A Novel Deployable Reflector using a Lazy Tong Truss

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Received on:8 /6/2011 & Accepted on:2/2/2012

ABSTRACT

In this paper, a new concept for a deployable parabolic reflector used in SAR satellite has a support based on a two-dimensional lazy tong mechanism is proposed. Two sets of lazy tong mechanisms are used, one at each side, to support the structure through its operation (deploying and folding). The deployment of the structure is controlled by the vertical inward movement of the ends of the first two bars in each mechanism. This movement is directed by the strain energy stored in the helical springs that work at those places. A preliminary design of the quarter scale model of the suggested reflector in deployed configuration is modeled and analyzed using ANSYS software program. The value of the maximum deflection and Von Mises stress due to effect of inertia load only are obtained for the composite material (T300/Hexcel 8552) of the reflective surface and hard aluminum alloys (2024-T3) of the truss. They were 0.465 mm and 23.3 MPa respectively. The linear buckling behavior of the structure is as well studied. The first eigenvalue factor was about 19.639. The theoretical total deploying time and velocity is considered and their magnitudes are about 0.14 sec. and 0.46 m/s respectively. The packaging volume was about $1/6^{th}$ of deploying volume. The total estimated mass of this quarter scale reflector was approximately 2.5 kg.

Keywords: deployable reflector, lazy tong mechanism, stowed, deployed.

العاكس الجديد القابل للانتشار باستخدام آلية الـــ (Lazy Tong)

الخلاصة

في هذا البحث، تم تطوير و تعديل عاكس ذو شكل قطع مكافئ قابل للانتشار يعمل بهيكل ساند من نوع آلية (Lazy tong) ثنائي الأبعاد. عملية انتشار العاكس يتم التحكم بها عن طريق الحركة الشاقولية لاحد طرفي الآلية وهذا يتم بفعل الطاقة الانفعالية المخزونـــة للنابضـــين الحلـــزونيين الموضوعين عند طرف الآلية الثابت اما الطرف الثانية للآلية فتكون حرة وبحركتها الأفقية سوف يأخذ السطح العاكس شكله النهائي وذلك بعد مرور جانبيه الطويلين على مجموعة من المسامير الموجودة على الجوانب الداخلية لقضبان الساند المثبتة بكيفية تعطي الشكل النهائي للعاكس. تـــم تصميم نموذج تمهيدي بمقياس 1/4 من النموذج الاصلي وذلك باستخدام برنـــامج ال ANSYS. وتم افتراض ان السطح العاكس قد صنع من المادة المركبة (T300/Hexcel 8552) وان الآلية مصنوعة من قضبان (links) من الألمنيوم السبائكي القاسي (2024-T3). وقد تم احتساب مقدار التشوه الأعظم وكذلك الإجهاد الأعظم المكافئ للهيكل وهو منتشر فـــي الفضـــاء حيـــث كانـــا

https://doi.org/10.30684/etj.30.11.6
2412-0758/University of Technology-Iraq, Baghdad, Iraq

0.465mm و23.3 MPa على التوالي. وكذلك تم دراسة linear buckling لهيكل العاكس. وقـد استكمل ايضا حساب السرعة والوقت اللازم لانتشار العاكس .

Nomenclature

a_i, a_{avi} = instantaneous and average deploying acceleration (m/s^2)
E_1, E_2 , G_{12} = longitudinal, transverse, and shear Young's modulli (N/m^2)
H_{max}, H_{min} = maximum and minimum height of the slider (mm)
L_{max} = maximum deploying length (mm)
M_{shell}, M_{truss} = Total mass of the reflective surface, and truss (kg)
m = mass of one slider (kg)
T_i, V_i = instantaneous deploying time (sec.) and velocity (m/s)
X_{max} = maximum deploying length of one pair of the mechanism (mm)
x_i = instantaneous deploying horizontal movement (mm)
y_i = instantaneous deploying vertical movement (mm)
θ_i , θ_1, θ_2, = instantaneous, initial and final angle of deployment (º)
Δ_1 = minimum. spring deflection (mm)

INTRODOCTION

Deployable antennas are an essential element of future space technology. The maximum diameter of launch vehicles limits the diameter of a satellite to about 4 m [1], but antennas larger than this are required for earth observation, astronomy and communications.

Deployable antennas can be split into two wide categories, those with a furlable reflector surface and those with a solid reflector surface. Furlable antennas are antennas where the surface itself can be folded; when deployed, the reflector surface is stretched out to form the required shape. Therefore, the flexible surface are much preferable for relatively large deployable reflectors [2].

This paper deals with one of the classic problems in furlable reflectors, how to make a deployable reflector that is easy to design, inexpensive to make, and works well. A single solution, satisfying all of these general requirements, has yet to be found. Instead, several partial solutions have been identified, in the form of different structural concepts, such as solid surface deployable reflector, pantographs, coilable masts, etc. The particular solution relevant to this paper is the deployable truss structure, whose simplest, two-dimensional form is the lazy tong, an assembly of straight links of equal length and rectangular cross-section connected by pivots in the middle and at the ends. The pivot in the middle of each pair of link is sometimes referred to as a scissor joint, because the pair of links behaves in the same way as a pair of scissors.

The final support is a deployable truss obtained by joining two lazy tongs at the right and left side of the structure. Due to use of those mechanisms, this will give the reflector a good packaging, fast deployment and folding, easy control, and relatively high stiffness to weight ratio. The structure designed in this paper is purely spring driven.

THE ORIGINAL REFLECTOR

Figure 1, represents the traditional satellite configuration. From this the large main reflector surface is only required in this paper. The main full scale reflector has an aperture of 6.5x3.2 m^2; the surface itself forms a 3.2 m wide offset parabolic cylinder with arc-length of 7.888 m. European Aeronautic Defense and Space Company (EADS) Astrium had estimated the mass of this reflector structure made from lightweight, curved panels with self-locking hinges to be around 79 kg [3].

DESCRIPTION OF THE NEW REFLECTOR

In general, the structure of the new suggested reflector comprises of two main parts. The first part is the deployable support (truss) and the second part is the reflective surface.

a) The Deployable Truss Structure Mechanism

The truss is constructed from two lazy tong mechanisms. Each mechanism contains of ten pairs of straight and equal length links. The pairs arranged one beside another to give the required overall aperture dimensions when the truss is fully deployed. In each pair there are two links pivoted together from the middle. The pairs are interconnected from the ends of their links by hinges to form the structure, as shown in Figure 2. The links have rectangular cross section and are made from hard aluminum alloys. Each link contains three holes one at each end and the third is at the middle. In addition, each mechanism has two vertical extensible struts. The strut consists of a slider, an extension spring, and a hollow rod (tube), as shown in Figure 2. The sliders connected to the first two ends of the truss to give it its movement. The springs are in tension when the structure is in stowed configuration. Therefore, the structure designed in this paper is purely spring driven.

b) The Reflective Surface

In present work, the reflective surface of our structure has the same required shape and dimensions of the traditional one. Its parabolic shape is formed through deploying process of truss as follows: one edge (short edge) of reflective surface is attached to the end of truss that jointed to the sliders (fixed end in x direction) while the second edge is connected with the free end of the truss. Therefore, when the truss is starting to deploy, its free end pulls the reflective surface away from the fixed end. During this movement, the long edges of the reflective surface pass over several cylindrical pins that are located on the inner surfaces of the links by a certain way to give the required parabolic shape of the surface.

So, the reflective surface in this concept is based on a thin flexible composite shell that is elastically folded and deployed as a single piece, without any joints or hinges. In this paper, a quarter scale of this reflector has been designed and analyzed.

PRINCIPLE OF THE MECHANISM

As mentioned above, that the mechanism is a purely spring driven. When the structure is completely stowed, the springs are fully tensioned and exert forces on the sliders in the y and negative y direction. The sliders in this moment are constrained to move. In the moment of releasing the restraint and due to the strain energy stored in the springs, the sliders come to be free to slide vertically. In this instant, the mechanism will gradually start its movement in the x direction and then fully deployed. The vertical movement of the slider is proportional to the sine of the angle θ_i, as shown in Figure (3). The total vertical movement of the slider is $y_{max} = H_{max} - H_{min}$ and fully deployment length of the structure is L_{max} = no. of pairs * X_{max}.

When the structure is fully deployed the springs are also remain in tension for small values to keep the structure opened.

From Figure 3, the relationship between the vertical and horizontal movement of the truss can be obtained as follows:

$$y_i = H_{max} - \frac{l}{2} * \sin(\theta_i) \tag{1}$$

$$x_i = l * \cos(\theta_i) \tag{2}$$

$$\therefore y_i = H_{max} - \frac{l}{2} \sin\left(cos^{-1}\left(\frac{x_i}{l}\right)\right) \tag{3}$$

Where y_i is the instantaneous vertical movement of point A, $0 \leq y_i \leq y_{max}$, see Figure 4.

x_i is the instantaneous horizontal movement of point A, $X_{min} \leq x_i \leq X_{max}$ (for one pair of links)

θ_i is the angle of deploying, $\theta_1 \geq \theta_i \leq \theta_2$

l is the length of the link

At any instant of time, the distance traveled by any points of the truss (e.g. pivots and hinges centers) in x-direction is not equal to the deflection of the spring. This factor contributes to the non-linear behavior of the mechanism.

According to Newton's second law of motion (F = ma), the force applied by the springs is always equal to the product of the mass (m) of the vertical member and its acceleration. Therefore, for one spring, Figure 5:

$$M_t \frac{\partial^2 y_i}{\partial t^2} - F_{si} + F_f + mg = 0 \tag{4}$$

$$F_{si} = K_s[y_i + \Delta_1] \tag{5}$$

$$\Delta_1 = H_{min} - l_o \tag{6}$$

Where F_{si} is the instant force applied by the springs, l_o is the free length of the spring, F_f is the friction force between the slider and the tube, and M_t is the total mass affecting on one spring ($1/4M_{shell} + 1/4M_{truss}$).

The minimum stiffness of the spring can be determined from,

$$K_{s,min} = \frac{F_f+mg}{(y_i+\Delta_1)} \tag{7}$$

The maximum value of $K_{s,min}$ is taken to be the minimum required stiffness of the spring.

After determination of $K_{s,min}$ required for the spring, the instantaneous time, velocity and acceleration with respect to position are calculated, as follows:

$$a_i = \frac{K_s(y_i+\Delta_1)-F_f-mg}{M_t} \tag{8}$$

And

$$a_{avi} = \frac{a_{i-1}+a_i}{2} \tag{9}$$

$$T_i = \frac{-V_{i-1}+\sqrt{V_{i-1}{}^2+2a_{avi}*(y_{i-1}-y_i)}}{a_{avi}(y_i)} \tag{10}$$

And

$$V_i = V_{i-1} + a_{avi} * T_i \tag{11}$$

GEOMETRY, PROPERTIES AND LOADING

The particular structure of the reflective surface that has been analyzed forms an offset paraboloid of equation, $x^2 = 4py$, where p is the focal length and is about 772.25 mm, see Figure 6. The arc-length and width of the reflective surface is about 1975 mm and 800 mm respectively. The reflective surface was assumed made of an elastic homogeneous orthotropic material from thin sheet of carbon fiber (T300) /matrix (Hexcel 8552), has a uniform thickness h_S = 0.3 mm, and lamina stacking is $[0°/90°]_S$.

The links have been assumed made from hard aluminum alloys 2024-T3. Each link has a total length L_L = 274 mm, a width w_L = 10 mm, and a thickness t_L = 1 mm. The properties of the material used in this work are listed in Table 1.

Table (1) Fiber, matrix, and aluminum material properties

Properties	**T300 [4]**	**Hexcel 8552 [4]**	**AL [5]**
Density, ρ [kg/m^3]	1760	1301	2770
Longitudinal stiffness, E_1 [N/mm^2]	233000	4760	72700
Transverse stiffness, E_2 [N/mm^2]	23100	4760	72700
Shear stiffness, G_{12} [N/mm^2]	8963	1704	27600
Poisson's ratio, ν_{12}	0.2	0.37	0.31
Maximum strain ε_{max} %,	1.5	1.7	-----

The reflector body is subjected to gravity load of 0.02 m/s^2 along the focal axis, the y-axis in Figure 6.

COMPUTATIONAL DETAILS

The geometry and the finite element mesh were created in the ANSYS CAE,[6]. To keep computational times low, half of the structure was analyzed (for static analysis), using the appropriate symmetry boundary conditions. Figure 7 shows the loading and boundary conditions nature on the reflector body. The red vector shown in Figure 7 represents the gravitational load applied to the whole reflector.

The reflective surface was modeled as linear, 8-node with six degrees of freedom at each node, quadrilateral, and 3D shell element (SHELL281). This shell element may be used for layered applications for modeling laminated composite shells or sandwich constructions.

All members represent the support (truss) were modeled using element BEAM188. This element is a linear, quadratic, 2-node with six degrees of freedom at each node, and 3D beam element. In addition, a multi-point constraint element, MPC184 rigid beam element, has been used to model a rigid constrain between two deformable bodies (truss and reflective surface), see Figure 7.

Reliable convergence was achieved with a fine mesh of 1275 and 172 elements of SHELL281 and BEAM188 respectively per full structure.

A typical analysis consisted of three steps. The first being a static analysis to determine the maximum displacement and maximum equivalent stress of the structure. The second step being a linear buckling (eigenvalue) analysis to determine the first eigenvalue of the reflector, whereas the third step being a modal analysis to obtained the fundamental natural frequency, which is used as a measure of reflector's stiffness in the deployed configuration.

RESULTS AND DISCUSSIONS

The results that obtained from the analysis of the reflector are categorized in the following points:

1. The magnitudes of vertical (y_i) and horizontal (x_i) movement for the first pair of links have been determined from equations 1 and 2 with respect to the angle of deploying (θ_i). The angle of deploying changes from 85º to 43.174º for fully deployed. Therefore, the maximum vertical displacement that the slider must be moved is limited at about 41mm and that for the maximum horizontal displacement that a one pair can be covered is about 192.6 mm. From that, the free end of reflector executes a total movement about 1926 mm. The minimum spring's stiffness is found using equation 7, and it was between 33.22 to 78.28 N/m. The variations of y_i and x_i against the deploying angle have been plotted in a single graph with the minimum stiffness of the spring as shown in Figure 8.
2. From equations 10 and 11, the total time required for deploying the reflector and the velocity at instant of full deployment has been obtained and they were about 0.14 s and 0.46 m/s respectively. The variation of the

linear velocity, linear acceleration, and time against the angle of deployment is illustrated in Figure 9.

3. From the static analysis of the structure, the magnitudes of maximum displacement (USUM) and maximum Von Mises stress (SEQV), due to applied inertia load only, are found. They were 0.465 mm and 23.3 MPa and their natures along the reflector structure are shown in Figures 10 and 11 respectively. Note that the maximum displacement value occurs at the last third of the reflective surface.
4. The linear buckling of the reflector is also performed in this paper. The minimum eigenvalue factor (load multiplier) is obtained and it was about 19.64. Figure 12 illustrates the first buckling mode shape of the reflector. It is clear that the reflective surface with the shell thickness of 0.3 mm was observed to fail purely due to global skin buckling and make the reflector to twist around the longitudinal axis.
5. Finally, the free vibration behavior of the structure is studied. Figure 13 displays the first mode shape of this behavior. The fundamental natural frequency of this structure was about 0.8846 Hz. It is a pure translational motion of the truss in + z direction.
6. The total mass of the QR scale model was estimated at about 2.5 kg.

CONCLUSIONS

1. A new idea of large deployable reflector for SAR applications has been presented.
2. For quarter scale, it has been estimated that this proposed model leads to a structure weighing 2.5 kg including the mass of the pins, clips, washers, and frame. This is 7.9 times lighter than the traditional reflector structure.
3. The initial analysis of the proposed structure showed that the structure is not need to add any stiffening to its reflective surface.
4. A slight increasing of the link's thickness leads to increase the overall stiffness of the structure.
5. The deploying time is short and this may be destroying the reflector itself through deployment. Therefore, it is better to use springs have low minimum stiffness.
6. The lowest natural frequency of the proposed concept in the deployed configuration was 0.885 Hz while the required value for the traditional one was 0.4 Hz or even less. This frequency can be improved by joining the two mechanisms with one or more intermediate hollow circular link.

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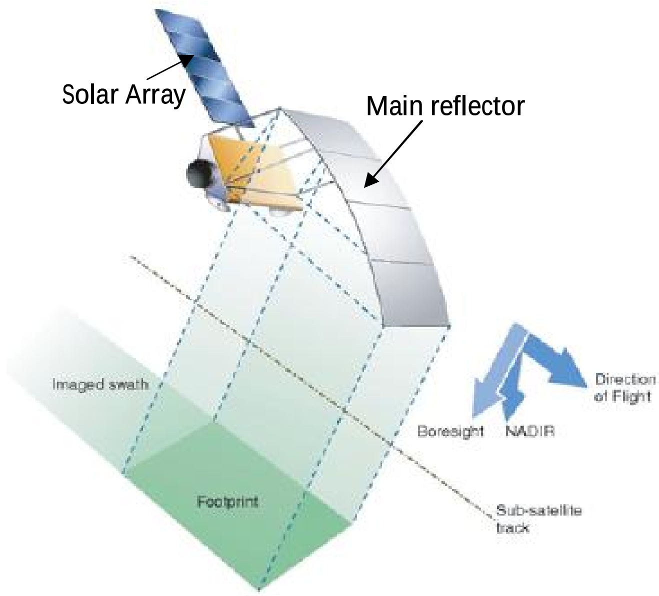

Figure (1) Traditional reflector, flight configuration.

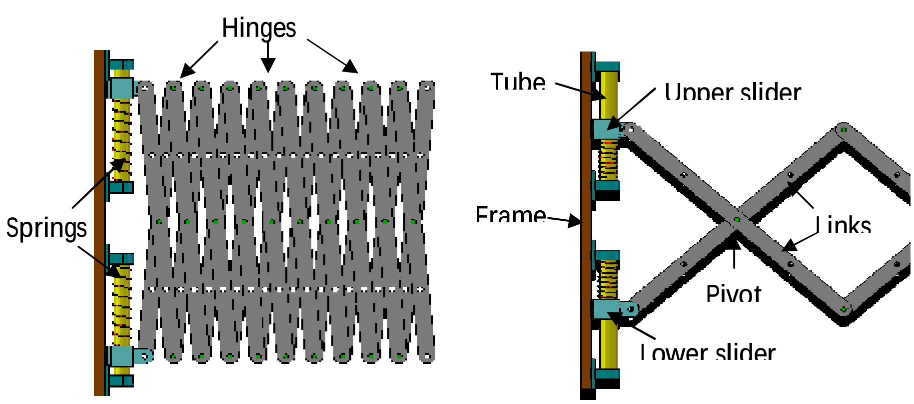

Figure (2)Solid Works representation of folding and deploying Truss (reflective surface is not shown).

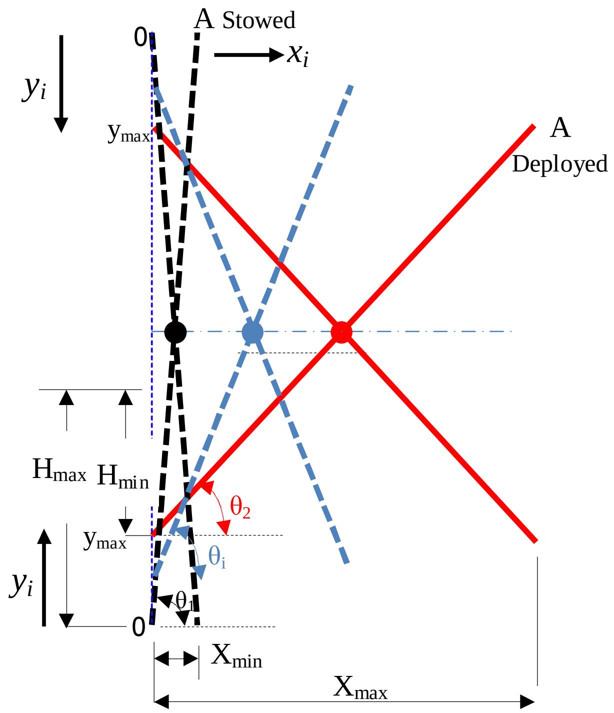

Figure (3) A sketch diagram of the stowed and deployed Configuration for one pair of the truss.

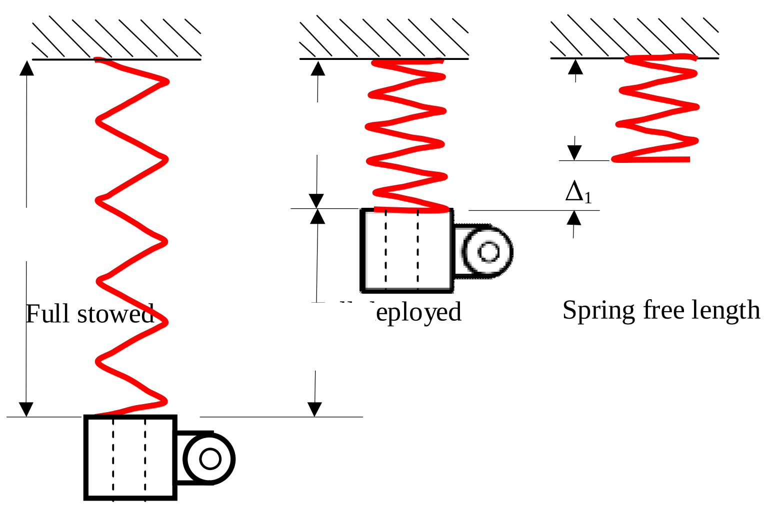

Figure (4) Length of the lower spring at fully stowed, Deployed, and free length of the spring.

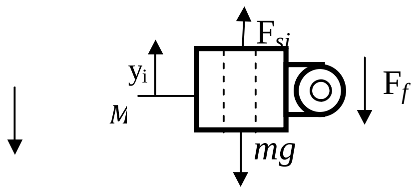

Figure (5) free body diagram of the lower moving Mass during deployment.

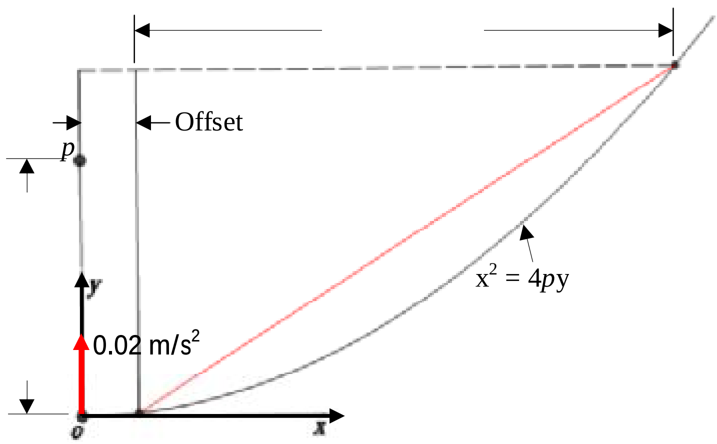

Figure (6)Profile of the reflective surface.

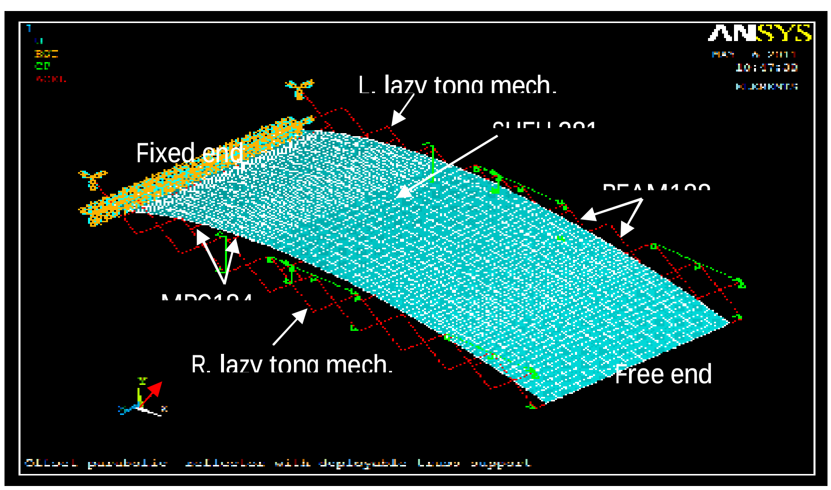

Figure (7) Loading and boundary conditions of quarter-scale of the deployable reflector.

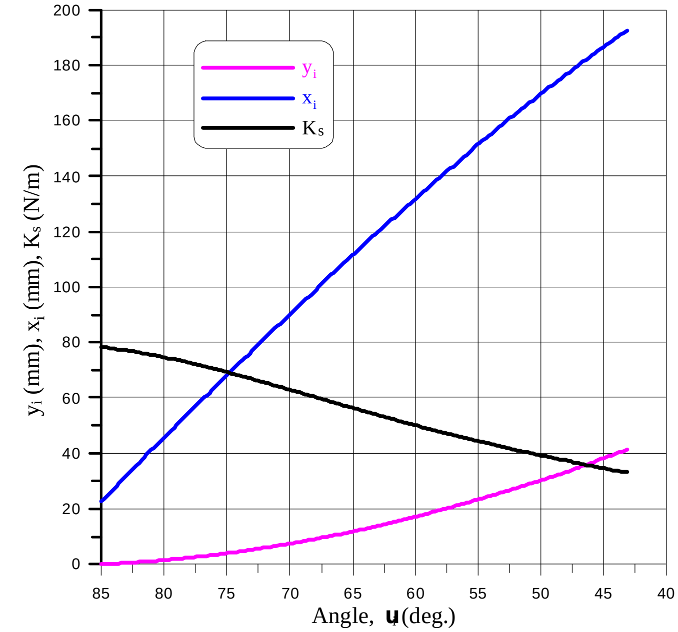

Figure (8) Variation of y_i, x_i, and spring stiffness K_s through Deploying process for one pair of the truss structure.

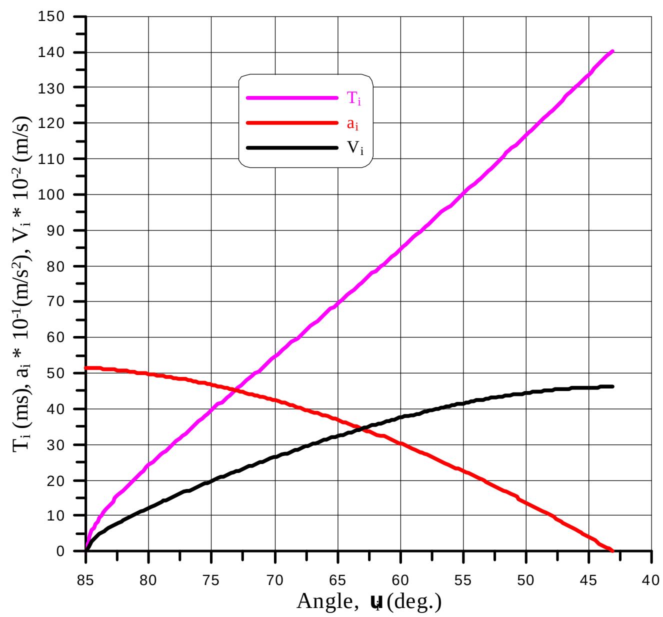

Figure (9) Time, velocity, and acceleration analysis Through deploying process.

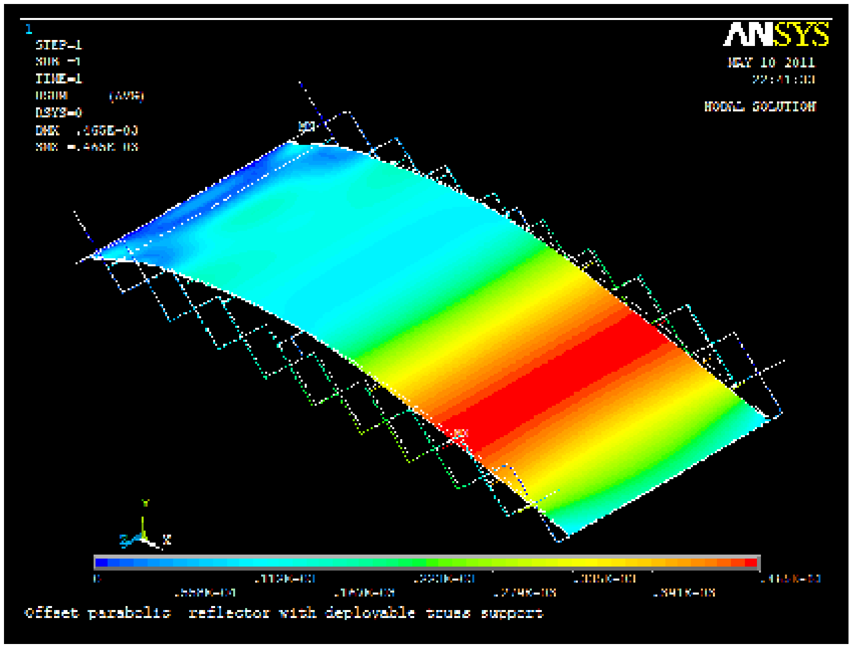

Figure (10) Distribution of maximum displacement (m) of QR scale reflector.

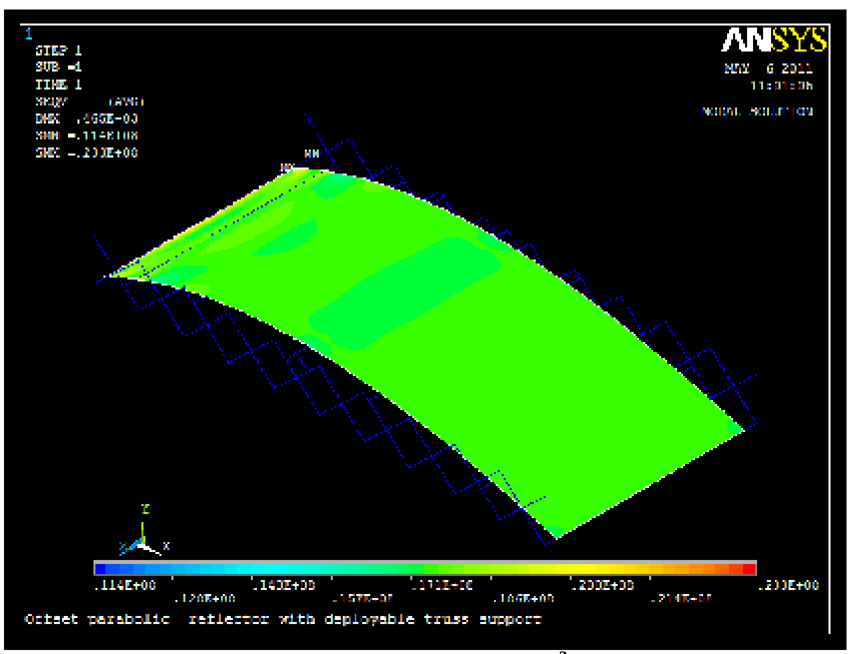

Figure (11) Von Mises stress distribution (N/m^2) of QR scale reflector.

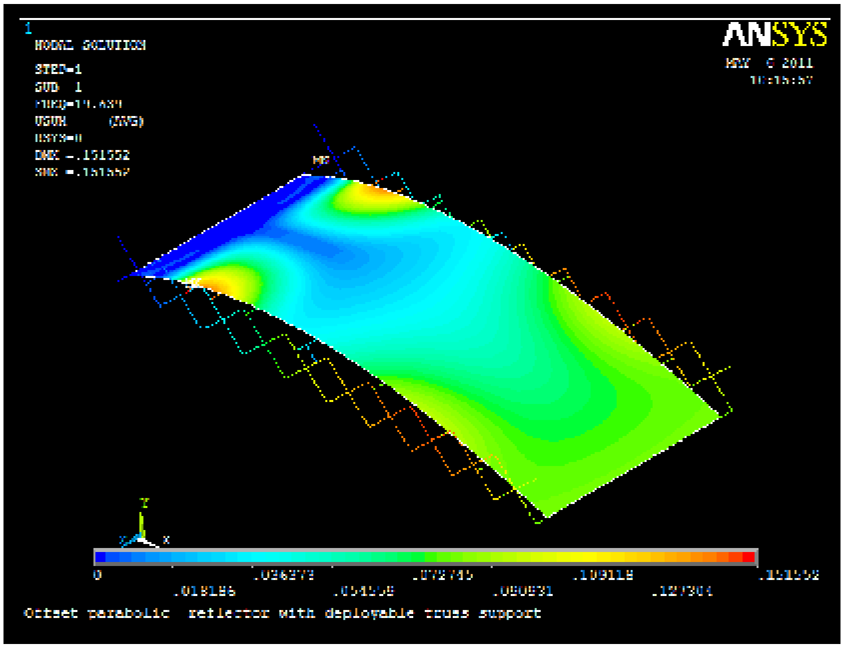

Figure (12) First mode of linear buckling (eigenvalue = 19.64) of QR scale reflector.

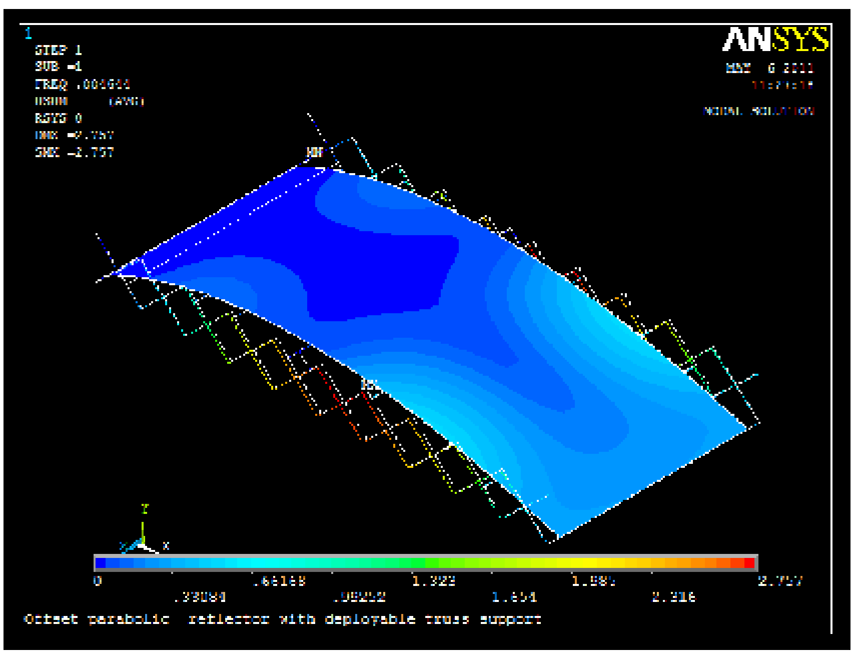

Figure (13) Fundamental natural frequency (Freq. = 0.8846 Hz) of QR scale reflector.