International Journal of Scientific & Engineering Research, Volume 5, Issue 7, July-2014
ISSN 2229-5518

Experimental and Numerical study of influence the loading mode on Fatigue Life in Notched Steel Beam

Qasim Bader and Emad K. Njim

Abstract: Fatigue life experiments were carried out on plain cylindrical specimens made of low carbon steel alloy AISI 1020 at room temperature, on a cantilever rotating-bending machine .Test results are obtained for constant amplitude load in fully reversed bending with mean stress equal zero with various notch geometries and dimensions. By using Stress Life approach, fatigue life analysis are obtained experimentally. Simulation by FEA method for smooth (reference) and notched specimens have been done based on three types of loading mode (Reversed bending ,Reversed Axial and Reversed Torsional). The results show that the life of the specimen tested based on reversed bending is greater than those subjected to Axial and Torsional loading mode respectively and there is acceptable error between experimental and numerical works.

Keywords: Notched Specimens, Stress life approach, S-N curve ,Loading mode, ,FEA

1.INTRODUCTION

Many machine members are subjected to repeated, reversed, or fluctuating cyclic stresses during their operation .If the number of cycles of stressing is very large, it has been observed that the failure of the member often occurs with a sudden rapture and without any pronounced amount of deformation. This phenomenon which occurs under cyclic stressing is termed fatigue failure [1]. Fatigue is a process in which damage accumulates due to the repetitive application of loads that may be well below the yield point [2]. The three major fatigue life methods used in design and analysis are the stress-life method, the strain-life method, and the linear-elastic fracture mechanics method. These methods attempt to predict the life in number of cycles to failure, N, for a specific level of loading. Life of $1 \le N \le 10^3$ cycles is generally classified as low-cycle fatigue, whereas high-cycle fatigue is considered to be $N > 10^3$ cycles. The stress-life method, based on stress levels only, is the least accurate approach, especially for low-cycle applications. However, it is the most traditional method, since it is the easiest to implement for a wide range of design applications, has ample supporting data, and represents high-cycle applications adequately [3]. Reference [4], described notch effects in fatigue and fracture and explained that the notch effect in fracture is characterized by the fact the critical gross stress of a notched structure is less that the critical net stress and the notch effect in fracture is sensitive to structure geometry. The presence of shoulders, grooves, holes, keyways, threads, and so on, results in modifications of the simple stress distributions of so that localized high stresses occur ,this localization of high stress is known as stress concentration ,measured by the stress concentration factor. Stress concentration will be produced in these notches as a result of external force and depend on geometry.

Nasim Daemi & Gholam Majzoobi [5], developed experimental and theoretical life on notched specimens under bending, fatigue life of notched specimens with various notch geometries and dimensions was investigated by experiment and Manson- Caffin analytical method.

A. Fatemi et al. [6] studied Fatigue behavior in circumferentially notched round specimens made of a vanadium-based micro-alloyed forging steel was under constant amplitude axial and torsion loads under axial and torsion loads .

Presentation of notches in structural components causes stress intensification in the vicinity of the notch tip [7]. Fillets are commonly used in mechanical parts to provide smooth transition in regions where there is a sudden change in cross-section as in the case of shoulders hence, they are usually the most critical regions in mechanical parts especially under fatigue loading, considering that an increase in the maximum stress level considerably shortens the fatigue life of a part. Shoulders are introduced in bars for various purposes, to provide bearing support, cam profile, etc. However, this leads to an increase in local stress levels. Stress concentration factor Kt , which is the ratio of the maximum stress developed in this region, σ_{max}, to nominal stress σ_{nom} [8].

Reference [9] shows Fatigue analysis by FEA method of a simple shaft under combined Bending and Torsion. Short cracks at notches and fatigue life prediction under axial and mode Torsion loadings have been studied in R [10]. Optimization of shaft design under forces and the torques acting on the shaft in drive system and frequent failure of a shaft employed in a spinning machine is studied R [11].

The aim of this paper is study experimentally fatigue life behavior obtained based on the effect of V notch shape on cylindrical specimens with different dimensions under three loading modes (Reversed bending ,Axial and Torsion loading) ,then results compared with finite element analyses. Finally, a summary section provides the conclusions drawn based on the results obtained.

2- Experimental Work:

The experimental work included assessment of fatigue life specifications by using stress life approach for low carbon steel alloy supplied from the local market with and without notches of various geometries , and effect of stress loading mode on Fatigue life . A Typical applications of this alloy include steel sheets for pressing out such as components of automobile body, structural shapes (I-beams, channel and angle iron), and sheets that are used in pipelines, rivets, nails and others. The experimental procedure consist of three parts, the first one deals with the selection of materials used and the specimens preparation, the second part deals with different mechanical tests, the third includes details of the fatigue test .

2.1 Material Selection:

In this work, low carbon steel alloy treated commercially, had been used in this investigation, this type of alloy has a wide application in industry. The chemical composition test was done by use Spectrometer instrument type (ARC. MET 8000), the results was within the specification limits [12], and as shown in table 1.

Table 1:Chemical composition of tested materials (mass%)

C	Si	Mn	P	S%	Fe
0.208	0.27	0.603	0.012	0.021	Bal.

2.2 Mechanical Tests:

The tensile test is a standard test which was conducted using the microcomputer controlled electronic universal testing machine .Average value of four readings for each test have been taken to satisfy an additional accuracy. The hardness of a material is a measure of its resistance to penetration by an indenter. Hardness is also a measure of strength and often has the units of stress.

There are two hardness tests have been done in this investigation Brinell's and Vicker's Hardness test. The average value of four readings was recorded, the results of basic mechanical properties for low carbon steel alloy are given in table (2). ; For more details about the Tensile and Hardness tests procedure see [13].

Table 2 : Mechanical Properties of the AISI 1020

Property	Value
σ_u (MPa)	470
σ_y (MPa	350
Elongation [%]	26
Modula's of Elasticity (Gpa)	202
Brinell Hardness (HB)	135
Vickers Hardness (HV)	142

2.3 Fatigue Test :

In the revolving fatigue testing machine, a rotating sample which is clamped on one side is loaded with a concentrated force. The load is applied at one end of the sample and with the help of a motor, rotation about its own axis is achieved. Due to this rotation , a load reversal condition is achieved at two opposite sides on the circumference of the specimen. A triangular bending moment is developed in the specimen. Following a certain number of load cycles, the sample will rupture as a result of material fatigue A cantilever bending fixture was designed to test the steel specimens based on the critical (i.e. failure) location. Cantilever bending was used in order to minimize the magnitude of the applied loads necessary to achieve the desired nominal stresses. Stresses at which the material fails below the load cycle limit are termed fatigue limit . S-N curves are plotted by using software of Fatigue instrument presented in PC which is connected directly to fatigue machine, for more details about the fatigue test specimens geometry and procedure , see [13].

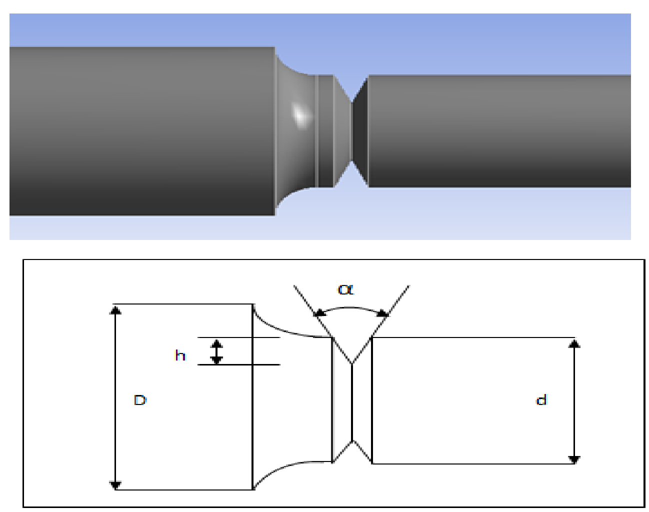


Fig. 1: Schematic dia. of Specimen No.1 at max. Bending region

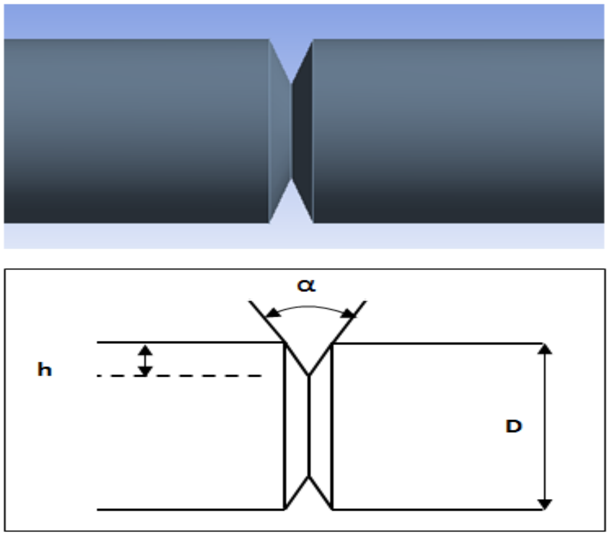


Fig. 2: Schematic dia. of Specimens No.2 at central part

To calculate Stress amplitude for each load case ,below relation are used.

$$\sigma_a = \frac{32\ M_b}{\pi d^3} = \frac{32\ F.a}{\pi d^3} \quad \text{For reversed bending}$$

$$\sigma_a = \frac{F}{\pi r^2} \quad \text{For axial loading}$$

$$\tau_a = \frac{2T}{\pi r^3} \quad \text{For torsional loading}$$

Table 3: Dimensions of V notch shape specimens

Spec. no.	D (mm)	d (mm)	h (mm)	h/D
1	12	8	0.5	0.0416
2	8	-	2	0.25

3- Stress Life model

The stress-life curve is a graphical representation of fatigue data. It represents the relationship between fatigue life, in cycles, and the applied stress amplitude .

Basquin's relation the most commonly used model and provides an analytical expression of the S-N curve, for finite life (low or high cycle fatigue). By use this technique an estimation of life prediction, with little information on the material, can be obtained , see [14].

The simple Basquin's curve is represented by :

$\sigma_a = a\ N_f^{\ b}$ or

$\tau_a = a\ N_f^{\ b}$

Where :

σ_a and τ_a is the fatigue stress and shear stress amplitudes (MPa), respectively

N_f is the number of cycles to failure (MPa),

The parameters a and b are both constant, depending on the material and on the geometry, respectively. The coefficient a is approximately equal to the tensile strength. The coefficient b is the fatigue strength exponent. These coefficients can be evaluated by use least square method (linearizing the power law in logarithmic form) ,it is important to mention that the S-N curve is represented in the log-log scale .

The value of Fatigue limit is not clearly obvious on the S-N curve; therefore, the Fatigue limit can be calculated by using the fatigue life estimation equation at 10^6 cycles.

4- CORRELATION OF STRESS-LIFE DATA

For a set of amplitude-mean-life data, it is useful to employ Least Square Method where in many branches of applied mathematics and engineering sciences we come across experiments and problems, which involve two variables. For example, it is known that the Stress amplitude σ_a of a steel specimens in S-N curve varies with the cycles of failure N according to the Basquin's formula $\sigma_a = aN_f^{\ b}$ Here a and b are the constants to be determined. For this purpose we take several sets of readings of stress amplitude and the corresponding Cycles. The problem is to find the best values for a and b using the observed values of σ_a and N, thus, the general problem is to find a suitable relation or law that may exist between the variables x and y from a given set of observed values,(xi , yi), i = 1, 2,.....,n . Such a relation connecting x and y is known as empirical law. For above example, x = σ_a and y = N.

The process of finding the equation of the curve of best fit, which may be most suitable for predicting the unknown values, is known as curve fitting. Therefore, curve fitting means an exact relationship between two variables by algebraic equations. There are following methods for fitting a curve. The graphical method has the drawback in that the straight line drawn may not be unique but principle of least squares provides a unique set of values to the constants and hence suggests a curve of best fit to the given data. The method of least square is probably the most systematic procedure to fit a unique curve through the given data points [15].

5- Numerical Investigation:

ANSYS program was used to perform fatigue analysis where a cylindrical notched fatigue specimens with angle 45 degree and one two depths 0.5 and 2 mm are modeled and FE simulated results are generated for fatigue three loading modes (Bending, Axial and Torsion) at different stress amplitude by using ANSYS program Version 11. Solid hexahedral elements (solid187), with 20 nodes were considered. The mechanical properties of the selected material and Stress life data obtained by experiments on reversed bending machine have been employed in building model .The element meshes were generated, boundary condition for each load type corresponding to maximum loading condition was given . Through ANSYS results that are common to stress life approach of fatigue analyses are include :Fatigue life ,Fatigue damage at a specified design life ,Fatigue factor of safety at a specified design life Stress biaxiality and Fatigue sensitivity chart.

To calculate stress concentration factor by FEM ;

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} \quad \textit{for normal stress (bending or tension)}$$

$$K_{ts} = \frac{\tau_{max}}{\tau_{nom}} \quad \textit{for shear stress (torsion)}$$

where the stresses σ_{max}, τ_{max} represent the maximum stresses to be expected in the member under the actual loads and the nominal stresses σ_{nom}, τ_{nom} are reference normal and shear stresses. The subscript *t* indicates that the stress concentration factor is a theoretical factor. Knowledge of the stress gradient provides a means of determining the nominal stress (σ_{nom}), for example we can calculate K_t for cylindrical shaft with fillet under bending load, by using the FEA ,

σ_{Max} (586.39 Mpa) at the beginning of the fillet and

$\sigma_{nom} = \sigma_{alternating} = (M.y/I)$

$\sigma_{nom} = \frac{32\,M}{\pi d^3}$

$K_t = (\sigma_{Max} / \sigma_{nom}) = (586.39/400)$

$K_t = 1.4659$

By using analytical method the K_t using the analytical formula from (Roark) is:

$K_t = 1.40767$

% error = {(K_t(formula) –K_t(FEA)) / K_t (formula)} * 100

= {(1.40767 –1.4659) / 1.40767} * 100 = - 4.136%

To calculate fatigue strength factor ,

$$K_f = 1 + (K_t - 1)\,q$$

where q is the notch sensitivity it can be defined from the Kunn-Hardarth formula in terms of Neuber's constant (a) and the notch radius (r).

$$q = \frac{1}{1 + \frac{a}{r}}$$

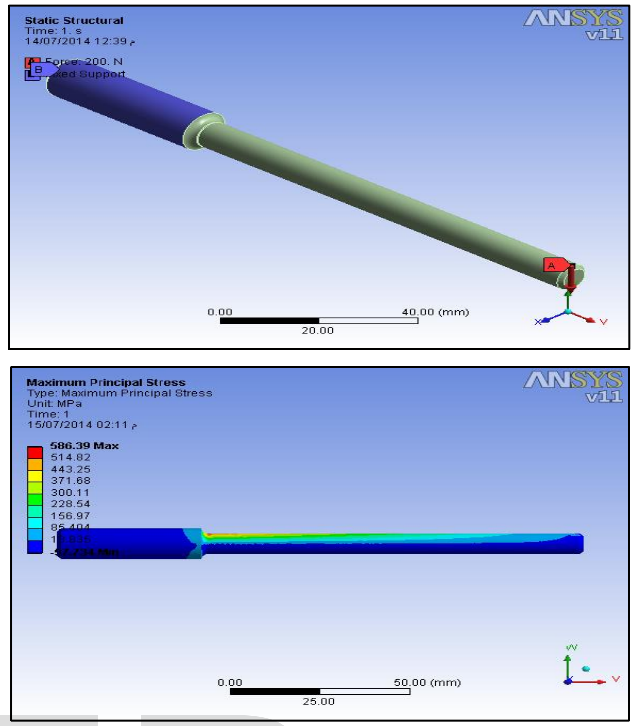


Fig. 3: Fatigue analysis for the model under Reversed Bending Load

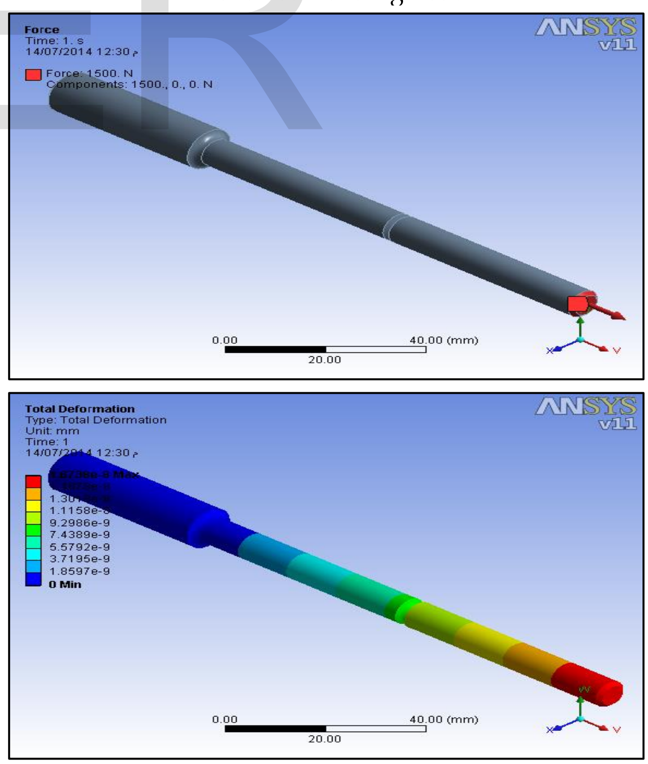


Fig. 4: Fatigue analysis for the model under Tension Load

4- RESULTS

In this work experimental fatigue lives for circumferentially notched round specimens with various geometries subjected to completely reversed bending loading have been investigated . By use FEA , fatigue analysis based on stress life approach have been found for cylindrical specimens subjected to completely reversed bending ,axial and torsion and according to the accurate stress concentration factors corresponding to different amplitude and shear stresses ; *S-N* curve for smooth specimen (reference) behavior (α = 180° and K_t = 1) and notched is also plotted as shown in Fig. 7 and the fatigue life equation using least square method for each type of loading mode was found as shown in in table 4. Stress concentration factor based on different angle orientation and notch depth for V notch shape location at the maximum bending region and at the central of the effective length due to bending ,axial and torsion was found ,hence different curves related with (t/D) ratio were generated as shown in Fig. 8, for more details of the procedure for stress concentration determination see [16].

Fatigue stress concentration factor for different fillet radii was found and plotted in Fig 9.

Also the results obtained show that the maximum value of stress occurs at the vicinity of change in cross section of the cylindrical specimens where a V notch is found.

From FEA results it is found that the Torsion loading is less damaging than bending and axial loading in a circumferentially notched bar .

On the other hand it is found that fatigue analysis obtained in torsion and axial loading modes are less affected by the notch position compared to bending load.

From the fracture surface for each case of loading type ,it is notice that for the same maximum principal stress amplitude, crack growth was higher in bending than torsion and the crack length in torsion is higher than this occurred in axial loading at same condition as shown in Fig. 6.

Table 4: The fatigue Limit for different loading mode

Case	S-N Equation	Fatigue limit (Mpa)
Reversed bending	$\sigma = 3234.5\ N^{-0.195}$	218.672
Reversed Axial	$\sigma = 1372.1\ N^{-0.168}$	134.7
Reversed Torsion	$\tau = 2005.8\ N^{-0.211}$	108.714

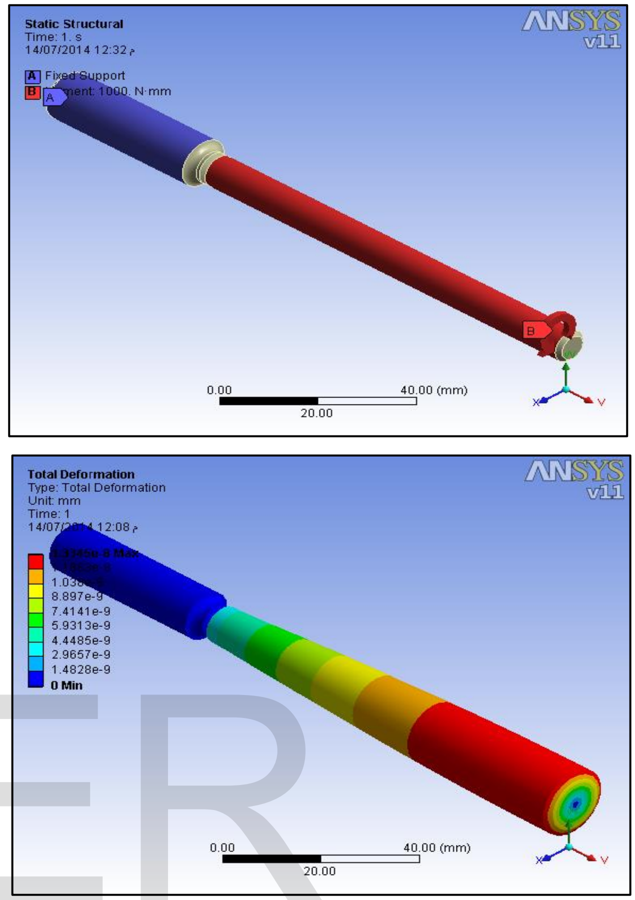


Fig. 5: Fatigue analysis for the model under Torsional Load

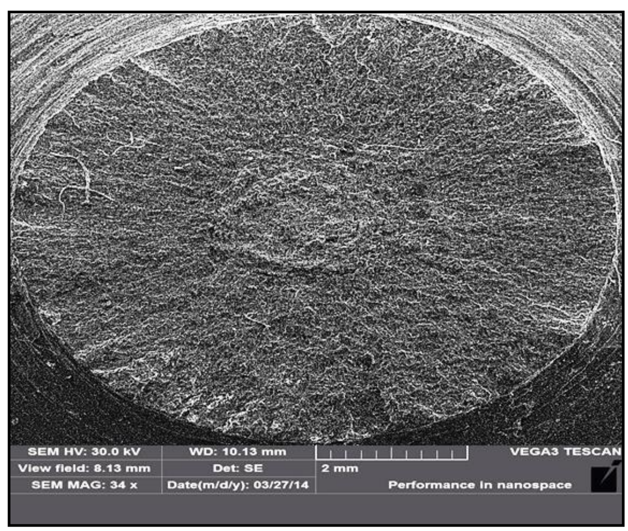


Fig. 6:Fracture surface of a specimen subjected to the loading of bending

ISSN 2229-5518

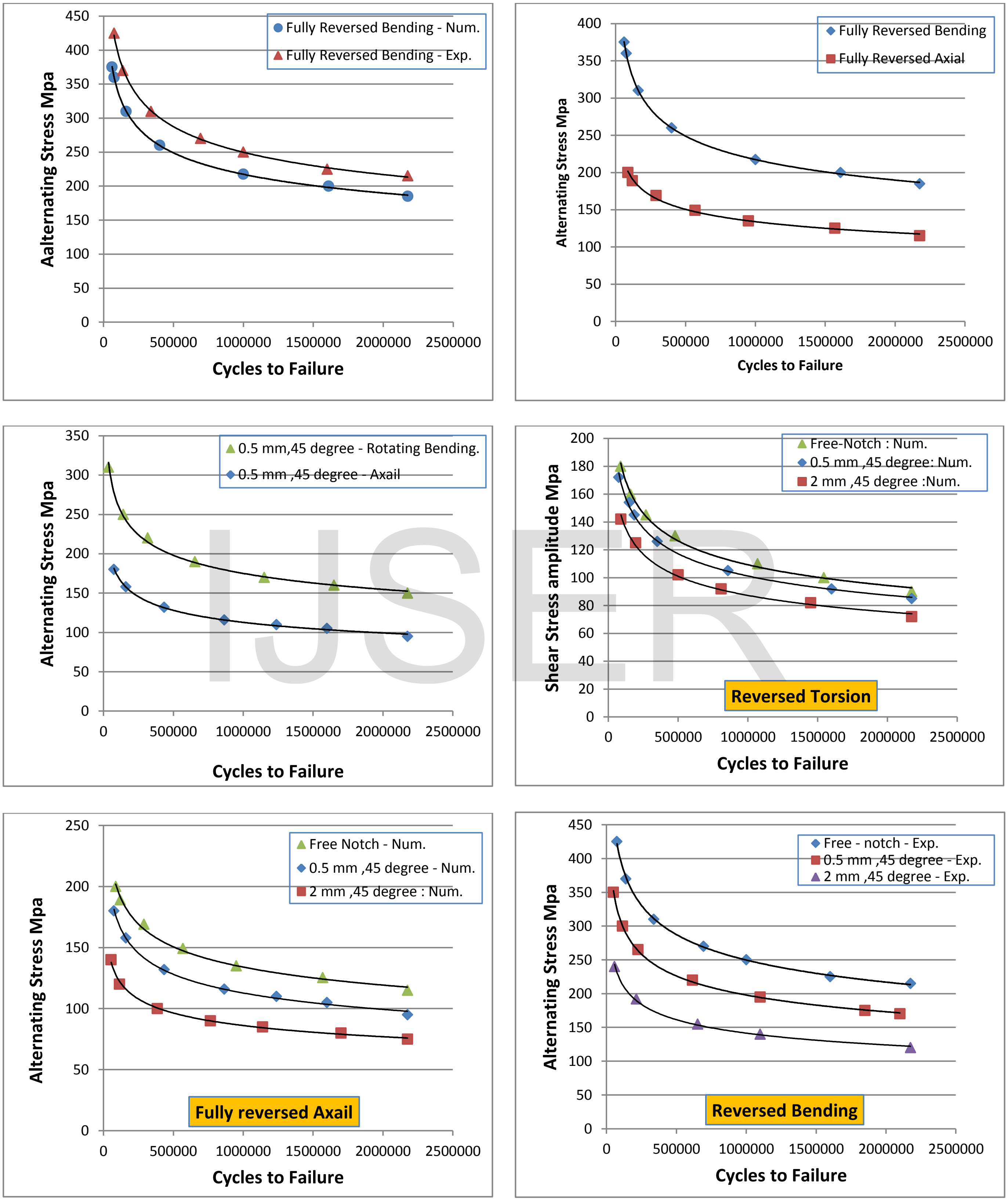
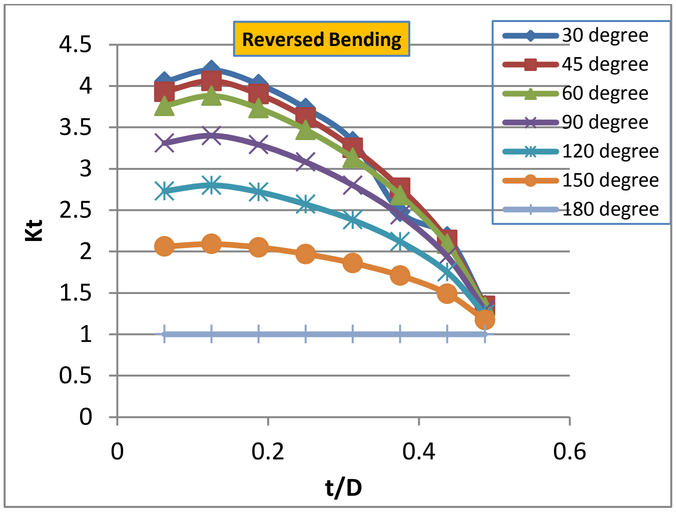
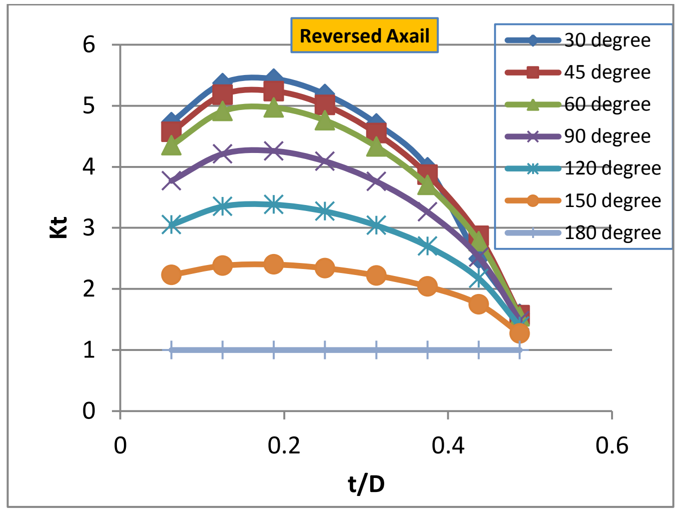
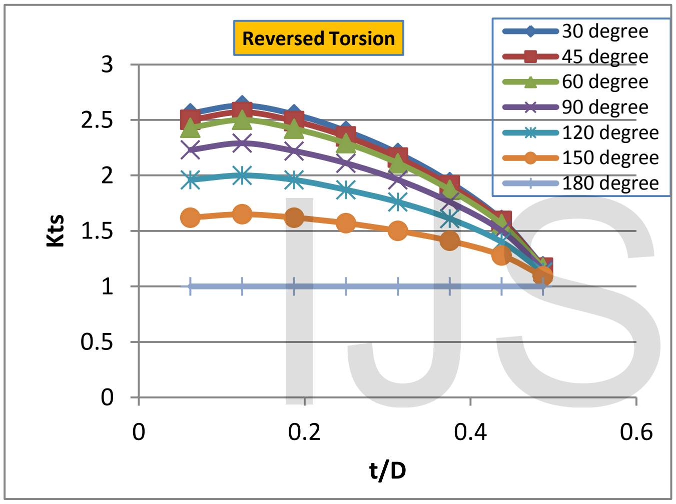
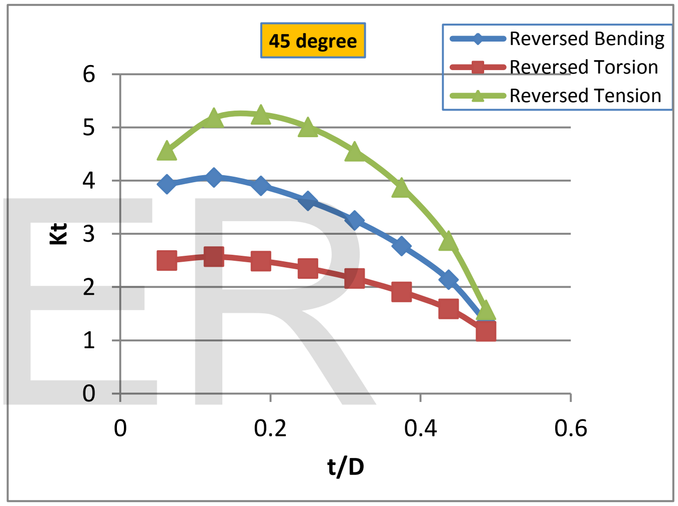


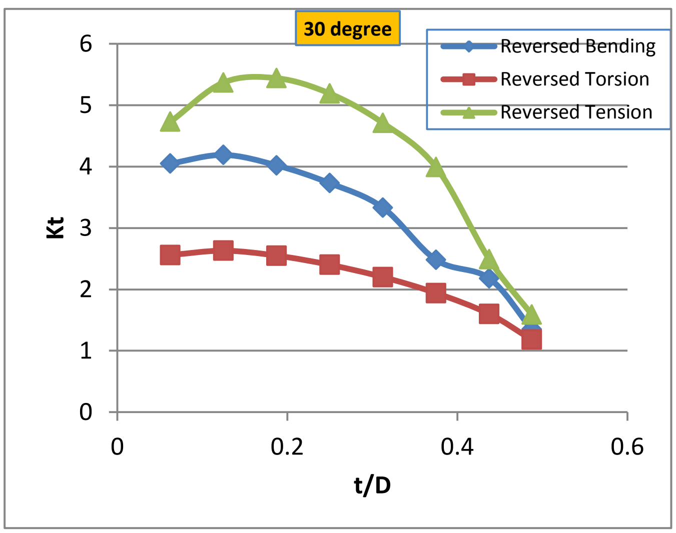
Fig. 7 : Experimental & Numerical S-N curve for different load conditions

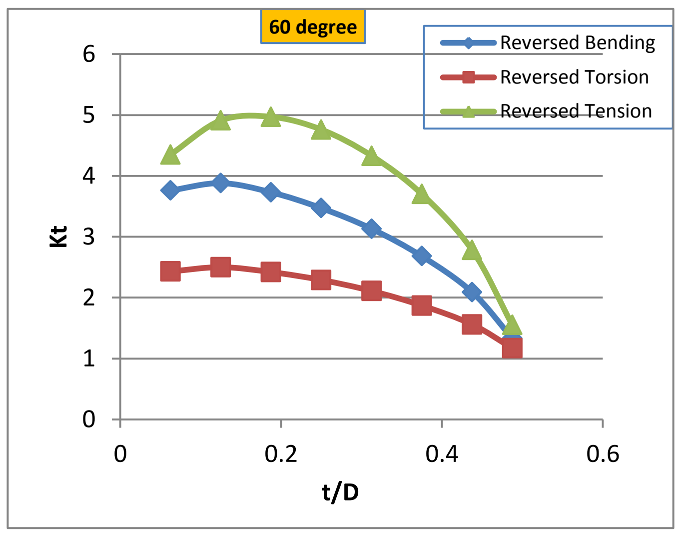

Reversed Bending
30 degree
45 degree
60 degree
90 degree
120 degree
150 degree
180 degree
Kt
t/D


Reversed Axail
30 degree
45 degree
60 degree
90 degree
120 degree
150 degree
180 degree
Kt
t/D


Reversed Torsion
30 degree
45 degree
60 degree
90 degree
120 degree
150 degree
180 degree
Kts
t/D


45 degree
Reversed Bending
Reversed Torsion
Reversed Tension
Kt
t/D


30 degree
Reversed Bending
Reversed Torsion
Reversed Tension
Kt
t/D


60 degree
Reversed Bending
Reversed Torsion
Reversed Tension
Kt
t/D

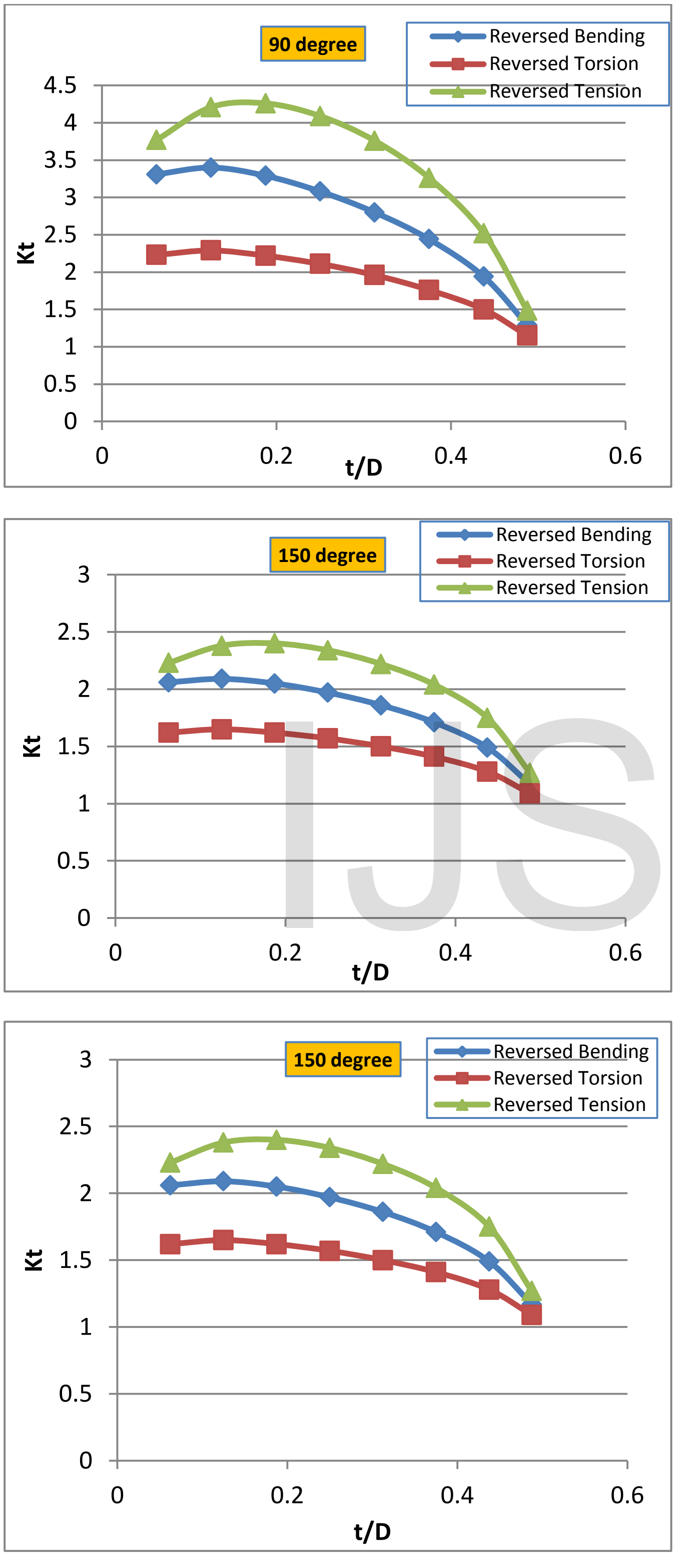


Fig. 8: Stress Concentration Factor under different Load

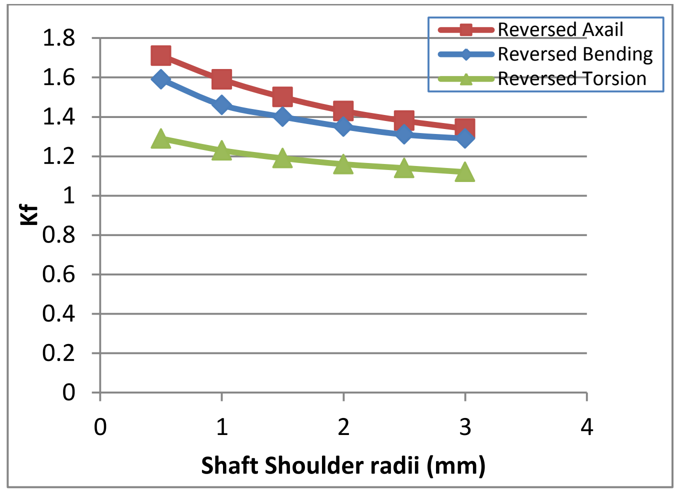


Fig. 9: Variation of fatigue stress concentration with shaft shoulder radii

5- CONCLUSIONS:

In this work, bending fatigue life of plain and notched specimens with V shape notch of various angle orientation and notch depth under different loading mode was investigated by the experiment and FEA method. The mathematical form of fatigue life equation of each load was obtained. From the results it is also observed that in most of the cases the ratio of fatigue limits for a material using axial , and reversed bending tests using von Mises failure criterion is greater than of those calculated by torsional and reversed bending load respectively as explained in below relationships and this applicable with information in R [17].

$$\sigma_e(\text{Axial}) = 062\ \sigma_e\ (\text{Bending})$$
$$\sigma_e(\text{Torsion}) = 0.5\ \sigma_e\ (\text{Bending})$$

It is observed that the life prediction FEA simulation is acceptable for different stress amplitudes and also at different no. of cycles. The accuracy of the predicted life by FEA simulation depends on the selection of appropriate material model and the accuracy of the value of the material parameters used . It is very important to know that the prediction in this method depends on the correctness of the material total S-N curve (simple regression) generated from experimental results of high cycle fatigue data of cylindrical specimens and the accuracy of simulated value of maximum stress of notched specimens .The maximum error between two methods found 9 %. The common stress life curve generated from several specimens gives a better prediction, which is apparent from the figure 7. The stress concentration factor for the notched specimen where the V notch located at maximum bending region and at the central location under each load type were found by FEA and analytically based on tables presented in Peterson curves [18] . The results indicate the FEA method is applicable to the experiments with overhaul error no more 6 %. Also the results show stress concentration factor increase with decrease angle notch orientation and decrease with increase of notch depth

As well as it is seen also from Fig. 8 that stress concentration factor resulted from axial load is greater than other two loading modes and the notch effect is very small in torsional fatigue in comparison to bending and axial fatigue [19].

ACKNOWLEDGMENT

This work was supported by the Laboratory Staff of Kufa University ,College of Engineering, Mechanical department.

REFERENCES

[1] G.M.Maitra, "Handbook of Mechanical Design", 2nd Edition, Tata McGraw-Hill co., Newdelhi , 1997.
[2] Chris Hinnant ,"Fatigue Design of Process Equipment", ASME Plant Engineering & Maintenance Technical Chapter, 2009.
[3] Richard G. Budynas and J. Keith Nisbett ," Shigley's Mechanical Engineering Design" , The McGraw-Hill Companies, Inc.2011
[4] Guy Pluvinage and Marenglen Gjonaj , "Notch Effects in Fatigue and Fracture ", NATO Advanced Research, Kluwer Academic Publishers, 2001.
[5] Nasim Daemi, Gholam Hossein Majzoobi "Experimental and Theoretical Investigation on Notched Specimens Life Under Bending Loading", International Journal of Mechanical and Materials Engineering 2011.
[6] A. Fatemi ,D. Fang , and Z. Zeng ,"Notched Fatigue behavior under Axial and Torsion loads : Experiments and predictions" ,8th International Fatigue Congress , Stockholm, Sweden, Vol. 3, PP. 1905-1914, June 2002.
[7] Ralph I. Stephens, Ali Fatemi , " Metal Fatigue In Engineering" , Second Edition , McGraw Hill, 2001.
[8] F O Sonmez ,"optimal shape design of shoulder fillets for flat and round bars under various loading", JMES1457 , Vol. 223 Part C ,2009.
[9] Bourbon Atibaia Spa Resort " Fatigue Analysis of a Simple Shaft under Combined Bending and Torsion",Esss conference & Ansys users meeting ,2013.
[10] Y.Verreman1, H.Guo, "Short cracks at notches and fatigue life prediction under mode I and mode III loadings", Nordam Group, Tulsa, OK, USA,2000.
[11] Deepan Marudachalam M. G ,et.al, "Optimization of shaft design under fatigue loading using Goodman method", International Journal of Scientific & Engineering Research Volume 2, Issue 8, August-2011.
[12] John E .Bringas," Handbook of Comparative world Steel Standards", ASTM data series ; DS67B,3rd edition,2004.
[13] QASIM BADER & EMAD K. NJIM "Effect of V notch shape on Fatigue Life in Steel beam made of Mild Steel AISI 1020 ", International Journal of Mechanical and Production Engineering Research and Development (IJMPERD) ,ISSN(P): 2249-6890; ISSN(E): 2249-8001 ,Vol. 4, Issue 4, 2014.
[14] Jaap Schijve , Fatigue of Structures and Materials, Springer ,Second Edition, 2009.
[15] M.R. Mitchell, "Fundamentals of Modern Fatigue Analysis for Design, Fatigue and Fracture", Vol. 19, ASM Handbook, ASM International, 1996.
[16] G.H. Majzoobi, N. Daemi, "The study of notch geometry on fatigue life using notch sensitivity factor", Transaction of the Indian Institute of Metals Journal. Vol. 63, 2010, pp.547-55.
[17] Bannantine, J.A. Comer, J.J., Handcrock, J.L., "Fundamentals of Metal Fatigue Analysis" , Prentice-Hall 1990.
[18] Walter D. Pilkey, Deborah F. ,"Peterson's stress concentration factors", John Wiley & Sons, Inc., 2008 by
[19] Ali Fatemi ," Notched Fatigue Behavior under Axial and Torsion Loads ", International Congress France , 2013.

- Qasim Bader is currently Ph.D. ,lecturer in College of Engineering ,University of Babylon ,Iraq , E-mail: drqasimbader@yahoo.com
- Emad K. Njim is currently pursuing master's degree program in applied mechanics, College of Engineering ,University of Babylon, Iraq, E-mail: emad.njim@gmail.com