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AN OPTIMUM DESIGN OF A LEAF SPRING FROM COMPOSITE MATERIAL

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ABSTRACT

Leaf springs are commonly used in the suspension system of automobiles and are subjected to varying stress from dynamic loading leading to failure .An analytical methods using formulation and solution technique classical laminated plate theory (CLPT) for design optimization of composite leaf springs under different dynamic loading is presented here. Different ratio of high/ width (h/b), and duration of time effect are conclusions. A numerical method using FEM (Ansys11.0) have been for solving the dynamic stresses distribution in composite leaf spring. Both theoretical and numerical results gives good agreement and the studies reveal that the model proposition flexibility has significant influence on the system dynamic behavior and using likelihood with car industry automobile less than 500 kg. per wheel.

Keywords: Ansys, Composite Material, Layers, Leaf Spring, Orthotropic

التصميم المثالي لنابض ورقي من المواد المركبة

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الخلاصة

تستخدم النوابض الورقية عموما كعنصر اساسي في نضام تعليق السيارات , وهي تتعرض الى اجهادات متغيرة بسبب الاحمال الديناميكية المتغيرة والتي عادتا تودي الى الفشلتم اجراء التحليل النظري للنموذج واعتمد في الحل النظرية التقليدية للصفائح الطبقية(CLPT). تم حساب تاثير تغير نسبةالارتفاع الى العرض ,فترة تسليط الحمل و تغير مسافة المنطقة الوعرة. كذالك استخدم الحل العددي بطريقة العناصر المحدادة لايجاد توزيع الاجهادات الديناميكية في نابض ورقي مصنع من مواد مركبة. كلتا النتائج النظرية والعددية كانت متوافقة بشكل جيد, وكشفت الدراسة بأن نموذج المرونة المقترحة لها تأثير هام على سلوك النظام الديناميكي, وامكانية استخدامها للسيارت المصنعة لوزن اقل من 500 كيلو غرام لكل عجلة.

Nomenclature

$c_o; c_1; c_2$	Constant
D_{ij}	Bending stiffness's
E_{ii}	Young's modulus (N/m^2)
G_{ij}	Shear modulus (N/m^2)
h, b	Cross section (high, width) (mm)

I_{yy}	Moment of inertia (m^4)
k_{th}	Layer number
L	Length of leaf spring (mm)
M_{xx}; M_{yy}; M_{xy}	Moment about xx, yy, and xy plane $(N.m)$
Q_{ij}	Plane stress-reduced stiffness's
S	shear strength (N/m^2)
t	Time $(Sec.)$
ω_{ij}	Deflection of spring (mm)
X	Longitudinal strength (N/m^2)
σ_{ij}	Stresses (N/m^2)

INTRODUCTION

Recently, automotive industry requires higher level of design and calculation almost in every part in both fabrication and testing which can make it possible to improve and develop products. Leaf spring, one of automotive parts, was mainly design based on trial-and-error techniques and simplified equations using a 3-link mechanism and beam theory for stress calculation [Kamnerdtong-2005].

Spring are unlike other machine/structure are components in that they undergo significant deformation when loaded, their compliance enables them to store readily recoverable mechanical energy. It is well known that springs, in general, are designed to absorb and store energy and then release it. Hence, the strain energy of the material and the shape become a major factor in designing the springs. In a vehicle suspension, when the wheel meets on obstacle, the springing allows movement of the wheel meets on obstacle, the springing allows movement of the wheel over the obstacle and thereafter returns the wheel to its normal position (i.e., to be resilient).[Mahdi-2006]

There are different types of materials for metallic springs depending on the application. The materials used for such springs are principally, SAE-1080, 1095, 5155-60, 6150-60 and 9250-60. These are initially pre-stressed so as to increase the carrying capacity of the springs [Qureshi, 2001].

Hence, composite material becomes a very strong candidate for such applications. However, during the last two decades, and particularly in recent years, great effort has been made by the automotive industries in the application of leaf springs made from composite materials [AL-Qureshi, 2001]. However, due to the availability and cost limitation, the present work was restricted to the study of leaf springs made from fiberglass/epoxy (glass fiber reinforced plastic GFRP). It present advantages over graphite/epoxy such as lower sensitivity to cracks, impact and wear damage. In other words, fiberglass/ epoxy leaf springs are almost similar to metallic springs with regards to life requirements, since the have sufficient impact strength, and their mechanical properties are not greatly in flounced by the typical vehicle working conditions.[AL-Qureshi,2001]

Using genetic algorithms (GA) for design optimization of composite leaf springs by Rajendran et al. different methods are in use for design optimization, most of which use mathematical programming techniques, good agreement was obtained between these methods an applying the GA, the optimum dimensioned, which contributes towards achieving the minimum weight with adequate strength and stiffness also reduction of (75.6%) weight is achieved when a seven-leaf steel spring is replaced with a mono-leaf composite spring under identical conditions of design parameters and optimization.[Rajendran,2001].

Analytical loads in leaf spring bushing can be used to perform finite element analysis on brackets that connect the leaf spring to a truck from. Two models of leaf spring in MSC/NASTRAN and MDI/ADAMS were created to compare the bushing loads predicted by each model. The analysis simulated the standard jounce and roll test at the UMTRI (University of Michigan Transportation Research Institute) [Tavakkoli, 1992].

The purpose of this research is to study the effect of impact dynamic loading on the leading to failure using classical plate theory assumed the leaf spring as a curved simply supported strip plates with initial radius of curvature R and initial deflection δ .

These results have been supported by using Numerical solution of finite element method (Ansys 11.0) for determining the stress distribution and strain along the surface of composite leaf spring.

DESIGN CONCEPT

The flexural rigidity and angel of fibers are important parameters in the leaf spring design [Reddy, 2004]. This idea gives different type of design possibilities, number of layers, cross section, and angel of fibers i.e. the design of composite leaf spring aims at the replacement of six-leaf steel spring of an automobile with number of layered-leaf composite spring

i- General parameters

The general parameters designs requirements are taken to be identical to that of the steel of composite leaf spring are:

- Design load (static load),$P_o = 14715\ N\ (1500\ Kg.)$
- Maximum allowable vertical deflection see **Fig.1**, $\delta_{max.} = 160\ mm$
- Distance between eyes in straight condition see **Fig.1**, $L = 1220\ mm$
- Cross section see **Fig.1**,width($b = 20 \rightarrow 100$ mm) ;thickness ($h = 100$ mm)
- Number of layers are 24 (4.1997mm thickness of layer)

The mechanical properties of orthotropic E-glass/epoxy are [Swanson, 1997]:

$E_{11} = 38.6 GPa.;\ E_{22} = 8.27 GPa;\ G_{12} = 4.14 GPa; \nu_{12} = 0.26; Density\ \rho = 1.99 \frac{g}{cc}$

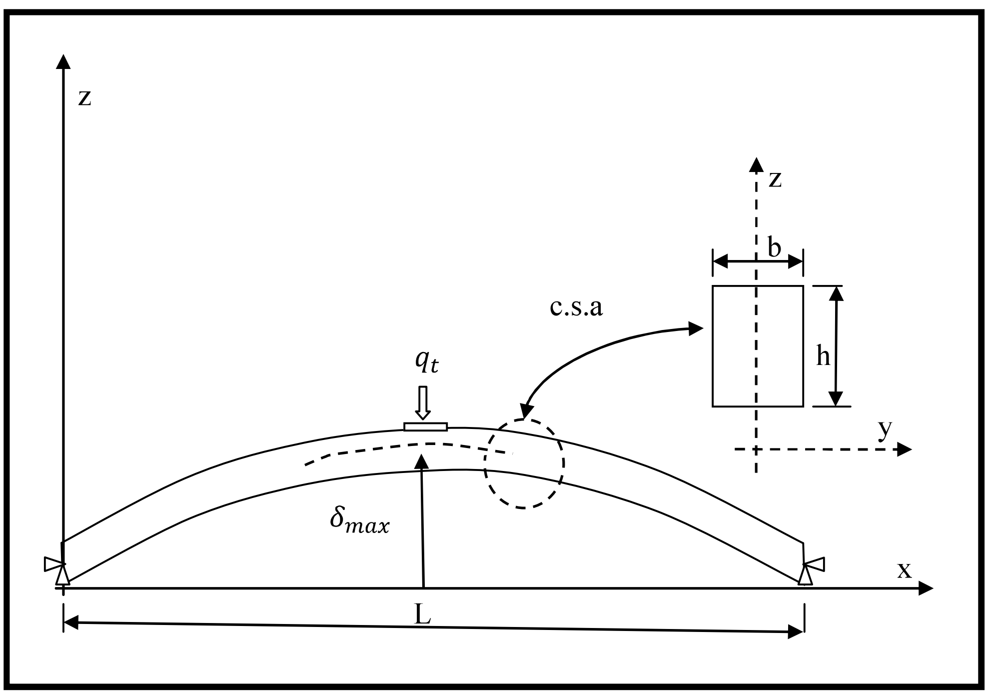

Figure 1: Dimension of leaf spring

ii- transient load

When the simple model of a motor vehicle that can vibrate in the vertical direction while traveling over a rough road with short time. The suspension vehicle subject to the transient load effect on the leaf spring with very short time describe by **Eq.1** [Mario Paz, 1980]:

$$q_t = P_o\left(1 - \frac{t}{t_o}\right)e^{-t/t_o} \tag{1}$$

Where

q_t Instantaneous toad
P_o Constant load (weigh of car per unit wheel) =14715 N(1500kg)
t Time (Sec.) $0 \le t \le t_o$
t_o Total time of applied load, **table (1)**

The deflection of simple leaf spring is symmetric about the point $x = l/2$ **Fig.1**. The expiration for the bending moment **Eq.2** [Reddy, 2004]

$$M_x = \frac{q_t\, b\, x}{2} \quad for \quad 0 \le x \le \frac{l}{2} \quad where\ b\ \ per\ unit\ width$$

Or

$$M_x = \frac{P_o\left(1-\frac{t}{t_o}\right)e^{(-t/t_o)}\, b\, x}{2} \tag{2}$$

The velocities of car are assumed are variably from 54 km/h to 198 km/h, driving quick on declining rough road. The parameters are arraignment in **Table 1.**

Table 1

rough road destine		5 m	10 m	15 m
Velocity km/h	Velocity m/s	t_o Sec.	t_o Sec.	t_o Sec.
54	15	2.6e-1	5.33e-1	8.11e-1
90	25	1.6e-1	3.21e-1	4.81e-1
126	35	1.14e-1	2.28e-1	3.42e-1
162	45	8.88e-2	1.77e-1	2.66e-1
198	55	7.27e-2	1.45e-1	2.18e-1

iii- General theory

There are two cases of laminated plates that can be treated as one-dimensional problems, i.e. the displacements are functions of just one coordinate:

1- laminated beams
2- Cylindrical bending of laminated plate strips.

When the width b (length along the $y-axis$) of a laminated plate is very small compared to the length along the ($x-axis$) and the lamination scheme. And loading is such that the displacements are functions of x only; the laminate is treated as a beam **Fig.1**. The cylindrical bending problem is a plane strain problem. Whereas the beam problem is a plane stress problem. [Reddy, 2004]

For a single especially orthotropic layer, the stiffness can be expressed in terms of the Q_{ij} and thickness h **Eq.3**

$$Q_{11} = \frac{E_1}{1-\nu_{12}\nu_{21}};\ Q_{12} = \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}};\ Q_{22} = \frac{E_2}{1-\nu_{12}\nu_{21}};\ Q_{66} = G_{12} \tag{3}$$

When a laminates of multiple especially orthotropic layers that are symmetrically disposed, both from a material and geometric properties standpoint, about the midplane of the laminate does not exhibit coupling between bending and extension. The in-plan stress in the K_{th} layer can be computed from the **Eq.1**

$$\begin{Bmatrix} M_{xx} \\ M_{yy} \\ M_{xy} \end{Bmatrix} = -\begin{bmatrix} D_{11} & D_{12} & D_{16} \\ D_{12} & D_{22} & D_{26} \\ D_{16} & D_{26} & D_{66} \end{bmatrix} \begin{Bmatrix} \partial^2\omega/\partial x^2 \\ \partial^2\omega/\partial y^2 \\ 2\partial^2\omega/\partial x\partial y \end{Bmatrix} \tag{4}$$

Where

$$D_{11} = \frac{Q_{11}h^3}{12};\ D_{12} = \frac{Q_{12}h^3}{12};\ D_{22} = \frac{Q_{22}h^3}{12};\ D_{66} = \frac{Q_{66}h^3}{12} \tag{5}$$

And

$D_{16} = D_{26} = 0 \quad for\ orthotropic\ material\ and\ symmetry\ [Reddy, 2004]$

GOVERNING EQUATION

Consider the bending of symmetrically laminated beams according to CLPT. For symmetric laminates, the equation for bending deflection is uncoupled from those of the stretching displacements. If the in-plane forces are zero. The in-plane displacement $(u_o, v_o)$are zero, and the problem is reduced to one of the solving for bending deflection and stresses in deriving the laminated beam theory we assume that[Reddy, 2004]

$$M_{yy} = M_{xy} = 0 \tag{6}$$

Everywhere in the beam the CLPT constitutive equations for symmetric laminates. From **Eq. 4 ,** in inverse form. We have

$$\begin{Bmatrix} \partial^2\omega/\partial x^2 \\ \partial^2\omega/\partial y^2 \\ 2\partial^2\omega/\partial x\partial y \end{Bmatrix} = -\begin{bmatrix} D_{11}^* & D_{12}^* & D_{16}^* \\ D_{12}^* & D_{22}^* & D_{26}^* \\ D_{16}^* & D_{26}^* & D_{66}^* \end{bmatrix} \begin{Bmatrix} M_{xx} \\ M_{yy} \\ M_{xy} \end{Bmatrix} \tag{7}$$

Where

$$D_{11}^* = (D_{22}\ D_{66})\ D^*$$
$$D_{12}^* = (-\ D_{12}\, D_{66})\ D^*$$
$$D_{22}^* = (D_{11}\ D_{66})\ D^*$$
$$D_{66}^* = (D_{11}\ D_{22} -\ D_{12}\, D_{12})/D^*$$
$$D^* = (D_{11}\ D_1\ - D_{12}D_2)$$
$$D_1 = (D_{22}\ D_{66})\ ; D_2 = (-\ D_{12}\, D_{66}\ ) ; D_3 = (D_{12}\ D_{26}\ )$$

Also the stress component per layer in matrix form is [Reddy, 2004]

$$\begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \tau_{xy} \end{Bmatrix}^k = \begin{bmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} \\ \bar{Q}_{12} & \bar{Q}_{16} & \bar{Q}_{26} \\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} \end{bmatrix}^k \begin{Bmatrix} -\partial^2\omega/\partial x^2 \\ -\partial^2\omega/\partial y^2 \\ -2\partial^2\omega/\partial x\partial y \end{Bmatrix} \quad (8)$$

And

$$\bar{Q}_{11} = Q_{11} \quad cos^4\theta + 2(Q_{12} + 2Q_{66}) \quad sin^2\theta \quad cos^2\theta + Q_{22} \quad sin^4\theta$$
$$\bar{Q}_{12} = (Q_{11} + Q_{22} - 4\,Q_{66}) \quad sin^2\theta \quad cos^2\theta + Q_{12}(\quad sin^4 + \quad cos^4\theta)$$
$$\bar{Q}_{16} = (Q_{11} - Q_{12} - 2\,Q_{66})\ sin\,\theta \quad cos^3\theta + (Q_{12} - Q_{22} + 2Q_{66}) \quad sin^3\theta\, cos\,\theta$$

For orthotropic material and symmetry given $D_{16} = D_{26} = 0$, with **Eq.6** gave

$$\sigma_{xx}{}^k_{(x,z,t)} = \frac{q_o\, x\, z}{2}\left(\bar{Q}^k_{11}\, D^*_{11} + \bar{Q}^k_{12} D^*_{12} + \bar{Q}^k_{16} D^*_{16}\right) \quad (9)$$

Now from the assumption **Eq.6**, we have

$$\partial^2\omega/\partial x^2 = -D^*_{11} M_{xx}$$

$$\partial^2\omega/\partial y^2 = -D^*_{12} M_{xx} \quad (10)$$

$$2\partial^2\omega/\partial x\partial y = -D^*_{16} M_{xx}$$

Eq. 10 indicates that the deflection we cannot be independent of the coordinate y due to the position effect D^*_{12} and anisotropic shear couplingD^*_{16}. These effects can be neglected only for lon as a function of coordinate (x) and time.

$$\omega = \omega(x,t) \quad (11)$$

From used in the classical Euler Bernoulli beam theory, we introduce the quantities

$$M = bM_{xx}\ , E^b_{xx} = \frac{12}{h^3 D^*_{11}} = \frac{b}{I_{yy} D^*_{11}}\ ,\ I_{yy} = \frac{bh^3}{12} \quad (12)$$

And write **Eq. 11**, as

$$\partial^2\omega/\partial x^2 = -D^*_{11} M_{xx} \quad or \quad M_x = -E^b_{xx}\, I_{yy} \frac{\partial^2\omega}{\partial x^2} \quad (13)$$

Where b is the width and h is the total thickness of the laminate.

ANALYSES

The deflection of curve simply support beam a center point load is symmetric about the point $x = l/2$, the expression for the bending moment is :

$$M_{(x,t)} = \frac{q_{(t)}\, bx}{2} \quad for\ 0 \le x \le a/2 \quad (14)$$

Sup**. Eqs.1 and 13** into **Eq. 14** we give

$$E_{xx}^{b}\, I_{yy}\, \frac{\partial^3 \omega}{\partial x^2 \partial t} = -P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, b\, \left(\frac{x}{2}\right) \tag{15}$$

The general solution is:

$$E_{xx}^{b} I_{yy}\, \omega_{(x,t)} = -\int_0^{t_o/2} \left\{ \int_0^x \left(\int_0^\xi P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, b\, \left(\frac{\zeta}{2}\right) d\zeta \right) d\xi \right\} dx + c_o \frac{t^2}{2} + c_1 t + c_2$$

$$E_{xx}^{b} I_{yy}\, \omega_{(x,t)} = -\int_0^{t_o/2} P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, b\, \frac{1}{12} x^3\, dx + c_o \frac{x^2}{2} + c_1 x + c_2$$

$$E_{xx}^{b} I_{yy}\, \omega_{(x,t)} = -P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, b\, \frac{1}{48} x^4 \Big]_0^{t_o/2} + c_o \frac{x^2}{2} + c_1 x + c_2$$

$$E_{xx}^{b} I_{yy}\, \omega_{(x,t)} = -P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, b\, \frac{1}{48} \frac{t_o^4}{16} + c_o \frac{x^2}{2} + c_1 x + c_2$$

$$E_{xx}^{b} I_{yy}\, \omega_{(x,t)} = -\frac{P_o\, b}{768}\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, t_o^4 + c_o \frac{x^2}{2} + c_1 x + c_2$$

Using boundary condition to determined the constant $c_0, c_1, and\ c_2$

B.C.

$\omega_{(x,t)} = 0 \;\; when\;\; x = 0 : \;\; c_2 = 0$

$$\omega_{(x,t)} = 0 \;\; when\;\; x = l \;\; and\;\; \frac{d\omega}{dx} = 0 \;\; when\; x = \frac{l}{2}$$

$$\therefore\; c_o = -\frac{2}{(2xl - l^2)} \frac{b}{768} P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, t_o^4$$

And

$$\therefore\; c_1 = x\left[\frac{2}{(2xl - l^2)} \frac{b}{768} P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, t_o^4\right]$$

Now the total deflection and stress for composite material leaf spring under dynamic loading is Eqs. 16 and 17.

$$\omega_{(x,t)} = \frac{1}{E_{xx}^{b} I_{yy}}\left[-\frac{b}{768} P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, t_o^4 + \frac{x^2}{2}\left[-\frac{2}{(2xl-l^2)}\frac{b}{768} P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, t_o^4\right] + x^2\left[\frac{2}{(2xl-l^2)}\frac{b}{768} P_o\left(1-\frac{t}{t_o}\right) e^{-t/t_o}\, t_o^4\right]\right] \tag{16}$$

$$\sigma_{xx}{}^{k}_{(x,z,t)} = \frac{P_o\left(1-\frac{t}{t_o}\right)e^{-t/t_o}\, x\, z}{2}\Bigg(\Bigg[\frac{E_1}{1-\nu_{12}\nu_{21}}\cos^4\theta + 2\left(\frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}} + 2G_{12}\right)\ \sin^2\theta\ \ \cos^2\theta + \frac{E_2}{1-\nu_{12}\nu_{21}}\ \ \sin^4\theta\Bigg]^k \Bigg[\left(\frac{E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\frac{G_{12}\,h^3}{12}\right)\left(\frac{E_1}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\ \left(\frac{E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\ \frac{G_{12}h^3}{12}\right) - \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\left(-\ \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\ \frac{G_{12}h^3}{12}\right)\right)\Bigg] + \Bigg[\left(\frac{E_1}{1-\nu_{12}\nu_{21}} + \frac{E_2}{1-\nu_{12}\nu_{21}} - 4\,G_{12}\right)\ \sin^2\theta\ \ \cos^2\theta + \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}(\ \sin^4 +\ \ \cos^4\theta)\Bigg]^k \Bigg[\left(-\ \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}\ \frac{G_{12}h^3}{12}\right)\left(\frac{E_1}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\ \left(\frac{E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\ \frac{G_{12}h^3}{12}\right) - \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\left(-\ \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}\frac{h^3}{12}\ \frac{G_{12}h^3}{12}\right)\right)\Bigg]\Bigg) \quad (17)$$

Fig. 2a-b, represents the deflection and stress test for different angle. The code-19 is given perfect distributing of layers **Fig.4.**

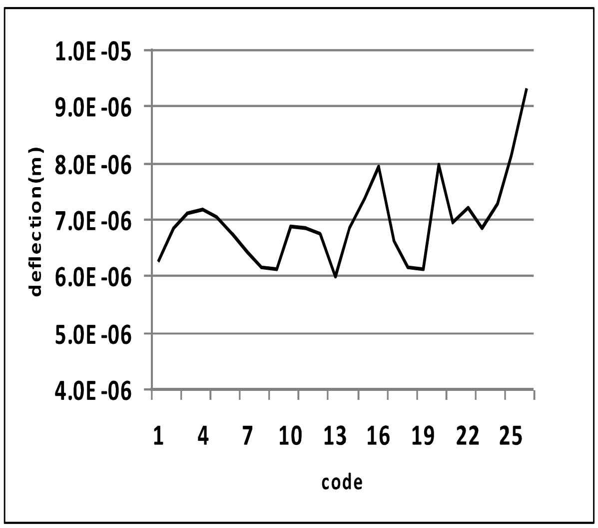

Fig. 2a: deflection-code relation FEA

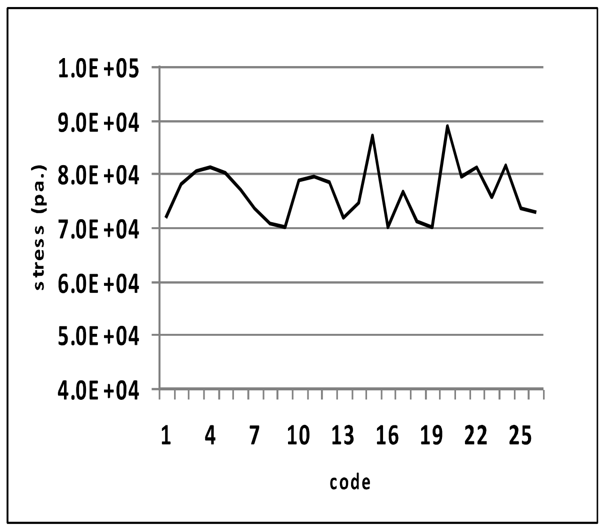

Fig.2b: stress –code relation FEA

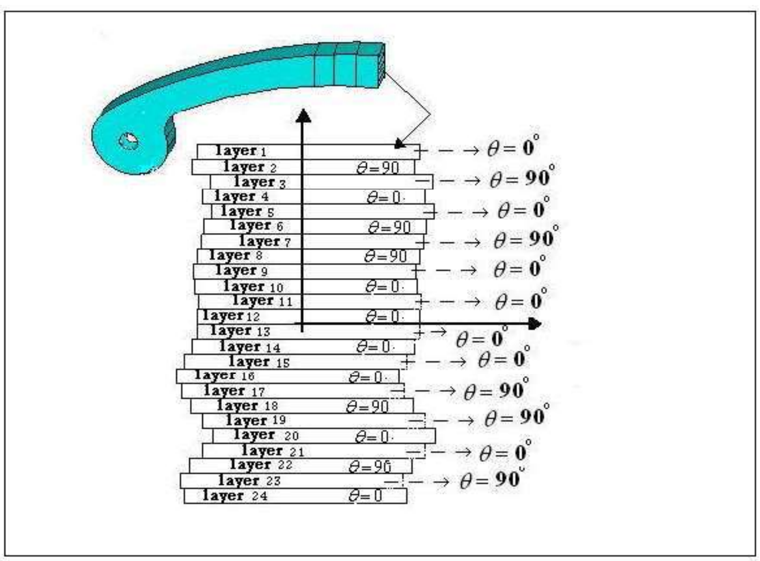

Figure 3: C.S.A. of leaf spring (angle of layer)

In addition, analytical analysis can be used to develop an expression which is a function of thickness and width along the spring. This could be manipulated by introducing the Tsai-Hill failure criterion of composite which yields the following equation [AL-Qureshi, 2001]:

$$\left[\frac{3\, q_t\, l}{2\, b\, h^2\, X}\right]^2 + \left[\frac{q_t}{b\, h\, S}\right]^2 = 0.3 \tag{18}$$

Where q_t is the applied load per wheel, $(l/2)$ the position along the horizontal axis, b, the width, h, the thickness , $(X \cdot and \cdot S)$ are the average longitudinal and shear strengths of laminate, respectively. Then the boundary conditions can be included, such as the design load carrying capacity which is taken from Eq.1, factor of safety equals 3, the mechanical properties of the glassfiber/epoxy layer $(X = 640MPa., S = 31MPa.)$ [AL-Qureshi, 2001]

LEAF SPRING MODEL IN ANSYS-11

A model of composite leaf spring **Fig.3** was created in Ansys-11.0. Each element of the spring is made of general purpose SOLID46 **Fig.4** element with constant rectangular cross section. The elements were created along the front end of the leaf spring. The general purpose SOLID46 element of Ansys-11.0 has a total of 24 degrees of freedom and can be used to build suspension mechanism. The spring is represented by simply support in the FE model. SOLID46 is a layered version of the 8-node structural solid SOLID46 designed to model layered thick shells or solids. The element allows up to 250 different material layers. If more than 250 layers are required, a user-input constitutive matrix option is available. The element may also be stacked as an alternative approach. The element has three degrees of freedom at each node: translations in the nodal x, y, and z directions.[Ansys11.0, 2007]

The summaries properties of FE model (SOLID46) are:

- Number the degree of freedom per node =3 (x,y,z)
- Number the node per element =8 node
- Number the total element in model=50
- Number of the layers =24 **(Fig.6a,b,c)**

Figure 4: leaf spring model

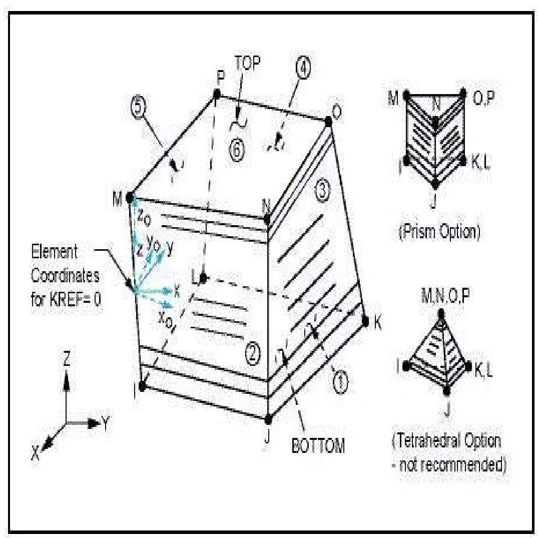

Figure 5: Solid46 Element

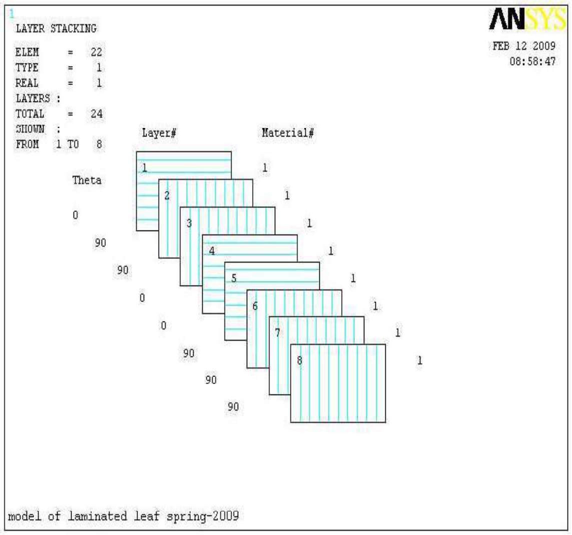

Figure 6a: Constriction layer-8

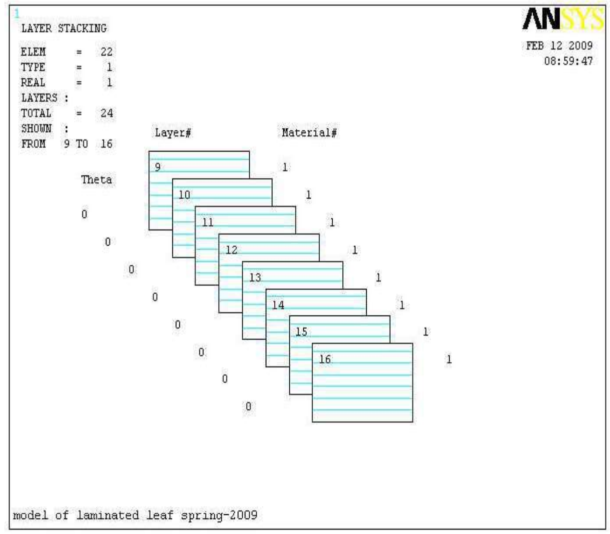

Figure 6b: Constriction layer 9-16

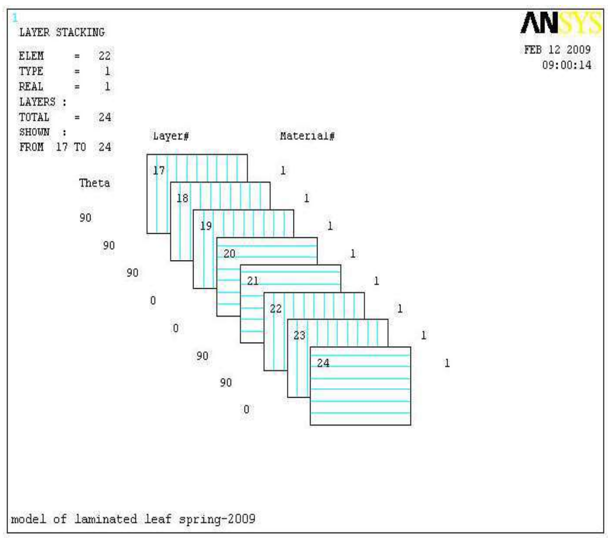

Figure 6c: Constriction layer 17-24

RESULTS AND DISCUSSION

The procedure described in the previous section has been applied to the design of minimum weight of composite leaf spring to replace the six-leaf steel spring in the rear suspension system of a passenger car. The design parameters such as distance between spring eyes, comber and load are kept as same in both steel and pomposity leaf springs. The input parameters used in this work are listed in **Table 1**; the geometric models of steel and pomposity leaf spring considered for optimization are shown in **Fig. 1**.

Since the in plane stresses due to variable velocity from low to medium speed of car are known, they are resolved into components of stress$(\sigma_x, \sigma_y, \sigma_{xy})$. The perpendicular component σ_y is approximate eσ_x causing an increase in the beam at the midpoint. **Fig. (7)** and **Fig. (8)** shows the stress propagation in first layer only in the center of beam and stress distributed that happen through the thickness at mid point, for composite leaf spring obtained analytically and numerically by Ansys11.0.

Fig. 9 and 10, show the normal stress accurse in layer one (for example), when the car trivial between the rang 15 m/s to 55 m/s, over 5m rough road and duration time effect.**Fig.11**, show the Tsai-Hill failure criterion of composite materiel under deferent loading, not the failure point(intersection point) accruing when the loading over 61 Kn.

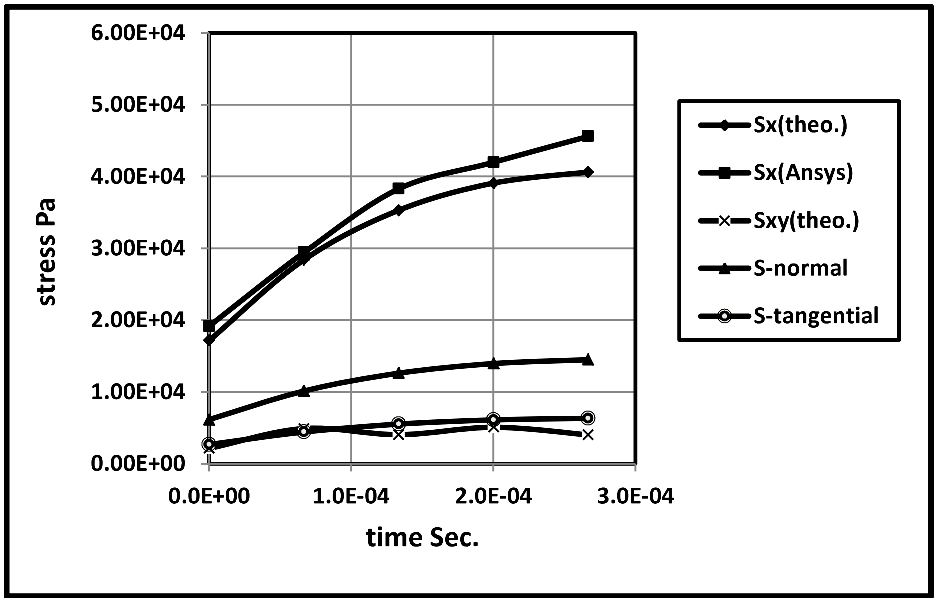

Figure 7: stress in layer 1

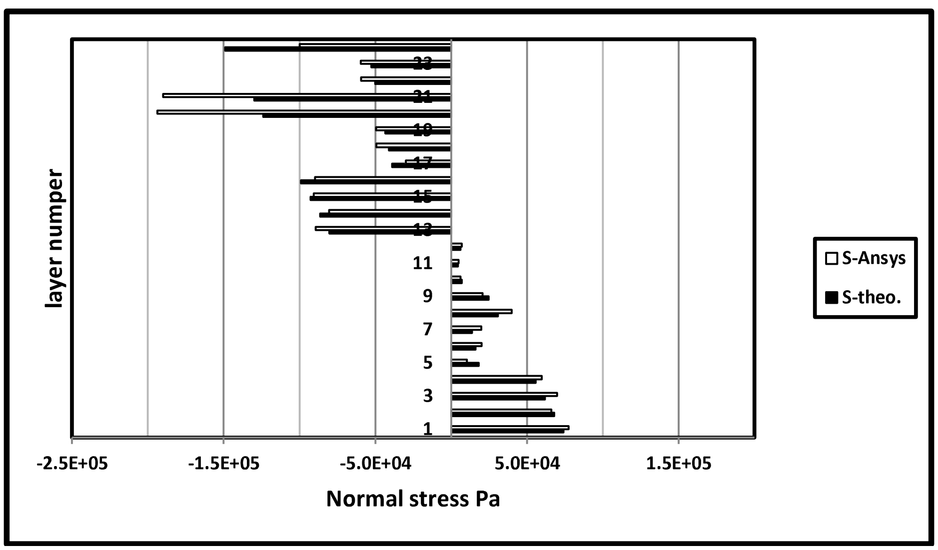

Figure 8: stress distribution through thickness of beam

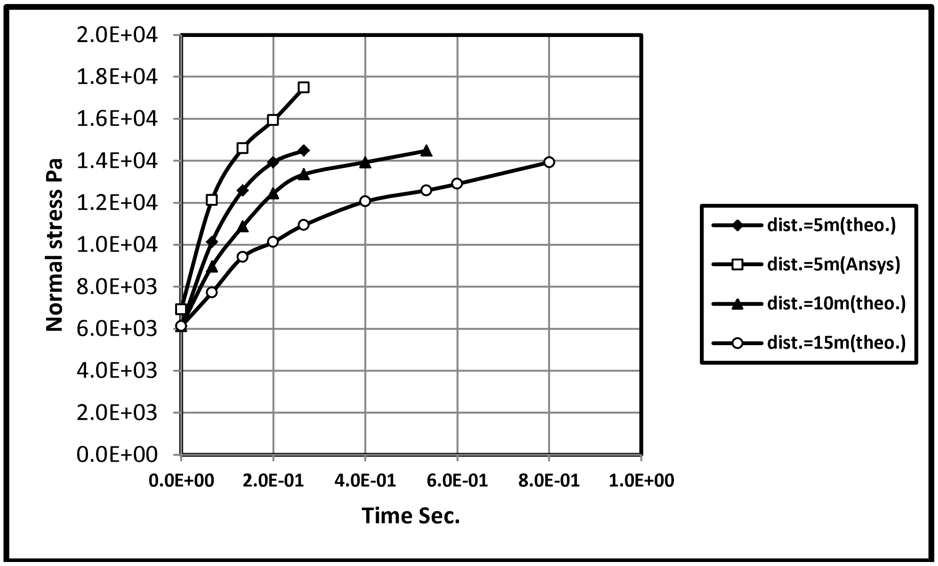

Figure 9: velocity of car 15 m/s, (layer 1)

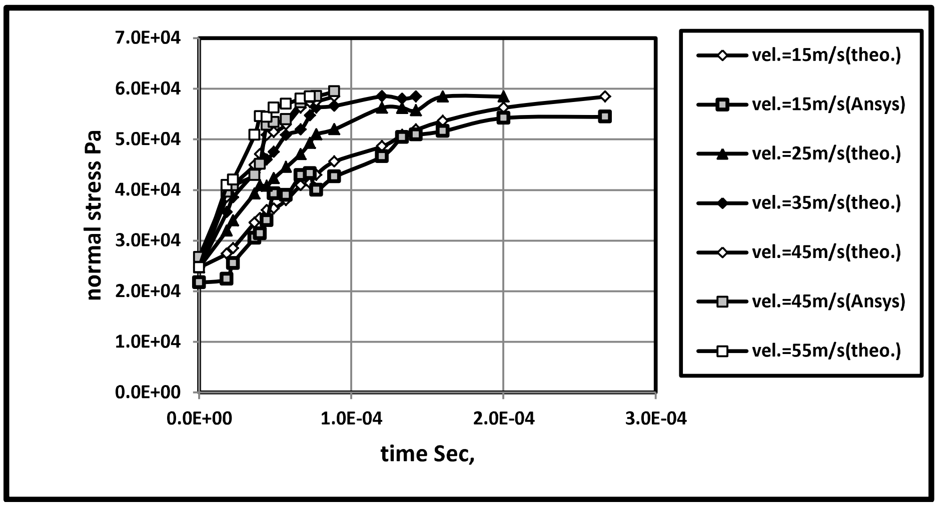

Figure 10: rough road distend is 5m,(layer 4)

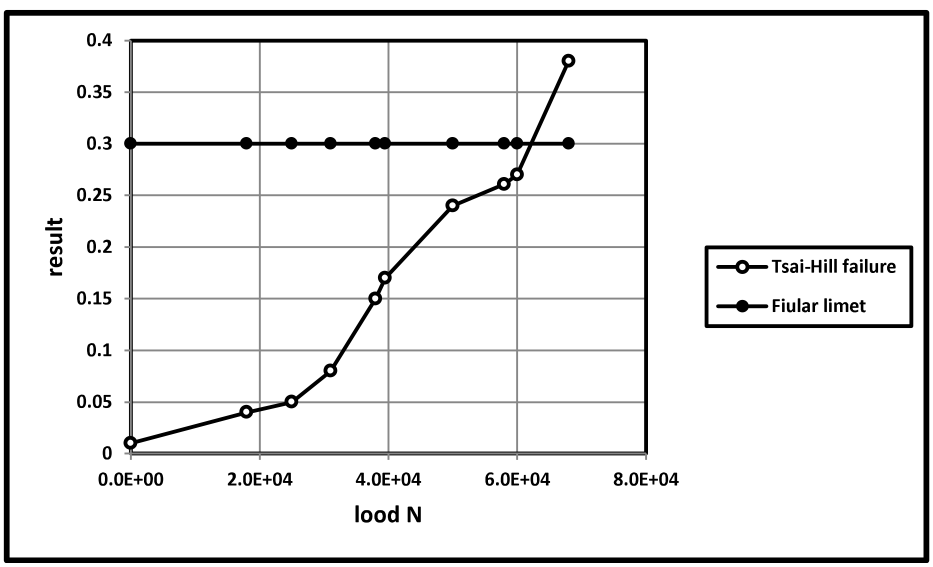

Figure 11: Tsai-Hill failure criterion

CONCLUSIONS

The major observation and conclusion from study dynamic analysis, simply supported composite materiel of leaf spring, under various velocities and rough road distance are listed as follow:

1. The perfect distributed of angles in layers there are symmetric angle-ply laminate (0,90,90,0,0,90,90,90,0,0,0,0). These results given the minimum stress and minimum deflection for car not increase weight over 500 Kg. per wheel with the safety factor is 3.

2. The decrees in total Wight of suspensions system are (77.436%), i.e. the composite leaf spring weight is 11.4286 Kg. but 50.651 Kg. for six leaf spring stainless steel.

3. The duration time of dynamic load is decreasing when the velocity of car is increasing, the normal stress are increases.

4. The normal stress in leaf spring are effect when the rough road as smallest with constant velocity of car.

5. When the velocity of car variables from 54 km/h to 198 km/h by constant interval the rough road the stress is increasing.

6. The compression of analytical results with numerical result Ansys11.0 indicated that the assumption of linear of multi-leaf spring behavior chance possibility.

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