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Investigating the effect of thermal cycles on the recovery of tensile strength in self-healing composites under impact loading

A. Talezadehlari[1], A. Nikbakh[2]

[1]Department of Mechanical Engineering, University of Larestan, Lar, Iran
[2] New Technologies Research Center (NTRC), Amirkabir University of Technology, Tehran, Iran

ABSTRACT: Here the healing process of glass/epoxy composites is studied experimentally. The healing agent is composed of ML506 epoxy resin and HA-11 hardener which is charged into microtubes. The microtubes are interleaved between the first and the second and the fifth and the sixth layers. The initial damage is introduced to the samples by drop weight impact tests and the recovery percentage of the tensile strength due to healing process is measured by tensile test. The effects of healing time (without thermal cycles), number of healing units and thermal cycles are investigated on the recovery of the tensile strength. Composite samples containing 8, 16 and 32 healing units are studied in the periods of 1, 6 and 12 days after the initial damage. Also, some damaged samples are set to 1, 3, 5 and 7 thermal cycles and after one day, tensile tests are carried out. The results show that without thermal cycles the healing process is almost completed after 6 days with an 83% recovery. In addition, the maximum amount of tensile strength recovery is equal to 86% which is related to the samples with 32 healing units after 12 days. This amount of healing efficiency can also be achieved by means of 5 thermal cycles. This is also true for thermal cycles where the effect of thermal cycles is tangible up to 5 cycles.

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1- Introduction

With the improvement of various sciences and the use of mechanical elements such as beams and rods in the devices and equipment of various sciences, the need for a more accurate model is felt in order to better describe these elements. Beams are widely used in various equipment, from very small dimensions such as microbeams in measurement devices and sensors to very big equipment such as airplane airfoil and industrial robots, and therefore require accurate modelling. One of the factors that influence the accuracy of modelling a problem and has received a lot of attention from researchers in recent years is the use of an accurate geometric model with as little approximation as possible to express the kinematics of the problem, which in most cases, accurate kinematics leads to a non-linear differential equation. Beams with small oscillation amplitudes can generally be modelled using a linear model, but when the amplitude of oscillations is large, the linear model and the use of linear geometry lose their validity. There are some factors that cause the beams to expect high vibration amplitude performance, among which the use of beams in electromechanical systems can be mentioned. In some cases, in order to have a large oscillation range and receive a larger signal, the beam is vibrated near the resonant frequency [1]. Another application of oscillation with a large amplitude, we can mention beams made of elastomer materials, which are used as resonators due to their high deformability [2].

*Corresponding author's email: anikbakht@aut.ac.ir

Generally, in the research conducted, the investigation of nonlinear geometry caused by considering more terms of strain can be divided into two general categories, analysis under large displacements with small strain, which is generally known as the von-Karman model in research, and the other case, analysis under large displacements with large strain, which is generally expressed as the limited strain model.

In the Von-Karman theory, several nonlinear terms are used in addition to the conventional linear model to express strains, and practically, large displacements, small strains, and rotation are considered in those relationships [3].

In the other theory called finite strain theory, more non-linear terms are used to express the strain, and practically the strains are considered finite, contrary to the assumption of being small.

Regarding the nonlinear displacement field under the title of finite deformation, the researches have been focused only on the static bending of the beam. In his first article, using the finite element method, Lee investigated the static bending of the beam without considering any assumption of displacement and small strain, and by examining some examples, he showed the correctness of the stated relationships [4].

Ronald investigated the static bending of the Timoshenko beam by considering the finite deformation theory for the bent beam under concentrated and extended load and solved the obtained equations with the finite element method and examined a semicircular beam to check the accuracy of the results [5].

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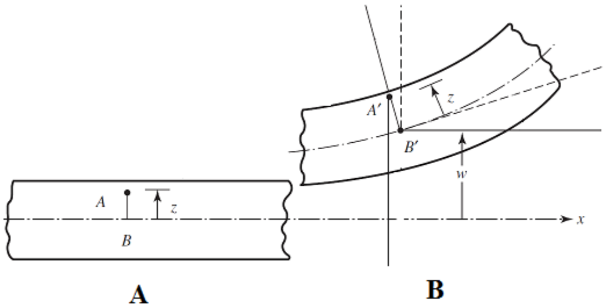

Fig. 1. A. Beam before deformation, B. Beam after deformation [4]

Table 1. Comparison of the results of three different van-Karman theories, reference [7] and the present research

Different theories	ω_n (Hz)	jumping point σ_p
Von-Karman theory	14923	1151
Reference [5]	14973	1161
present research	14748	1155

Beheshti investigated their static bending based on the Timoshenko model by using the nonlinear displacement field and considering the limited strain model and the size effect for beams with small dimensions and several examples of beams with boundary conditions were used to verify the accuracy of the obtained equations. and examined different loadings [6].

2- Methodology

In this research, in order to obtain an accurate model of dynamic bending of the beam, the finite deformation theory has been used, that is, the approximation of the displacement field has been avoided and the deformation field of the beam has been considered as non-linear, on the other hand, to express the strains, a more accurate non-linear theory, i.e. the finite strain model, has been used. Therefore, based on these assumptions, a more accurate and different nonlinear model can be obtained from the form of other articles, i.e. Duffing's differential equation, which has been fitted from the design and better performance of equipment that uses a vibrating beam with a large range of oscillations. Also, one of the other advantages of the current beam model is the more accurate prediction of the jump point frequency, which is very important in electromechanical systems to have a stable response. In other words, in these systems, the vibration response of the beam near the resonant frequency is used as a measurement parameter. Therefore, in order to have a stable response, the performance of such systems must be far from the jump point.

To model the beam based on the Euler-Bernoulli beam model, the nonlinear displacement field can be considered as Figure 1, where it is assumed that the cross-section perpendicular to the beam axis remains perpendicular to the beam axis after large nonlinear deformation

Therefore, the displacement field equations can be written as equation (1) [4]:

$$
\begin{aligned}
&U_1(\mathrm{X},\mathrm{Z},\mathrm{t}) = -\mathrm{z}\sin(\theta) = -\mathrm{z}\frac{W'}{\sqrt{1+W'^2}} \\
&U_2 = 0 \\
&U_3(\mathrm{X},\mathrm{Z},\mathrm{t}) = W(\mathrm{X},\mathrm{t}) - \mathrm{z}(1-\cos(\theta)) = W(\mathrm{X},\mathrm{t}) - \mathrm{z}\left(1-\frac{1}{\sqrt{1+W'^2}}\right)
\end{aligned} \tag{1}
$$

Based on the Green-Lagrange strain and considering the displacement field of equation (1), the only non-zero strain component is obtained according to equation (2):

$$
\begin{aligned}
E_{11} &= \frac{\partial U_1}{\partial X_1} + \frac{1}{2}\left(\frac{\partial U_1}{\partial X_1}\right)^2 + \frac{1}{2}\left(\frac{\partial U_3}{\partial X_1}\right)^2 \\
&= z^2\left(\frac{W''^2}{2(1+W'^2)} - \frac{W'^2 W''^2}{(1+W'^2)^2} + \frac{1}{2}\frac{W'^4 W''^2}{(1+W'^2)^3}\right. \\
&\left. + \frac{1}{2}\frac{W'^2 W''^2}{(1+W'^2)^3}\right) - \frac{zW''}{\sqrt{(1+W'^2)}} + \frac{W'^2}{2}
\end{aligned} \tag{2}
$$

3- Discussion and Results

In this article, in order to obtain the governing equations of the beam, the field-displacement relation is considered without approximation and in a non-linear form. The strain-displacement relations were calculated using the Green-Lagrange relation and the non-linear form of the equations was obtained using Hamilton's method. Using the Galerkin method, the partial differential equation is converted into an equation with ordinary derivatives. To obtain the approximate response of free and forced vibrations, the obtained model has been solved by the method of multiple scales. Also, the obtained equation is solved using the numerical method and compared with the obtained answer with three Euler-Bernoulli linear models, the Von-karman nonlinear model and the reference nonlinear model [7].

4- Conclusions

The obtained results show that in vibration with a very small amplitude, the response of all models coincides with each other, but by increasing the amplitude of the vibration, the response of different models will be different. Also, by considering the length of the beam as an effective parameter, it can be seen that by reducing the length of the beam, the difference between the current model and the other investigated models increases. Therefore, the natural frequency for the beam with a smaller length is calculated using three models and with each other has been compared. The equation of the frequency response of forced vibrations and the frequency of the jump phenomenon for the desired beam is obtained and compared with the results of reference [7]. The results obtained according to Table 1 show that the use of the present model, in addition to calculating the natural frequency of the system less than the other two models, predicts the frequency of the resonance phenomenon less than the model [7] and practically the performance status of the system in the steady state in Gets a smaller range.

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