Experiments and Simulations of Centrifugal Deployments of Membranes Stowed with Spiral Folding

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Abstract

Centrifugal deployment dynamics of hexagonal membranes stowed with spiral folding are investigated. Centrifugal deployments have a possibility of unfolding large membranes without using extendable masts. Understanding and predicting the deployment dynamics are important to realize large membrane space structures. In this paper, deployment experiments of hexagonal membranes with coarse and fine spiral folding patterns are conducted in vacuum under gravity. Numerical simulations of the centrifugal deployments are also performed employing spring-mass system which models in-plane stiffness of thin membranes and enables fast numerical analysis. Modelings of buckling, creases, air drag and damping are added to the spring-mass system model and initial shapes are adjusted to experimental models. The experimental and numerical results are compared to examine the characteristics of the deployment dynamics and the validity of the numerical model. Simple models for buckling strength, air drag, damping and crease stiffness are introduced to evaluate their effects on the deployment dynamics. The experimental and numerical results are compared to discuss the deployment behaviors and the validity of the simulation.

1. Introduction

Membrane space structures have been attracting a lot of attention because of high packaging efficiency and light weight. Recently, solar sails using extremely thin large membranes have been developed in the world. In Japan, centrifugally deployed solar sails have been studied because of the possibility of unfurling large membrane without masts and the folding methods, dynamics of centrifugal deployment, deployment mechanisms, and so on have been investigated.\(^{(1)-(9)}\) Since ground dynamic experiments of huge membranes are not possible, a sounding rocket experiment\(^{(2)}\) and small-scale experiments in a vacuum chamber\(^{(3),(4)}\) have been conducted. Numerical methods such as spring-mass system models and finite element methods have also been studied to efficiently analyze the behaviors of thin membranes.\(^{(3),(7),(9)}\)

In this paper, centrifugal deployment dynamics of hexagonal membranes stowed with spiral folding\(^{(10),(11)}\) are studied. Ground experiments are conducted in a vacuum chamber using small-scale membranes to observe deployment dynamics under a constant rotation speed. Numerical simulation is also performed employing the spring-mass system model. Simple models for buckling strength, air drag, damping and crease stiffness are introduced to evaluate their effects on the deployment dynamics. The experimental and numerical results are compared to discuss the deployment behaviors and the validity of the simulation.

2. Deployment Experiments

2.1 Folding method and membrane models

Deployment experiments of hexagonal membranes with spiral folding\(^{(10),(11)}\) are conducted. The spiral folding is used as a test case for the study of centrifugally deployed membranes. Its folding pattern is designed to fold and wind a flat surface into a cylindrical shape by taking account of membrane thickness and calculating quasi-logarithmic-spiral folding lines to avoid wrinkles in stowed shape. Two hexagonal membranes with coarse and fine folding patterns are used in this study. The deployed shapes of experimental models are shown in Figs. 1 and 2. The membrane material is a polyimide film with thickness of 7.5\(\mu\)m. Their geometries and the mechanical properties of the material are summarized in Table 1.
2.2 Experimental setup

The experimental setup is illustrated in Fig. 3. The deployment experiments are performed in a vacuum chamber to reduce the effect of air resistance. The diameter and height of the chamber are 0.66 m and 1 m, respectively. The air pressure in the chamber is approximately 1 kPa. The folded membrane is stowed around a cylindrical hub fixed at the output shaft of a rotation mechanism driven by a stepping motor. The rotation speed can be controlled by a personal computer. In this experiment, the rotation speed is held constant at 3 Hz in order to reduce deformations due to gravity. The deployment dynamics of the membrane is shot by two cameras placed at the bottom and side of the chamber and quantified by image processing. Figure 4 shows a release mechanism of the membrane. A nylon string is winded around the membrane, the ends of which are fixed on the parts attached to the rotating shaft. A heating coil is moved by a solenoid and cut the upper end of the string to start deployment.

2.3 Experiment results

Figure 5 displays bottom and side views of the coarse folding model during deployment, respectively. It is observed that the membranes with spiral folding can be deployed smoothly and that the vertical deflection is small during deployment in spite of gravity. However, it is noted that both membranes fall down for a moment at about 1.3-1.4 seconds just after maximum deployments.

Figure 6 shows time histories of deployment rates of the membranes. The deployment rate is the ratio of the maximum
radius of the corners of the deploying membrane to the radius of
the original hexagon. The membranes deploy almost linearly
with time and reach maximum radii in approximately 0.9
seconds. The deployment speeds are almost independent of the
density of folding lines. Deployment rates do not reach 1.0 and
the rate of fine folding model is a little smaller due to creases
and gravity. It is also noted that the deployment rates fluctuate
after 0.9 seconds since the membranes are unrolled excessively
after the maximum deployments and in-plane and out-of-plane
vibrations are excited.

3. Numerical Modeling

3.1 Spring-mass system model

Spring-mass system models (7)(8)(9) have been developed for
simple and fast numerical simulations of dynamic behaviors of
thin membranes. In this paper, one of the models using
triangular elements (9) as shown in Fig. 7 is used to discretize
hexagonal membranes with curved folding lines.

In Fig. 7, \( E, \rho, h \) and \( S \) represent Young’s modulus, density,
thickness and area of the element, respectively and \( L_i (i=1,2,3) \)
denote natural lengths of the sides of the triangle. The mass of
the element \( \rho h S \) is equally distributed to the lumped masses
\( m_1, m_2 \) and \( m_3 \). The spring constants \( k_1, k_2 \) and \( k_3 \) are determined by
Eqs. (1).

\[
\begin{pmatrix}
  k_1 \\
  k_2 \\
  k_3
\end{pmatrix}
= B^{-1}
\begin{pmatrix}
  1 \\
  1 \\
  1
\end{pmatrix}, \quad
B_{ij} = \frac{p_{ij}^2 L_j^2}{E h S},
\]

\[
p_{ij} = 1 - \frac{4(1+\nu)S^2}{L_i^2 L_j^2}(1-\delta_{ij}),
\]

where \( \nu \) and \( \delta_{ij} \) denote Poisson’s ratio and
Kronecker delta, respectively. A whole spring-mass

system model for a membrane is constructed as follows:

1. Partition a membrane into triangular elements
2. Calculate spring constants and masses of each element
3. Superpose all the springs and masses
4. Perform additional modeling
5. Formulate equations of motion

The additional modeling includes buckling strength, crease
stiffness, damping and air drag.

3.2 Buckling strength of springs

In order to simulate buckling of thin membranes, the
restoring force of a spring is assumed to become constant when
the length of the spring is less than a critical value \( l_{ce} \) as shown
in Fig. 8. The critical value \( l_{ce} \) is expressed using Euler buckling
strength of slender column as:

\[
l_{ce} = L_i - \alpha \frac{E h^2}{12 L_i}.
\]

The coefficient \( \alpha \) is determined so that the membrane
withstands gravity at the start of deployment as experimental
models when centrifugal force is small. If \( \alpha \) is not large enough,
the membrane collapses as shown in Fig. 9.

3.3 Crease model

Tensile property of creased thin materials has been
studied (12)-(14) to evaluate effective in-plane stiffness. In this
research, a simple crease model is newly introduced to the
spring-mass system model to take account of the rough effects
of creases as follows.

Two triangular elements ABC and ABD that contain a
crease are considered as shown in Fig. 10. It is assumed that the
crease generates concentrated forces \( F_c \) and \( F_D \) according to the
difference between angle \( \theta \) and natural crease angle \( \theta_0 \) and the
distances between the masses and the crease. \( F_c \) and \( F_D \) are
assumed to be orthogonal to the triangles and are described by:

\[
F_C = \frac{E J L_{AB} (\theta - \theta_0)}{l_{CE}^2}, \quad
F_D = \frac{E J L_{AB} (\theta - \theta_0)}{l_{DF}^2},
\]

where \( E \) and \( J = h^3/12 \) denote Young’s modulus and
geometrical moment of area per unit width, respectively and
\( l_{AB}, l_{CE} \) and \( l_{DF} \) are lengths when the membrane is deformed.

Equations (3) were obtained by performing nonlinear finite
element analysis of a cantilever beam subjected to a follower force at its end. Equations (3) give good approximation when the angle is large. \( F_A \) and \( F_B \) are forces acting on the masses A and B to cancel rigid-body motion of the triangular elements and described in vector form as:

\[
F_A = l_{BE} F_C - l_{BF} F_D, \quad F_B = l_{AE} F_C - l_{AF} F_D. \tag{4}
\]

### 3.4 Damping and air drag

Velocity-proportional dampers are added parallel to the springs to take account of structural damping. The damping coefficients \( c_i \) are described as:

\[
c_i = 2\zeta \sqrt{k_m m_i}, \quad m = (m_j + m_k)/2, \quad (i, j, k = 1, 2, 3), \tag{5}
\]

where \( \zeta \) denotes damping ratio.

In order to take account of residual air in the chamber, the air drag force which act on an element is distributed to three masses. The force applied to a mass is assumed to be

\[
D = \rho_{air} S_n V^2 / 6, \tag{6}
\]

where \( \rho_{air}, V \) and \( S_n \) denote the density of air, the average velocity of the masses and the area of element projected to the direction of the velocity, respectively.

### 3.5 Modification of initial radius

Since the experimental membrane model in stowed configuration is constricted by the nylon string, initial positions of the masses of numerical model need to be modified. The radii of innermost masses are kept unchanged while the radii of outermost masses are reduced to those of experimental model. The radii of internal masses are reduced proportionally.

### 3.6 Equation of motion

The equations of motion of the system are described using rotating cylindrical coordinate system as:

\[
d^2 \begin{pmatrix} r \\ \theta \\ z \end{pmatrix} \over dt^2 = \begin{pmatrix} F_r + r \dot{\theta}^2 \\ m \dot{\theta} \\ F_z \end{pmatrix}, \tag{5}
\]

where \( F_r, F_\theta \) and \( F_z \) denote forces applied to the mass which consists of forces by the springs, creases, damping and air drag. These equations are numerically integrated by the Runge-Kutta method.

### 3.7 Numerical models for the experimental membranes

Figures 12 and 13 show deployed and folded shapes of the coarse and fine folding models discretized with the triangular elements, respectively. Main parameters of the models are shown in Table 2. Natural crease angle is assumed to be \( \pi/4 \). Buckling parameter \( \alpha \) was found to be around 100 for both models to withstand gravity at the start of deployment.

### 4. Numerical results

The numerical simulations of the centrifugal deployments were performed employing the spring-mass system models. The numerical models were also deployed smoothly. Examples of membrane shapes of the coarse folding model during and

<table>
<thead>
<tr>
<th>Table 2. Main parameters of simulation models.</th>
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<tbody>
<tr>
<td>Model</td>
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<tr>
<td>Numbers of masses</td>
</tr>
<tr>
<td>Number of Elements</td>
</tr>
<tr>
<td>Number of springs</td>
</tr>
<tr>
<td>Number of crease edges</td>
</tr>
<tr>
<td>Natural crease angle</td>
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<tr>
<td>Buckling parameter ( \alpha )</td>
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<tr>
<td>Damping ratio</td>
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after deployment are illustrated in Fig. 14. Crease lines are successfully observed in Fig. 14(c) when the membrane falls down at 1.4 seconds as observed in the experiment.

Figure 15 shows the time histories of deployment rates by the simulation compared with experimental results. It is found that the numerical results are in good agreement with the experimental results although numerical models are still deployed a little larger. The deployments progress a little faster than the experiments and maximum deployments are achieved at about 0.82 seconds in both numerical models. The reason for this is probably that the release mechanism takes a little time to release the membranes in the experiments. Figure 16 shows the results of basic numerical models in which buckling parameter $\alpha$ equals one and crease stiffness, damping and air drag are ignored. The results of the simulations are rather different from the experiments. Deployment rates vibrate during the deployment and decrease severely after the maximum deployment. It is confirmed that the additional modeling improve simulation results significantly.

The effects of each additional modeling in coarse and fine folding models are displayed in Figs 17 and 18, respectively. Similar tendencies are found in both models. Air drag slows down deployment speeds and moderates vibrations after deployment. The crease stiffness model slightly decreases deployment rates but the differences are not large enough to agree with the experimental results. The buckling strength significantly affects fluctuations during deployment and deployment speeds.

The maximum deployments are achieved along with out-of-plane vibrations around the equilibrium shape under gravity and the vibration amplitudes of the numerical models are large enough to reach almost full deployments.
Improvement of the crease model and the modeling of bending stiffness may be necessary to suppress excessive deployment rates and vibrations. Further study will be needed on both the experiments and simulations to further improve the difference between the experiments and simulations.

5. Conclusion

Centrifugal deployment experiments of the membranes stowed with spiral folding were conducted and compared with the numerical simulations of spring-mass system models. The experiments demonstrated that the membranes could be smoothly deployed and that in-plain and out-of-plain vibrations were excited after deployments. It was also confirmed that the spring-mass system models could simulate the deployment dynamics with fairly good accuracy by taking account of the buckling strength, crease stiffness, damping and air drag and by adjusting initial conditions to the experiments.

Vibrations of membranes inevitably occur along with dynamic deployments and the rotating membranes may be unstable in steady state. The understanding of the deployment behaviors and the method to control the dynamics will be important to realize stable membrane structures. For that purpose, the development of fast and reliable numerical simulation methods and a methodology for predicting on-orbit behaviors of the structures based on the numerical simulations and small-scale ground experiments are needed.

References