29th Annual AIAA/USU Conference on Small Stallites, 2015.
- [14] A. Viquerat, M. Schenk, V. Lappas, B. Sanders, Functional and qualification testing of the InflateSail technology demonstrator, in: 2nd AIAA Spacecraft Structures Conference, 5–9 January 2015, Kissimmee, FL, 2015.
- [15] A. Hoskin, A. Viquerat, An analysis of a coiled tape spring during compression and extension, in: 3rd AIAA Spacecraft Structures Conference, San Diego, California, 2016.
- [16] A. Hoskin, Blossoming of coiled deployable booms, Ph.D. thesis, University of Surrey (2018).
- [17] A. Hoskin, A. Viquerat, G. Aglietti, Tip force blossoming of coiled deployable booms, *Int. J. of Solids and Structures* 118-119 (2017) 58–69.
- [18] D. A. Galletly, S. D. Guest, Bistable composite slit tubes. II. A shell model, *International Journal of Solids and Structures* 41 (16) (2004) 4503–4516.
- [19] J. Fernandez, M. Schenk, G. Prassinos, V. Lappas, S. Erb, Deployment mechanisms of a gossamer satellite deorbiter, in: 15th European Space Mechanisms and Tribology Symposium, Noordwijk, The Netherlands, 2013.
- [20] J.-F. Antoine, C. Visa, C. Sauvey, G. Abba, Approximate analytical model for hertzian elliptical contact problems, *Journal of Tribology* 128 (3) (2006) 660–664.
- [21] G. Huajian, C. Cheng-Hsin, L. Jin, Elastic contact versus indentation modeling of multi-layered materials, *International journal of Solids and Structures* 29 (20) (1992) 2471–2492.

- [22] S. Timoshenko, S. Woinowsky-Krieger, Theory of plates and shells, Vol. 2, McGraw-hill New York, 1959.
- [23] C. Calladine, The theory of thin shell structures 1888–1988, Proceedings of the Institution of Mechanical Engineers, Part A: Power and Process Engineering 202 (3) (1988) 141–149.