International Journal on Applications in Mechanical and Production Engineering
Volume 1: Issue 6: June 2015, pp 6-13. www.aetsjournal.com ISSN (Online) : 2395-3500

Fatigue Analysis of Lay Flat Pipe under Pulsating Pressure Loads

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Abstract — Flexible Lay Flat hoses are sometimes placed at compressor discharge to minimize the effect of vibrations on piping and vessels. Doing well for this purpose, these elements can suffer dynamic loads from gas pulsation and fail. The present thesis will only focus on the response of flexible pipes used in compressor discharge of a fully dynamic riser systems connecting offshore production and offshore systems is investigated. The connection is a 100 mm Vulcanized Nitrile PVC hose with 5 mm thickness, subjected to a varying pressure load from the compressor. In this paper, Flexible Lay Flat hoses were subjected to a series of fatigue tests in which the pipes at different radii of curvatures, were subjected to the varying pressure loads similar to working conditions, which generates a three dimensional stress–strain states, then, the life of the structures were extracted in term of number of cycles, further, a finite element model is created and the numerical simulations were used to estimate and compare with the experimental fatigue life by using different approaches: a stress–strain life approach which is widely used in commercial codes, a critical plane approach, an energy approach and a continuum damage based approach, then, it is shown that for these type of structures the critical plane model is more convenient to predict the fatigue life.

Keywords**—**Lay Flat Hose, Fatigue models, pulsating pressure, Multiaxial stress state.

I. INTRODUCTION

Lay Flat hoses are tubes invented in the early 1790's. It allows different placement configurations like bending, tension-compression, and shearing, in general it is made from polymer materials like Polyvinyl Chloride where it is used widely in the industry as in piping systems for most factories, power plants, all kinds of engines, oil industry and ship buildings, because it provides all the properties wanted in a pipe which are the low weight, the flexibility, high pressure resistance up to 20 MPa, vibration and noise damping and high temperature environment between 0 C and 400 C. The fact that such structures are usually linked to an exhaust system, engines, or a high velocity fluid sources, makes them subjected to a repeated mechanical and thermal loadings, which lead to a fatigue damage in most cases, in this study, the thermal loading is not taken into account to simplify the experiment.

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This study could be divided into two parts, the first one is an experimental investigation about the lifetime of the Lay Flat Hose tube where a test is designed to reproduce the service conditions of the product like the varying pressure loads applied on it. The second part is a numerical study in which finite element analysis is used to simulate the experimental test using the commercial code ANSYS Workbench 14.0, then the data resulting from the FE simulations are implemented into a written algorithms that compute the fatigue life based on different criteria, which are: Coffin– Manson[1] as a strain based approach, Fatemi and Socie[2] as a critical plane approach, Halford[3] as an energy based approach and Chaboche[4] as a continuum damage based approach.

II. FLEXIBLE LAY FLAT PIPES

Flexible lay flat pipes are widely used in offshore oil and gas industry as connections for both fixed and floating platforms. Examples are risers connecting seabed to the surface and well product flowlines. The presence of flexible pipe offers the permanent connection between floating structures and subsea installations under extreme dynamic conditions because it can suffer larger bending deformations. At the same time, it is easy to transport and install. The most important characters for flexible pipe are lower bending stiffness and higher volume stiffness.

Application

The Common applications[5] for flexible pipes are listed below:

- Riser used to connect subsea installations with above water production facilities during the production phase.
- Flow lines for in-field connection of wellheads, templates, loading terminals, etc.
- Jumper lines between fixed and floating platforms.
- Loading hoses for offshore loading terminals.
- Small-diameter service lines, such as kill and choke lines, umbilicals, etc.

The present thesis will only focus on the response of flexible pipes used in fully dynamic riser systems connecting offshore production and offshore systems. Thus, only details on riser systems are discussed.

Key parameters for a riser system are the number, sizes, pressure, rating and internal coating requirements related to transfer function

According to [5], the requirements for the riser system support depend on

International Journal on Applications in Mechanical and Production Engineering
Volume 1: Issue 6: June 2015, pp 6-13. www.aetsjournal.com ISSN (Online) : 2395-3500

- Water depth
- Support vessel motions
- Current and wave loading

Key parameters for external loadings are:

- Tension
- Curvature
- Torsion
- Contact with other structures(interference)
- Compression

Common used riser configurations are given in Fig. 1[6]

A. General configuration

B. Free hanging catenary

This is the simplest and cheapest configuration. This type of riser has poor dynamic properties and severe response at the touch down area where the riser is connected to the vessel. It is reported that this type of configuration is likely to have compression buckling under large vessel motions[3] and large top tension when applied in deep water.

C. Steep S

For this shape of pipe, a buoy, either fixed or floating, is used to connect to the pipe to create the S-shape. The buoy can absorb the tension variation from the platform, and such the touch down point (TDP) has only small variation in tension. The S configuration is costly and only used for complex installation procedure. The steep S has larger curvature than lazy S.

D. Steep wave

In this configuration, buoyancy elements which is made by synthetic foam, is mounted on a longer length of the riser. A negative submerged weight is created by the buoyancy elements and a wave shape is made. This will decouple the vessel motion from the TDP. The steep wave configuration also needs a subsea base and a subsea bending stiffener. This type of configuration is commonly used in cases where the internal fluid density changes during life time. One problem of wave configuration is the buoyance elements tend to lose volume under high pressure and thus results an increased submerged weight. The wave configuration has to be designed to have up to 10% loss of buoyancy.

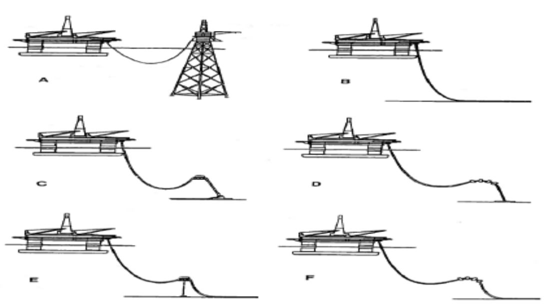

Fig 1. Flexible Riser Configurations

E. Lazy S

Lazy S configuration is similar with steep S configuration but with smaller curvature.

F. Lazy wave

The lazy wave configuration has similar principle as for steep wave configuration but with smaller curvature. The main difference is that the lazy wave does not require a subsea base and bend stiffener. Lazy wave is lower cost and more often preferred.

In this thesis paper the Lazy Wave configuration is taken as it is lower cost and more often preferred. The Standards used in the Lazy Wave configurations are: 5D, 7D, 10D, 13D and 20D where D(100mm) is the diameter of the pipe.

III. MULTIAXIAL FATIGUE CRITERIA

Several researches were meant to express the fatigue life of a structure without doing an experimental study, where the fatigue life is expressed as a function of different independent variables like the stress amplitude, the mean stress, the strain amplitude, and also some material parameters that should be defined experimentally. In this time there is four approaches that give the lifetime: the stress–strain approach, the critical plan approach, the energy approach and the continuum damage mechanics approach.

A. The stress–strain based approach

The stress–strain approach is the simplest and the oldest way to compute the uniaxial fatigue life, it began with a stress based model known as Basquin's[7] relation between the stress amplitude σ_a and the lifetime N_f:

$$\sigma_a = (E\Delta\varepsilon)/2 = \sigma_f'(2N_f)^b \tag{1}$$

where σ_f', E, b are respectively the fatigue strength coefficient, Young's modulus and the fatigue strength exponent to be determined experimentally. For the cases with plastic strain; Coffin–Manson[8] proposed a strain based model which have the same form as the stress based one:

$$\varepsilon_a^p = (\Delta\varepsilon^p)/2 = \varepsilon_f'(2N_f)^c \tag{2}$$

where ε_a^p, ε^p, c are respectively the plastic strain amplitude, the fatigue ductility coefficient and the fatigue ductility exponent. Because equation represent the fatigue life due to elastic strain and equation represent the fatigue of the plastic strain; the combination of the two equations leads to the total strain life of Manson and Hirschberg[9] :

$$\varepsilon_a = \varepsilon_a^p + \varepsilon_a^e = \varepsilon_f'(2N_f)^c + (\sigma_f'/E)(2N_f)^b \tag{3}$$

because of its simplicity; equation is the most used formula in the Industry especially in the commercial codes like Ansys, MSC-Fatigue and FE-Safe, but, it requires five material parameters to be determined, to avoid this problem; a simpler formula known as the universal slope was proposed by Manson[10] where the constants b and c are given respectively -0.12 and -0.6 for all metals, so only three constants are left which are ε_f', σ_f', E Due to its major influence on the fatigue life; the mean stress was considered in the previous models by applying what is known as the mean stress correction in which the stress amplitude is modified using a function of the mean stress, the most famous corrections are[11]: Gerber, Goodman, Soderberg, Morrow, Walker and SWT, for example SWT[12] introduce σ_{max} as the sm of stress amplitude and mean stress. so Eq.(3) becomes:

$$\varepsilon_a \sigma_{max} E = \varepsilon_a^p + \varepsilon_a^e = E \varepsilon_f' \sigma_f' (2N_f)^c + (\sigma_f')^2 (2N_f)^b \quad (4)$$

In most engineering structures, the stress–strain state is multiaxial, so clearly, the previous formulations are not applicable, because in the three dimensional cases; there is at least 3 stress components and the same for the strain, which means that each component has its own amplitude and mean value, often, to overcomethis issue; engineers use an equivalent stress and strain using

the three dimensional values to convert the multiaxial fatigue state into an equivalent uniaxial fatigue case that generates the same fatigue life, often, the equivalent stress or strain are expressed by famous relations like Von Mises, Tresca, and hydrostatic stress, like Crossland[13] who used the maximum hydrostatic stress during the cycle, and Sines[14] who took its average value. The stress– strain approach equations were the base for the second approach which is the critical plane theory.

B. The critical plane approach

The critical plane approach relies on the physical fact stating that the damage will always occur on the plane which has the maximum shear strain, and the crack will propagate as a result of the normal stress acting on it, in on other way, during the fatigue cycle; the most shear loaded plane, known as the critical plane; will be found and expected as a crack nucleation region, which means that the multiaxial fatigue behavior should be governed only by the stress–strain components of this plane. This concept was firstly postulated by Findley[15] who introduced the effect of the normal stress on the maximum shear plane:

$$((\Delta\tau/2) + k\sigma_n)_{max} = C \quad (5)$$

where k, C are material constants, $\Delta\tau$ and σ_n are the shear stress amplitude and the normal stress; both in the maximum shear plane. Then Brown and Miller[16] suggested that the fatigue should be expressed in terms of shear and normal strains and not stresses:

$$((\Delta\gamma/2) + k\varepsilon_n)_{max} = C \quad (6)$$

With this modification and the shear form of Eq. (3); the fatigue life is expressed as:

$$((\Delta\gamma/2) + k\varepsilon_n)_{max} = \gamma_f'(2N_f)^{c_0} + (\tau_f'/G)(2N_f)^{b_0} \quad (7)$$

where G is the shear modulus. Further, Fatemi and Socie[2] showed that the hardening of the material effects the fatigue life, so they proposed a modified formula:

$$(\Delta\gamma/2)\left(1 + k\frac{\sigma_n}{\sigma_y}\right) = \gamma_f'(2N_f)^{c_0} + (\tau_f'/G)(2N_f)^{b_0} \quad (8)$$

where σ_y is the yield stress and b_0, c_0 are material parameters, other research are done in the same spirit depending on the materials and the load type like McDiarmid[17].

C. The energy based approach

The idea of using energy to characterize fatigue is relatively recent. Moreover, this approach is so convenient when more than one type fatigue loading is acting at the same time on the structure, like the cases when a multiaxial load is joined with thermal and creep cycles[18]. The dissipated energy during a cyclic loading Δw is the some of the elastic and the plastic strain energy, where the plastic part is established using hardening models and it could be expressed for steels as:

$$\Delta w = \frac{1-n}{1+n} \Delta\sigma \Delta\varepsilon_p \quad (9)$$

where n, $\Delta\sigma$, $\Delta\varepsilon_p$ are respectively the hardening exponent, the stress amplitude and the plastic strain amplitude, meanwhile, an equivalent values for the stresses and the strains are established depending on the model but usually Von Mises equivalent stress–strain criteria is a good assumption. Using this energy concept, Halford[3] suggested that the fatigue life could be written as follow:

$$\Delta w = A(2N_f)^{\alpha} \quad (10)$$

where A and α are material parameters. Moreover, the effect of the mean stress could be implemented using the classical mean stress corrections[11]. Other modifications were proposed for the cases where the loading amplitude is not constant because the difficulty rises with computing the plastic strain energy.

D. The continuum damage approach

Using the Damage concept in the fatigue life prediction was first introduced by Miner[19] where the life of a structure subjected to a different successive fatigue cycles is expressed as a linear accumulation of the residual life from each amplitude:

$$D = \sum_{i=1}^{n} \frac{n_i}{N_i} = \sum_{i=1}^{n} r_i \quad (11)$$

where n_i is the number of cycles at a given constant amplitude and N_i is the number of cycles to damage under the same stress amplitude, while D is the damage variable, it takes a value between zero and one, when it is equal to 1, it means that the structure is out of service. This formulae is known as Miner's rule because its widely used but usually in another form because this one gives acceptable results only for simple uniaxial cases with some materials[20], for example, at each constant load ni, an exponent is applied which depend on the material, the load itself and the boundary conditions[21]. Moreover, other models were proposed by joining the damage variables and the loading and usually each model is validated for a given material and load type, most of these models are discussed in Fatemi and Yang[22] review. In this paper Chaboche[23] is used because it is one of the most sited cumulative damage models and its material parameters could be easily found in the literature for most metals especially steel alloys. The damage variable for the uniaxial case is expressed as:

$$dD = \left[1 - (1 - D)^{\beta+1}\right]^{\alpha} \left[\frac{\sigma_{max} - \sigma}{M_0(1 - b\sigma)(1 - D)}\right]^{\beta} dN \quad (12)$$

while β, b and M_0 are a material parameter, σ_{max} is the maximum stress value during the cycle, σ is the mean stress, and α is a load dependent parameter expressed as:

$$\alpha = 1 - a \left\langle \frac{\sigma_{max} - \sigma - \sigma_l}{\sigma_u - \sigma_{max}} \right\rangle \quad (13)$$

where a is a material parameter, the operator $\langle \frac{\sigma_{max} - \sigma - \sigma_l}{\sigma_u - \sigma_{max}} \rangle$ is defined as:

$$\langle x \rangle = \begin{cases} x, & x \geq 0 \\ 0, & x < 0 \end{cases} \quad (14)$$

σ_l and σ_u are respectively the fatigue limit and the ultimate tensile stress, in which the stress amplitude effect and the hardening could be considered as follow:

$$\sigma_u = \sigma_{u_o}(1 - b\sigma)(1 + k_1 Z) \quad (15)$$

$$\sigma_l = \sigma_{l_o}(1 - b\sigma)(1 + k_2 Z) \quad (16)$$

where k_1, k_2 are a material coefficients, σ_{u_o} and σ_{l_o} are respectively fatigue limit and the ultimate tensile stress when the stress amplitude is equal to zero, and Z^2 is the equivalent plastic strain at the beginning of the loading cycle. By integration the damage variable from equation between 0 and 1, and the fatigue life variable between 0 and N_f the number of cycles to damage for uniaxial cases is expressed as:

$$N_f = \frac{1}{(1+\beta)(1-\alpha)} \left(\frac{\sigma_{max} - \sigma}{M_0(1 - b\sigma)}\right)^{-\beta} \quad (17)$$

In the multiaxial cases, a similar formulae is used as in uniaxial stresses with equivalent stresses, where the stress amplitude σ could be replaced by the average hydrostatic stress during the loading cycle σ_H where $\sigma_H = moy\,(tr(\sigma))/3$, also, the maximum stress σ_{max} is replaced by an equivalent maximum stress $\sigma_{eq\ max}$, and the value $\sigma_{max} - \sigma - \sigma_l$ in the load variable α becomes with $(A_{II} - A_{II}^{*})$ with:

$$\sigma_{eq\ max} = \max\sqrt{\frac{1}{2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]} \quad (18)$$

$$A_{II}^{*} = \sigma_l(1 - 3b\sigma_H) \quad (19)$$

$$A_{II} = \max\sqrt{\frac{1}{2}\left[(a_1 - a_2)^2 + (a_2 - a_3)^2 + (a_3 - a_1)^2\right]} \quad (20)$$

where σ_i are the principal stresses, and a_i are the amplitude of each one $a_i = \Delta\sigma_i/2$, so the loading parameters α gets the form:

$$\alpha = 1 - a \left\langle \frac{A_{II} - A_{II}^{*}}{\sigma_u - \sigma_{eq\ max}} \right\rangle \quad (21)$$

finally the fatigue life under multiaxial loading could be expressed as:

$$N_f = \frac{1}{(1+\beta)(1-\alpha)} \left(\frac{A_{II}}{M_0(1 - 3b\sigma_H)}\right)^{-\beta} \quad (22)$$

IV. EXPERIMENTAL PROGRAM

A. Determining the Frequency of the varying Pressure curve

The Fig 2 shows the pressure variation with respect to time taken from the outlet of a 0.25 HP reciprocating compressor pump with 720 RPM and 30mm stroke length giving a discharge of 30 lps. From the graph obtained for a time period of twenty seconds with a time interval of 0.005 seconds, the frequency of the pressure variation is obtained as 0.46 Hz by using Matlab program. The frequency plot is shown in Fig 3. From the frequency it is possible to obtain the number cycles the pipe is subjected to when it is connected to the compressor. Fig 4 shows the variation of pressure in two cycles, i.e 4.3 seconds

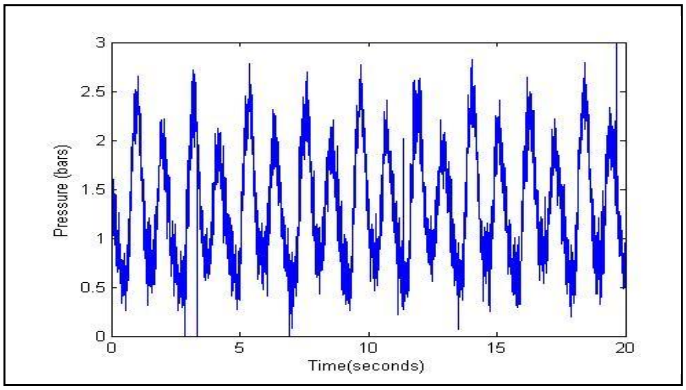

Fig 2 Pressure variation with respect to time from the reciprocating compressor

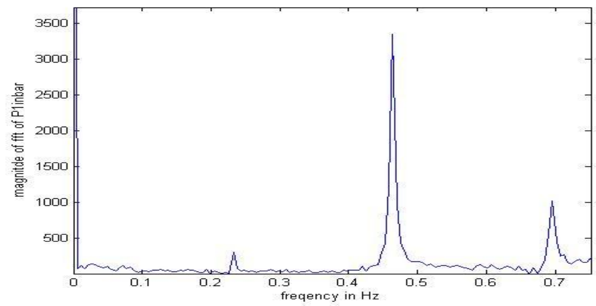

Fig 3 The frequency plot of the pressure curve

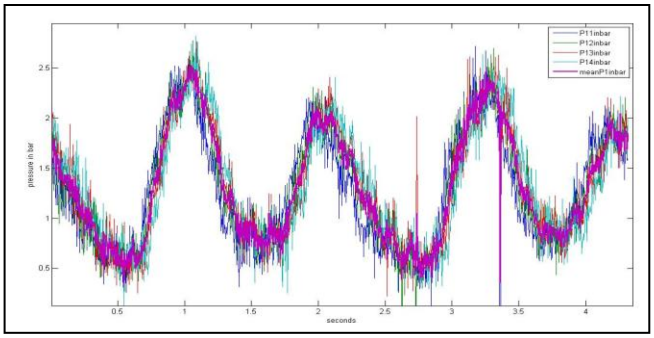

Fig 4 shows variation of pressure in two cycles

B. Experimental Determination of the Fatigue life

Now the experiment is conducted for each radii of curvature (5D, 7D, 10D, 13D and 20D where D = 100mm) of the 100mm diameter flexible Lay Flat hoses by connecting it to the compressor and recording the time for failure of the pipe. The flexible pipe is considered failed when it loses the pressure, which means that there is a crack on it, because the structure is made of a thin (5 mm), so any initiated crack will propagate immediately through the thickness which will eventually let the fluid dissipate from the tube. Fig 5 shows the failure of the flexible hose connected to the compressor.

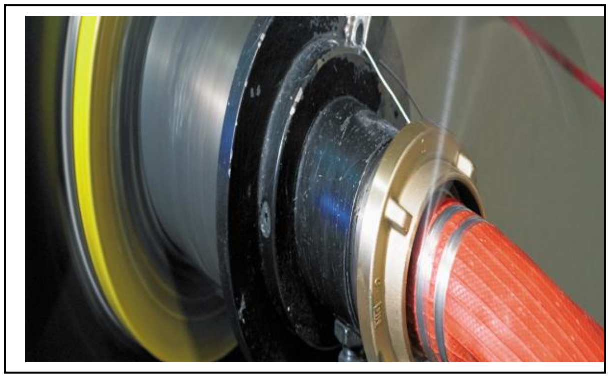

Fig 5 shows the failure or crack initiation in the flexible pipe

C. Result :

The experiment result is tabulated as shown in Table I. The experiment shows the there is a logarithmic relationship between the fatigue life and the radii of curvature. As the radius of curvature increases, the fatigue life also increases.

TABLE I

Fatigue Life Vs Radius Of Curvature

RADIUS OF CURVATURE (MM) D = 100 MM	**LIFE (CYCLES) 1 CYCLE = 2.15 SECONDS**
20D	80653
13D	54067
10D	42781
7D	23986
5D	11473

V. NUMERICAL STUDY

A. Finite Element Model :

A finite element model was created to simulate the experimental test by creating a three dimensional model for the tube geometry and choosing shell elements, because such the structures are made of thin metal sheets (5mm) where the thickness is way smaller than the other two dimensions. The model does not consider the effect of the pressure under the ocean to simplify the simulation and the fact that that this pressure does not affect the fatigue life of the pipe.

Figs 6 – 10 shows the stress distribution developed in the pipe for the pressure variation when subjected to two cylces as shown in the Fig 4.

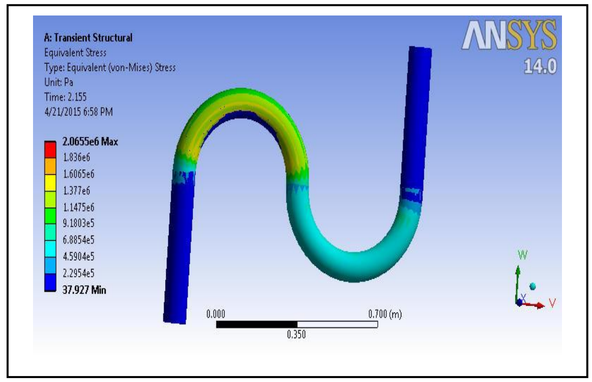

Fig 6 Shows the stress distribution in 5D (D = 100 mm) Curvature of the pipe

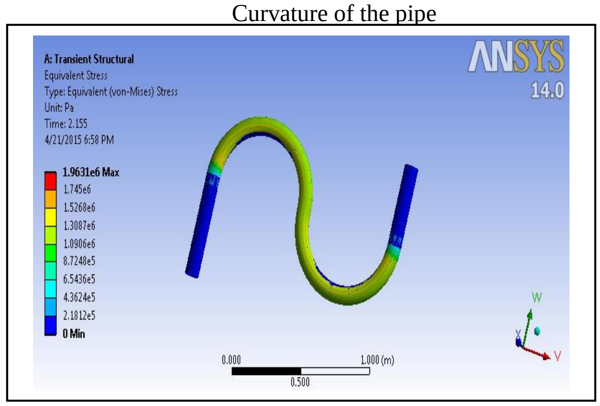

Fig 7 Shows the stress distribution in 7D (D = 100 mm) Curvature of the pipe

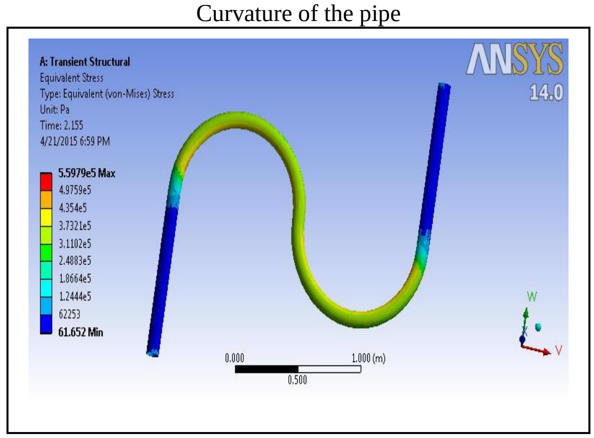

Fig 8 Shows the stress distribution in 10D (D = 100 mm) Curvature of the pipe

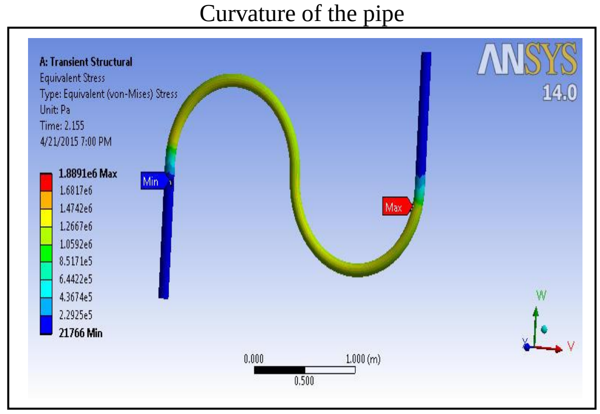

Fig 9 Shows the stress distribution in 13D (D = 100 mm)

Curvature of the pipe

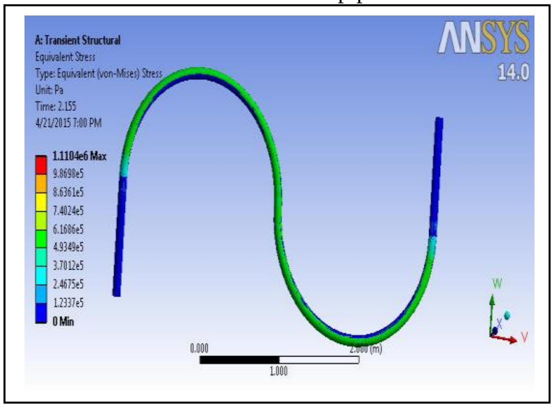

Fig 10 Shows the stress distribution in 20D (D = 100 mm) Curvature of the pipe

B. Fatigue life Estimation

Since, the pressure varies according to the time, transient structural non linear analysis is done. The pressure input for one cycle (2.155 seconds) is given. The stresses and the plastic strains data resulting from the FE simulation of a cycle; will be used by a written Matlab functions in order to compute the fatigue life. Four different formulations were used to get the life of the structure: the stress–strain life using Coffin–Manson relation Eqn(3), the critical plane approach using Fatemi and Socie model Eqn(8), the energy approach using Halford model Eqn(10) and the continuum damage approach using Chaboche model Eqn(22). Each Matlab program will compute the fatigue life in all the nodes using the data from the FE analysis, then it returns the minimum value which is the life of the pipe.

In the fatigue analysis, the common material parameters given in the Table II for the vulcanized Nitrile PVC compound is $\sigma_f' = 406$ MPa, $\varepsilon_f' = 0.391$, $\sigma_l = 37$ MPa and $\sigma_u = 40$ MPa. For the Coffin–Manson and Fatemi–Soci models the used parameters are b = -0.052, c = -0.601, k = 1, $\tau_f' = \sigma_f'/\sqrt{3}$ and $\sigma_y = \sigma_f'$. In the energy approach, the constants from Halford model are taken as A = 580.8 and a = -0.628, also the hardening exponent is taken as n = 0.0592. For the continuum damage approach the needed parameters are $\beta = 5$, b = 0.25, $M_0 = 1650$ MPa, a = 0.9, $k_1 = 0.4$, $k_2 = 2.2$.

TABLE II
Material Property

COFFIN MASON MODEL	
fatigue strength coefficient (σf')	406 Mpa
fatigue ductility coefficient (εf')	0.391
fatigue strength exponent (b)	-0.052
fatigue ductility exponent (c)	-0.601

FATEMI SOCI MODEL	
yeild strength (σy)	37 Mpa
shear strength coefficient (τf)	234 Mpa
shear modulus (G)	69 Mpa

HALFORD MODEL	
Material Parameters	
A	580.8
a	-0.628
n	0.0592

CONTINUUM DAMAGE APPROACH MODEL	
Material Properties	
B	5
B	0.25
Mo	1650 Mpa
A	0.9
k1	0.4
k2	2.2

C. Result and Discussion

Tables III - VI show the estimated fatigue life compared to the experiment. The experimental data for each radii of curvature is compared and the percentage deviation in the values is calculated and is tabulated.

TABLE III Comparison Of Coffin Mason Model With Experimental Fatigue Life

COFFIN MASON			
Radius Of Curvature D = 100 Mm	**Life (Cycles)**	**Experimental Life (Cycles)**	**Percentage Deviation**
20D	9.84E+04	80653	0.220537
13D	7.53E+04	54067	0.392716
10D	7.52E+04	42781	0.75779
7D	3.20E+04	23986	0.334112
5D	2.17E+04	11473	0.891397
Average deviation			0.51931

TABLE IV
Comparison Of Fatemi Soci Model With Experimental Fatigue Life

FATEMI SOCI MODEL			
Radius Of Curvature D = 100 Mm	**Life (Cycles)**	**Experimental Life (Cycles)**	**Percentage Deviation**
20D	8.27E+04	80653	-0.025876285
13D	3.14E+04	54067	0.418499269
10D	1.50E+04	42781	0.64937706
7D	1.47E+04	23986	0.3871425
5D	1.22E+04	11473	-0.065109387
Average Deviation			0.272806631

TABLE V
Comparison Of Halford Model With Experimental Fatigue Life

HALFORD MASON			
Radius Of Curvature (D = 100 mm)	**Life (Cycles)**	**Experimental Life (Cycles)**	**Percentage Deviation**
20D	5.54E+05	80653	8.54E-01
13D	3.28E+05	54067	8.35E-01
10D	3.24E+05	42781	8.68E-01
7D	2.10E+05	23986	8.86E-01
5D	1.52E+05	11473	9.24E-01
Average Deviation			8.74E-01

TABLE VI
Comparison Of Chaboche Model With Experimental Fatigue Life

CHABOCHE MODEL			
Radius Of Curvature D = 100 Mm	**Life (Cycles)**	**Experimental Life (Cycles)**	**Percentage Deviation**
20D	9.90E+04	80653	2.27E-01
13D	7.55E+04	54067	3.96E-01
10D	3.72E+04	42781	-1.30E-01
7D	3.20E+04	23986	3.34E-01
5D	3.28E+04	11473	1.86E+00
Average Deviation			5.37E-01

Fig 11 shows the comparison between all the models with the experimental fatige life. By taking the fatigue life of the experiment as a reference; the average error of chaboche model is 53%, in Fatemi– Socie model it is 27%, 51% for Coffin–Manson model and 87% in Halford's energy model, yet, these error averages are considered very close in fatigue analysis even for 50%.

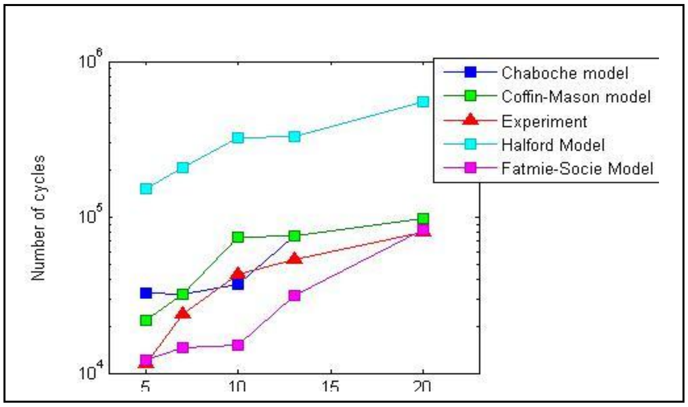

Fig 11 Comparison of all the models with Experimental Fatige life

VI. CONCLUSION

In this study, the experimental tests were conducted in order to investigate the effect of curvature on the flexible Lay Flat hose on the fatigue life, and it is shown that the fatigue life decreases exponentially when the radius of curvature decreases, i.e. the fatigue life is maximum when the tube is straight. the comparison between the four fatigue criteria show that Fatemi–Soci model is more suitable to estimate the life for such structures under such loadings, moreover, this model is more convenient because with acceptable amount of error; it requires less material parameters and less formulation complexity compared to Coffin Mason and Chaboche approach.

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