Non-linear static structural analysis of a butt weld joint in a drum pulley assembly

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Abstract— Drum Pulley Assembly used for material handling, conveyor systems, the most critical region for fatigue damage and failures are reported at the weld joint connecting plate and shell, hub and endplate as well as seam weld in drum shell. A failure analysis based on stress life approach may be useful for predicting the life time of weld in the structure. This study presents an upcoming methodology in new three dimensional Finite Element Model to calculate the fatigue life of weld. Ansys 12.1 simulation software uses stress-life method, based on a static non-linear Structural analysis. The weld material S-N curve were experimentally determined by the Fatigue testing of the dumbell specimen as per 7608 standard. This study assumes that a flaw exist in weld due to welding process, material in-homogeneity, air voids, slugs or impurities in weld, improper surface machining and many more. This material curve is used in simulation to get more accurate results. Thus the fatigue life prediction with the material curves from experimentation will give us more accurate and close to actual failure results.

KEYWORD: Drum Pulley Analysis, Fatigue Life by S-N approach, Equivalent Stress, Weld fatigue.

I. INTRODUCTION

Failures due to fatigue in welded structures lead to loss of life and substantial costs. Remedies to this situation include the introduction of various standards and fatigue design codes. The foundation of such codes rely, in some cases, on old concepts that do not easily translate to the output from modern computer programs and are also limited to rather simplified structures.

The development of new generations of products means, in general, increased capacity, increased speed and increased demands on life. Improved maintenance and higher utilization place additional demands on the supporting structures.

The requirements in society towards improved functionality and minimizing of Life Cycle Cost (LCC); force companies to design structures with reduced weights and “optimum” fatigue resistance. Actions to meet these demands are to introduce high strength steel, weld and/or surface improvement technologies and high productivity manufacturing technologies. The introduction of high strength steel in structures normally means higher stress levels and, hence, an increased sensitivity to defects, deviations in weld geometry (e.g. penetration, throat thickness, undercuts) and variations in material strength.

Expertise in developing and manufacturing fatigue loaded welded structure with low LCC is a key aspect in order to stay competitive. Shorter development time for new products means that it is important to make the correct design and fatigue assessment early on in the project.

A better understanding of the limits of the different fatigue design methods and the influence of fatigue strength due to the weld quality will improve the development of new fatigue loaded products. The understanding of the link between weld quality and the welding process would enable manufacturers to increase the utilization of high strength steel in fatigue loaded welded structures.

Being able to determine the rate of crack growth, an engineer can schedule inspection accordingly and repair or replace the part before failure happens. Being able to predict the path of a crack helps a designer to incorporate adequate geometric tolerance in structural design to increase the part life.

II. Objectives

The aims at design validation of a drum pulley assembly and fatigue analysis of weld joint which is most probable part to fail against fatigue. Hence it needs to find out the stresses in various components by nonlinear static analysis of the drum pulley assembly. Currently there are two models for the same application working under same conditions hence needs to suggest a model which will survive for a longer life.

The following are the main objectives

- Conventional design criteria:-

The conventional design procedure against static strength, fatigue strength is set for client use. Manufacturing criteria’s that should be satisfied during fabrication of assembly are also mentioned here.

- Non-linear Static Structural Analysis:-

To carry out non-linear static analysis of the conveyor pulley assembly using ANSYS Classic Version 12.1. This is done to find out safe value of resultant displacement (stiffness) and Von Misses stresses. A special attention is required to get convergence of non-linear system and then validation of converged results.

- Fatigue Life Calculation:-

To carry out the fatigue life prediction of welded joint using conventional methods. (S-N approach Method)

- Conclusions:- To conclude results obtained by each process and make a final remarks for the benefits of company and society.

III. THEORY OF DRUM PULLEY

The cost of a catastrophic failure in this power range can have grave consequences on personnel safety and on the plants profit and loss statement.

Manufacturers differ on the following criteria:

- Types of stresses that are important
- Fatigue stress range criteria
- Allowable stress limits in welded and non-welded members,
- Allowable load limits on the pulley components
- Material surface finish criteria in critically stressed areas
- Materials of construction and their limitations
- Fabrication techniques, constructed tolerances and tolerance controls.

IV. Description of Components of Drum Pulley

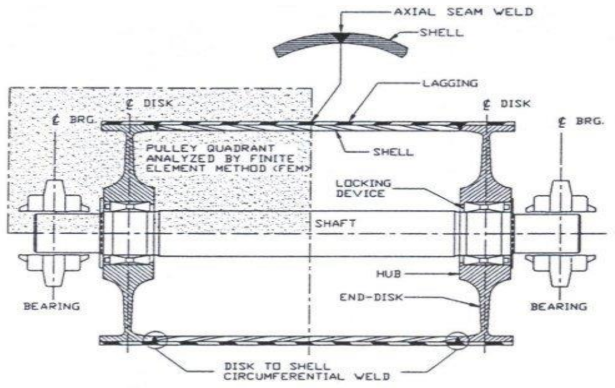

Figure 1. Components of Drum Pulley Assembly(9)

The main components of a pulley for a conveyor belt application are shown in Figure 1.

- Shell
- End-disk And Hub Assembly
- Locking Mechanism
- Shafts
- Lagging

A. *Design Criteria's for Pulley Components*

The stress criteria comprise of static and fatigue strength analyses. These stress criteria consist of setting limits on both the maximum stresses and on the stress range that can occur in different components of the pulley (shell, disk, hub and shaft). The three dimensional stress fields consist of radial, tangential and axial stresses, which are analyzed in the pulley.

B. *Static Strength Criteria*

While evaluating ductile materials, yield strength of the material is usually used as the failure criteria. In the case of brittle materials, like cast iron, which do not have a yield point, the ultimate strength of the material is used as the failure criteria. In general the Distortion Energy Theory for performing static strength analyses is used. This theory is meant for ductile materials as it predicts the initiation of yield. The Von Misse's stress is used in the theory. For a tri-axial stress state, the Von Misses stress is defined in terms of the principal stresses as:

$$\sigma_{von-misses} = [0.5 * \{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\}]^{0.5}$$

Principal stresses σ_1, σ_2 and σ_3 are normal stresses that act on planes that do not carry any shear stress. Maximum and minimum principal stresses act on mutually perpendicular planes, and are the algebraically largest and algebraically smallest normal stresses to be found at a paint in a given stress field.

According to this theory, yielding occurs when the Von Misses stress equals the yield stress. Experiments have shown that the distortion-energy theory predicts yield with the greatest accuracy amongst the accepted stress theories. The design criterion uses the Distortion Energy theory with a multiplier of 0.7 which accounts for probabilistic conditions such as variations in metallurgy, metal porosity, inclusions, and other uncertain conditions. This multiplier of 0.7 is slightly higher than the 0.6 to 0.66 multiplier used for welded structures. Thus the maximum acceptable Von Misses' stress in the shaft, end-disk and shell is (0.7 X yield stress of the component).

C. *Fatigue Strength Criteria*

1) *Shell*

In the case of most pulleys, the largest range stresses in the shell are usually in the tangential or hoop direction and occur close to the centreline of the pulley. Pulleys with wide shell faces may have the largest range stress in the axial direction due to bending in a region close to the shell/disk connection. The British Standard BS5400 Part 10 is used to determine the allowable stress ranges for the circumferential and seam welds in the shell for infinite fatigue life as shown in Figure 2.

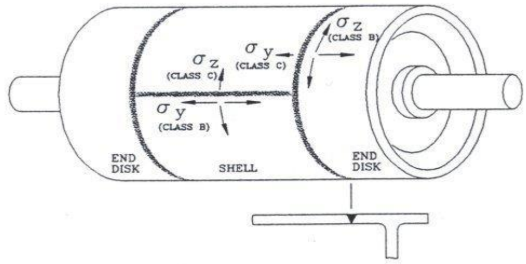

Figure 2. Circumferential and Axial Weld Classifications(9)

2) *Weld*

Shell Circumferential Welds have an allowable axial stress range of 77 MPa (11165 psi) (Class C weld) and allowable hoop stress range of 100 MPa (14500 psi) (Class B weld). These values apply if the welds are full penetration and have been ground flush and proven free of defects. If they are not ground flush and proven free of defects, the allowable axial stress range reduces to 55 Mpa (7975 psi) (Class D weld) and the hoop stress range to 77 MPa (11165 Ps) (Class C weld).

Shell Axial Seam Welds have an allowable axial stress range of 100 MPa (14500 psi) (Class B weld) and hoop stress range of 77 MPa (11165 psi) (Class C weld) if they are full penetration and have been ground flush and proven free of defects. If not, the allowable axial stress range reduces to 77 MPa (11165 psi) (Class C weld) and hoop stress range to 55 MPa (7975 psi) (Class D weld). These allowable stress ranges are for 10 million load cycles with a 97% confidence level. Radiographic and/or a full ultrasonic inspection must be performed to evaluate the welds.

3) *Disk*

For most pulleys, the largest fluctuating or range stresses in the disk are in the radial direction and are due to end-disk bending. The fatigue strength criteria used here is that the maximum stress should not exceed the endurance stress, Se, for infinite life. The endurance stress, Se is dependent on numerous factors including material type, surface finish, stress concentration effects, type of loading, failure mode, etc. A conservative endurance stress of 40% of yield stress (20% for shear) is used for ductile materials to account for the following possibilities, some of which are difficult to quantify:

Unlimited number of starts and stops
Dynamic loads
Irregularities in lagging thickness
Material buildup
Overloading of the conveyor

4) *Shaft*

As the pulley rotates the shaft contact pressure under the locking device changes at the inside and outside shoulders. The alternating stress introduced due to this can lead to fatigue failure if the range is large. Therefore limits are placed on how large this range stress can be this range stress should not exceed the limits imposed in the modified Goodman diagram.

V. NON-LINEAR STATIC STRUCTURAL ANALYSIS OF A DRUM PULLEY ASSEMBLY

Mathematically, the finite element method (FEM) is used for finding approximate solution of partial differential equations (PDE) as well as of integral equations. The solution approach is based either on eliminating the differential equation completely (steady state problems), or rendering the PDE into an equivalent ordinary differential equation, which is then solved using standard techniques such as finite differences, etc.

In solving partial differential equations, the primary challenge is to create an equation which approximates the equation to be studied, but which is numerically stable, meaning that errors in the input data and intermediate calculations do not accumulate and cause the resulting output to be meaningless. The Finite Element Method is a good choice for solving partial differential equations over complex domains or when the desired precision varies over the entire domain.

To perform an accurate analysis a structural engineer must determine such information as structural loads, geometry, support conditions, and materials properties. The results of such an analysis typically include support reactions, stresses and displacements. This information is then compared to criteria that indicate the conditions of failure. Advanced structural analysis may examine dynamic response, stability and non-linear behaviour.

Performing a Static Analysis

Following are the steps in brief to perform a static analysis:

1) *Build Geometry*
2) *Define Material Properties*
3) *Generate Mesh*
4) *Apply Loads*
5) *Obtain Solution*
6) *Present the Results*

B. Solid model details

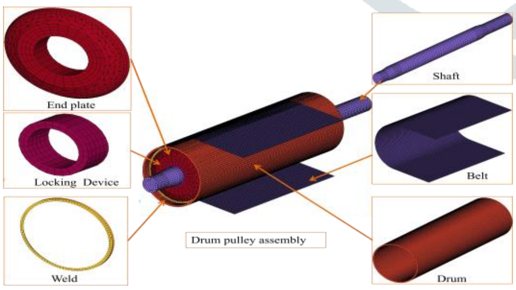

Figure 3. Exploded View of Meshed Drum Pulley Assembly

TABLE I. DIMENSIONS OF VARIOUS COMPONENTS

Sr. No.	Component	Dimension in mm				
		ID	OD	Length / Width	Thickness	Type
1.	Belt	-------	--------	1400	20	GRADE N
2.	Drum	598	630	2000	16	IS:2062 E250 B
3.	Weld	19 Face	5 root	35°	20	Single bevel E6013
4.	Endplate	260	588	---	Varying	ASTM 516 GR 70
5.	Locking Device	190	260	160	-------	BIKON 2006 190 X 260
6.	Shaft	-----	190 & 180	3005	--------	C45E+N / 080M40 BS:970

TABLE II. MATERIAL PROPERTIES OF COMPONENTS

Sr. No.	Component	E (GPa)	Poisson's Ratio	Syt (MPa)	Specification
1.	Belt	210	0.30	--	EP 800/4 GRADE N
2.	Drum	250	0.3	410	IS:2062 E250 B
3.	Weld	210	0.3	350	E6013
5.	Endplate	197	0.3	335	ASTM 516 GR 70
7.	Locking Device	210	0.3	410	BIKON 2006 190 X 260
9.	Shaft	210	0.3	340	C45E+N / 080M40 BS:970

C. Meshing

The process for generating a mesh of nodes and elements consists of three general steps:

- Set the element attributes.
- Set mesh controls (optional). Hyper mesh offers a large number of mesh controls from which you can choose as needs dictate.
- Meshing the model.

It is not always necessary to set mesh controls because the default mesh controls are appropriate for many models. Alternatively, you can use the Smart Size feature to produce a better quality free mesh.

Following are details of the elements used for meshing of given assembly.

a) *SOLID186 Element*
b) *SHELL181 Element*
c) *TARGE170 Element*
d) *CONTA174 Element*

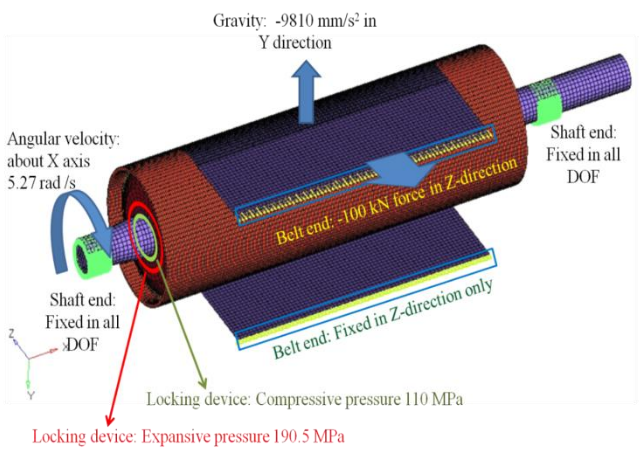

Figure 4. Boundary Conditions

D. Result Interpretation

Von misses stress = SEQV i.e. equivalent stress.

$$\sigma_v = \sqrt{\frac{[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_x - \sigma_z)^2]}{2}}$$

Where, σ_x, σ_y, σ_z are the corresponding stresses in X, Y and Z directions.

The von Mises Criterion, also known as the maximum distortion energy criterion, octahedral shear stress theory, or Maxwell-Huber-Hencky-von Mises theory, is often used to estimate the yield of ductile materials. The von Mises criterion states that failure occurs when the energy of distortion reaches the same energy for yield/failure in uniaxial tension. Mathematically, this is expressed as,

$$\frac{1}{2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right] < \sigma_y{}^2$$

This equation represents a principal stress ellipse as illustrated in the following Figure

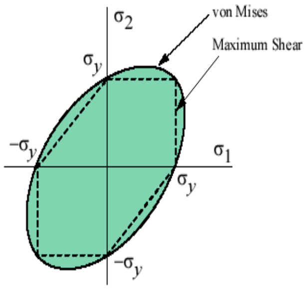

Figure 5. Von-Mises Criteria

Also shown on the Figure 5 is the maximum shear stress criterion (dashed line). This theory is more conservative than the von Mises criterion since it lies inside the Von Misses ellipse.

In addition to bounding the principal stresses to prevent ductile failure, the von Misses criterion also gives a reasonable estimation of fatigue failure, especially in cases of repeated tensile and tensile-shear loading.

E. Result Plots

1) Deformation Plot

Following figures are the Total Deformation Plots for different parts of the Drum Pulley Assembly.

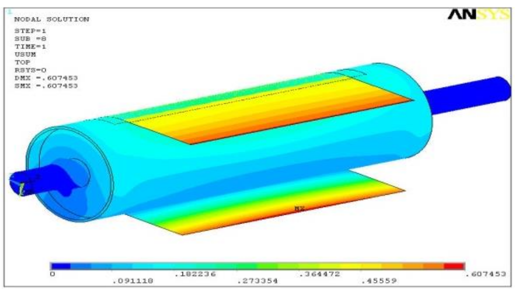

Figure 6. Total Displacement of assembly

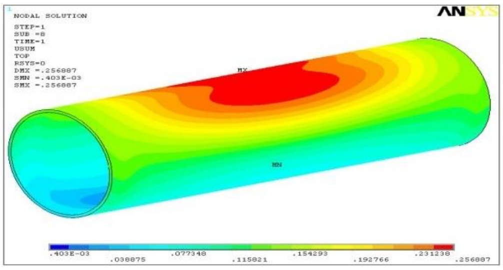

Figure 7. Total Displacement of Drum

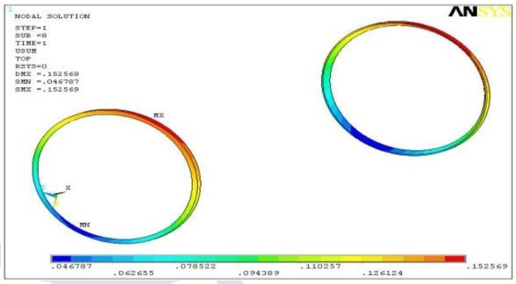

Figure 8. Total Displacement of Weld

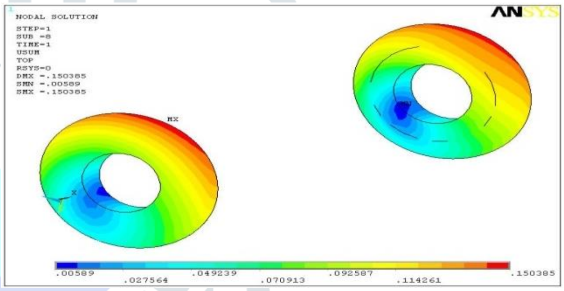

Figure 9. Displacement Plot of End disks

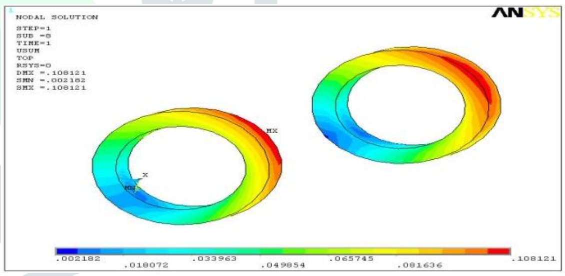

Figure 10. Displacement Plot of Locking Device

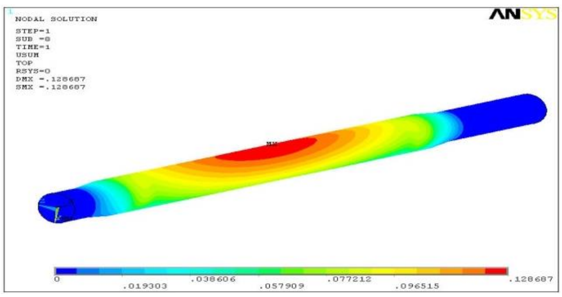

Figure 11. Displacement Plot of Shaft

2) Equivalent (Von-Misses) stress plot

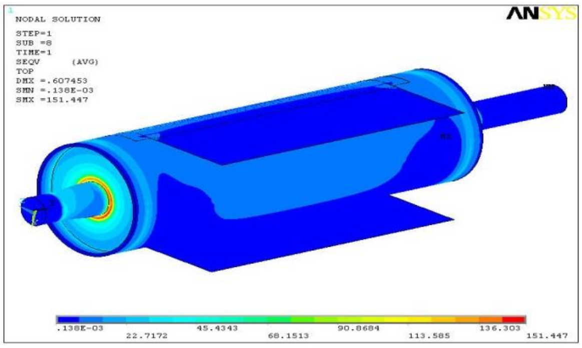

Figure 12. Stress Distribution for the total Assembly

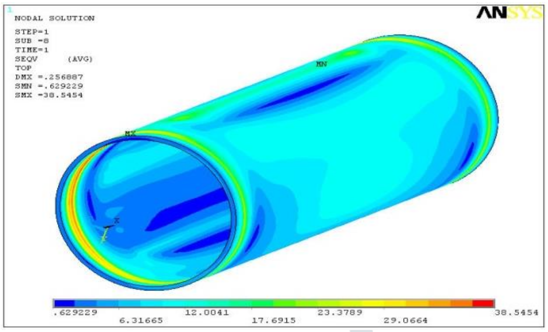

Figure 13. : Stress Distribution in Drum

Figure 14. Stress Distribution in Weld

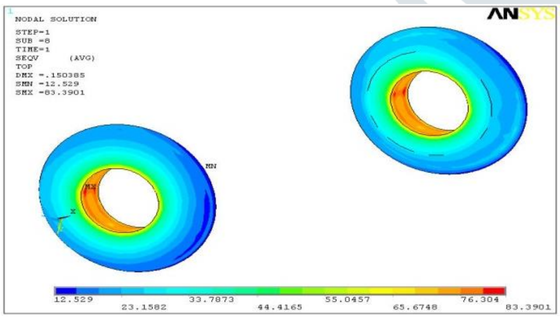

Figure 15. Stress Distribution in End Disks

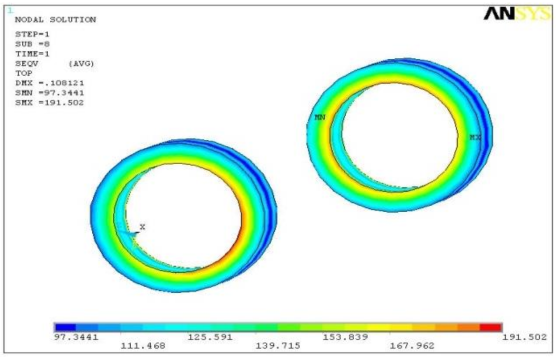

Figure 16. Stress Distribution in locking device

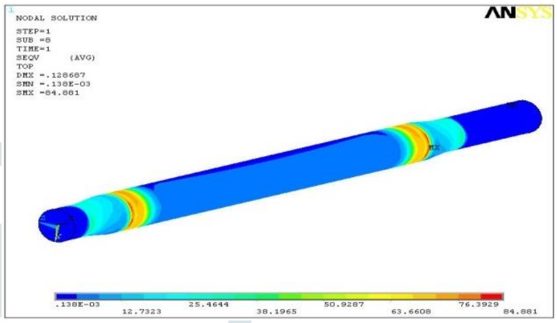

Figure 17. Stress Distribution in Shaft

F. Result Table

Result table is prepared with the current design criteria for static strength and fatigue strength as discussed. On the basis of these criteria safe or failure limit is also discussed here.

Static strength criteria = 0.7 x S_{yt}

Static strength criteria (for weld) = 0.6 x S_{yt}

A fatigue strength criterion is given for life of 1 E +06 cycles with 97% confidence level.

TABLE III. NON-LINEAR STATIC STRUCTURAL ANALYSIS RESULT TABLE

Component Name	Von-mises Stress (MPa)	Allowable Stress (MPa)		Factor of Safety		Remark
		Static Criteria	Fatigue Criteria	Static	Fatigue	
Drum	38.545	287	143.5	7.45	3.72	Safe
Weld	**46.919**	**210**	**55**	**4.48**	**1.17**	**Safe**
Endplate	83.390	234.5	134	2.81	1.61	Safe
Locking Device	191.502	300	280	1.57	1.46	Safe
Shaft	84.881	238	143.5	2.80	1.69	Safe

VI. CONCLUSIONS

1. This work succeeds to predict the fatigue failure of drum pulley assembly, with the analytical background of concepts, its analogy in simulation software and analytical calculations to validate the concept of analysis.
2. The drum pulley assembly has complex type of loading hence it is impossible to obtain analytical

solution. Also, there are many limitations to perform experimental analysis like cost of an assembly, cost of experimental setup, time required for fatigue testing and hence overall testing cost required is very high. Hence only simulation was done which is less expensive and more accurate.

3. At the end it is concluded that the design is strong enough to sustain 0.5 millions cycle for operating loading conditions with various cracks. The failure cause can be stated as the bad quality of weld material, improper welding, occurrences of multiple cracks, overloading, improper surface preparation, too much corrosive environment variables may have amplified the stress intensity by 20 times. At this amplified stress intensity weld component has failed to survive 0.5 millions cycles.
4. This simulation concept is not yet thoroughly implemented in industry as well as in academics because the simulation software's like ANSYS which are designed and developed on fatigue failure concepts are in initial stage of development. But definitely this will be the future asset of fatigue prediction for all FEA engineers.

REFERENCES

[1]G.R.Jinu, P.sathiya and G.Ravichandran, 2009, Investigation of the Fatigue Behavior of Butt-Welded Joints Treated by Ultrasonic Peening Process and Compared with Fatigue Life Assessment Standards, International Journal of Advance Manufacturing Technology, Vol. 40, pp 74–83

[2]M. Zadeh, Aidy Ali and B.B.Sahari, 2009, Three Dimensional Simulation of Fatigue Crack Growth in Friction Stir Welded Joints of 2024-T351 Al Alloy, International Journal of Scientific and Industrial Research, Vol. 68, pp 775 - 782.

[3]Martinsson M., Samuelsson J., 2002, Design and analysis of welded high strength steel structures, EMAS, Stockholm, pp 303-334.

[4]NiklasKarlsson, 2005, Analysis of Fatigue Life in Two Weld Class Systems, Department of Mechanical Engineering Linkoping University, Linkoping, Sweden.

[5]P.JohanSingh, B.Guha, 2001, Fatigue tests and estimation of crack initiation and propagation lives in AISI 304L butt-welds with reinforcement intact, Journal of Theoretical and Applied Fracture Mechanics, Vol. 14, pp60-67.

[6]Pettersson G., 2002, Design and analysis of welded high strength steel structures, EMAS, Stockholm, Vol. 1, pp 413-436.

[7] Terry J King, 1983, The Function and Mechanism of Conveyor Pulley Drums, Beltcon 2nd International Conference, Johannesburg.

[8]V.Balasubramanian, B.Guha, 2000, Fatigue Life Prediction of Welded Cruciform Joints using Strain Energy Density Factor Approach, International Journal for Theoretical and Applied Fracture Mechanics, Vol. 34, pp 85-92.

[9]Vinit Sethi, Lawrence Nordell, 1991, Modern Pulley Design Techniques and Failure analysis methods, Department of Research and Development, Conveyor Dynamics Inc, Bellingham, WA.

[10]D.Radaj, C.M.Sonsino and W.Fricke, 2006, Fatigue Assessment of Welded Joints by Local Approaches, Institute of Materials, Minerals and Mining, CRC Press, Boca Raton, New York, Washington DC.

[11]Yukitaka Murkami, 2006, Metal Fatigue: Effect of Small Defects and Non-Metallic Inclusions, Elsevier Science Ltd, Oxford, UK.

[12]ASTM standards E399 and E647

[13]AWS B1.10-1996: Guide for the Nondestructive Inspection of Welds

[14]AWS D1.1-2008: Structural Welding Code for steel

[15]BS 7448: Part 1: 1991 Method for determination of Kic, Critical CTOD and Critical J values of metallic material.

[16]BS 7608: 1993: Fatigue Design and Assessment of Steel Structures.

[17]ANSYS Theory and Manual Version 12.1