

Prediction of Fatigue Life Cycle on Steering Knuckle Ball Joint

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ABSTRACT: Automobile field is on the rise dynamically and their sub-industries. Life of vital component is concern of nowadays. One of the vital components in steering and suspension systems is the knuckle ball joints. The Ball joint has three rotational DOFs. This joint is used to connect the knuckle to the steering rod and the short and long arm. Ball joints allow movement of steering components and suspension Ball joints are a vital element to a car. They are the vital part of the vehicle's framework that connects the steering knuckles to the control arms next to the wheels. The purpose of the project is to modify the design of a steering knuckle to amplify its life cycle. Existing component is tested for its fatigue strength experimentally by FEA software and the prediction of the cycle is done.

KEYWORDS: Steering, Vital, Fatigue life, Knuckle ball joint, FEA.

I. INTRODUCTION

Manufacturing processes faces most crucial competitions in automotive industry to produce lighter, cheaper and more efficient components that exhibit more precise dimensions, needless machining and require less part processing. Material mechanical properties and manufacturing parameters play decisive roles and the weaknesses and strengths of each manufacturing process need to be available to designers in these respects, to enable them to choose the optimum choice for the specific component and application. Steel and aluminum forgings and castings, cast irons, and powder forgings have found expansive applications in automotive safety-critical systems. Many safety-critical components in the vehicle experience static as well as time varying loadings, and obviously they undergo the latter during a major portion of their service life. Fatigue behavior is, therefore, a key consideration which is to be considered during product design stage itself. In automotive design, durability estimation of such components based on experimental assessments is time-consuming and also expensive, so analytical approaches that include limited number of component verification tests have been used now-a-days. A problem that arises at the fatigue design stage of such components is the transferability of data from smooth specimens to the component. The component geometry, loading, and manufacturing process parameters such as surface conditions often deviate from that of the specimen investigated and neither a nominal stress nor a notch factor can be defined in most cases. The studies on optimization have focused on three aspects; geometry optimization, alternative materials and manufacturing process modifications. From the viewpoint of geometry and considering the boundary conditions, an optimized component could be obtained using different methods of geometry optimization. On the other hand, alternative materials are being put into trial and manufacturing processes are being re-evaluated to achieve lighter, cheaper and more efficient components.

II. PROBLEM STATEMENT

One of the important structural limitations is fatigue properties. This study is aimed at the Automotive industry, more specifically a SAE 4135 ball joint in the suspension system, where quality is of great concern most of the time to failure consists of crack initiation and a conservative approach is to denote the component as failed when a crack has initiated. This simplification allows designers to use linear elastic stress results obtained from multibody static and dynamic FE

(finite element) simulations for fatigue life analysis. The ball joints subjected to cyclic loading and it is consequently prone to fatigue damage. The best material that will be used for the manufacturing process of the component is important to predict the life.

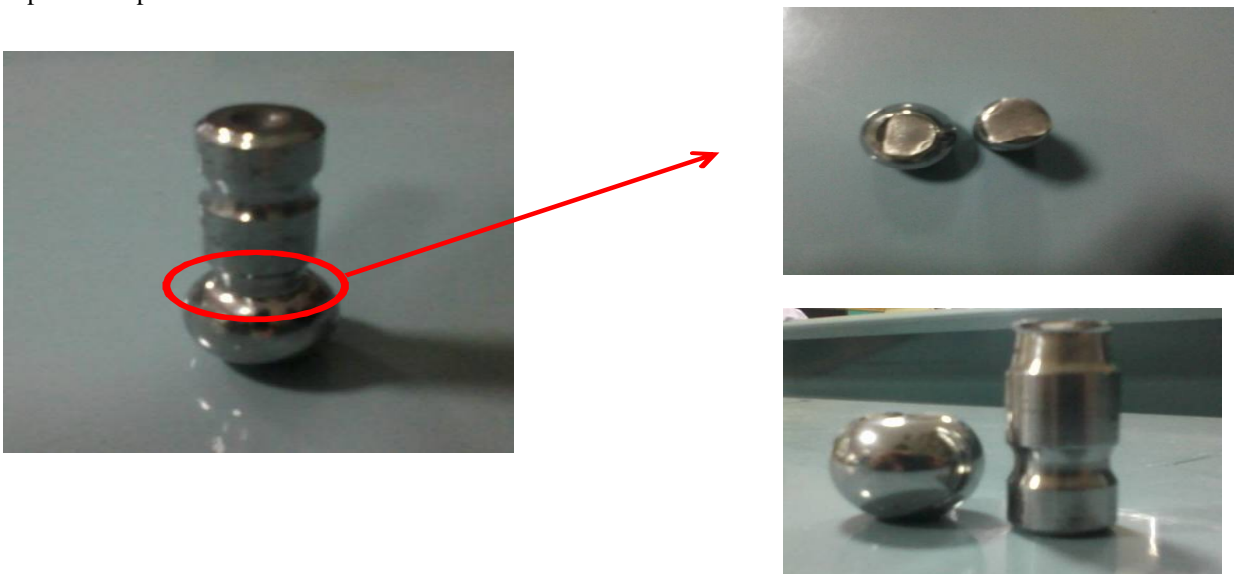


Fig 1 Image showing the failure components

III. FINITE ELEMENT ANALYSIS

The objective of the stress analysis was to obtain the complete three dimensional stress and strain distributions at a potential failure site, facilitating fatigue life predictions. Depending upon the method used for prediction of fatigue life, stress analysis can be linear or nonlinear. Linear elastic analysis is the most common type of stress analysis pursued in automotive design and analysis. The majority of automobile engine components like camshaft, crankshaft and connecting rod operate within the material elastic region, and therefore linear elastic analysis is feasible to model their behavior. On the other hand, most components in chassis and suspension undergo occasional overloads in service and cyclic plasticity becomes a major factor to define their stress-strain response, particularly at stress concentrations. In these cases, linear analysis is not sufficient to analyze component's behavior and nonlinear elastic-plastic analysis should be performed.

III.1 Ball Joint Modelling

The modeling is done using Solid works 10.0 software package. The finite element model is done using Ansys 14.0 software package is shown in fig 2. To generate the mesh for the components, free meshing feature of the software was employed since it has no geometry restrictions and it could be defined on complicated volumes. The free mesh generator uses an algorithm that minimizes element distortion. Three-dimensional linear tetrahedron solid elements with three translational degrees of freedom were used. Although parabolic element offers more accurate results, the analysis time, especially for nonlinear analysis, is greatly more than that of linear elements. Nonlinear analysis at the highest moment level with parabolic elements showed negligible difference in results with those using linear tetrahedral element and, therefore, the linear element was selected for modeling. For the complex three-dimensional geometry of the steering knuckles, solid element offers more accurate results and Ansys software employs tetrahedron solid elements.

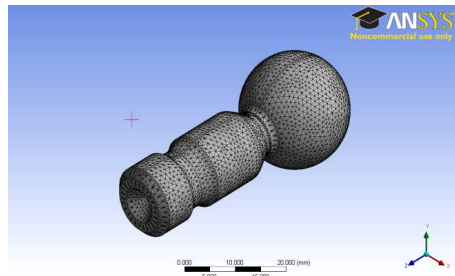


Fig-2 Finite element model of steering knuckle ball joint

III.2 Model Boundary Conditions

The boundary conditions and loading were selected to represent the assumed component testing conditions as shown in Figure 7. To verify the model with the specified boundary conditions other alternatives were analyzed by switching the loading and boundary conditions and also by releasing any one of the fixed points to ensure the critical locations remained the same

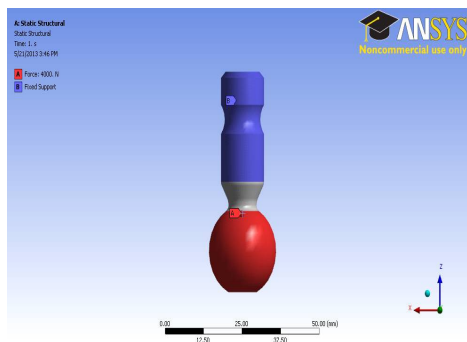


Fig-3 Boundary Condition

IV. FATIGUE LIFE PREDICTIONS AND COMPARISONS WITH EXPERIMENTAL RESULTS

Plastic In general, four fatigue life prediction models are commonly used. These include the stress-life model, the strain-life model. The first two models consider the macro-crack nucleation phase as failure. The stress-based approach is typically used in terms of nominal stresses. Therefore, it does not directly account for the plastic strain at the un-notched, which can have significant effect on fatigue behavior of components under cyclic plasticity conditions. The predictions of this approach for un-notched members of the forged steel steering knuckle ball joint are usually conservative. Local values of stress could also be used in stress-life predictions (here called local stress-life or σ -N method). The axial fatigue tests were performed at room temperature and under constant amplitude load control. A sinusoidal wave form was used in all the tests. Loading frequency varied from 5 to 15 Hz with a lower value used for high load levels and a higher value for the low load levels. Each test was performed under a constant load frequency condition. The fatigue tests were conducted at three load levels for forged steel, resulting in lives between 10^6 cycles. Three tests were conducted at each load level, to assess variability and scatter. All the ball joints were tested at a load ratio of $R = -1.25$. This ratio was obtained from FEA at the highly stressed locations in the transition regions. Prior to the testing, all the connecting rods were numbered, weighted, and diameters were measured. The loading conditions, area at fracture, cycles to failure and failure locations for all the tested ball joints. The fatigue life of the ball joints was defined as the number of cycles endured until it completely separated into two pieces. The forged steel joints mainly failed in the transition to the neck region. In the following sections the life prediction paths pursued are implemented and discussed. The details of fatigue life calculations for each method at the highest moment level for the forged steel steering knuckles ball joint are provided. The table 1 shows that the comparison between the experimental and software results.

EXPERIMENTAL RESULTS OF SPECIMEN

Constant amplitude completely reversed fatigue test results								
Specimen ID	Test control mode	Load (KN)	Test control Freq (± 5) HZ	At midlife ($N_{50\%}$)		No.of.cycles N_f	Failure Location [b]	E Gpa
				$\Delta\epsilon/2$, %	$\Delta\epsilon_p/2$ [a] % (calculated),			
Ball10-1	Strain	4	5	0.56	0.55	2.514E+05	NECK	190
Ball10-2	Strain	4	5	0.79	0.78	2.544E+05	NECK	190
Ball10-3	Strain	4	5	0.625	0.62	2.471E+05	NECK	190
Ball10-4	Strain	4	5	0.6	0.6	2.504E+05	NECK	190
Ball10-5	Strain	4	5	0.605	0.605	2.533E+05	NECK	190

[a] $\Delta\epsilon_p/2(\text{calculated})=\Delta\epsilon/2-\Delta\sigma/2E$. [b]Neck region NECK
Table 1-Component test data of forged steel steering knuckles ball joint Investigated.

V.SOFTWARE RESULTS OF DESIGN

V.1 Structural and Fatigue Analysis for Modified Design (Increasing the ϕ & R)

Structural and Fatigue analysis for Ball Joint

Case study a) - Existing Design Vs Modified Design

- Stress analysis
- Deformation for the applied load
- Factor of safety
- Life
- Damage
- Safety factor
- Biaxiality indications

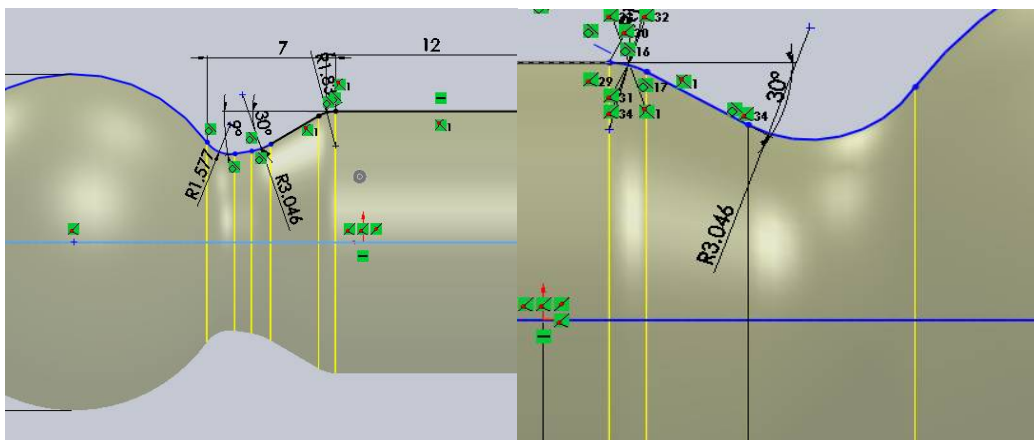


Fig-4 Existing neck region

Fig-5 Modified neck region

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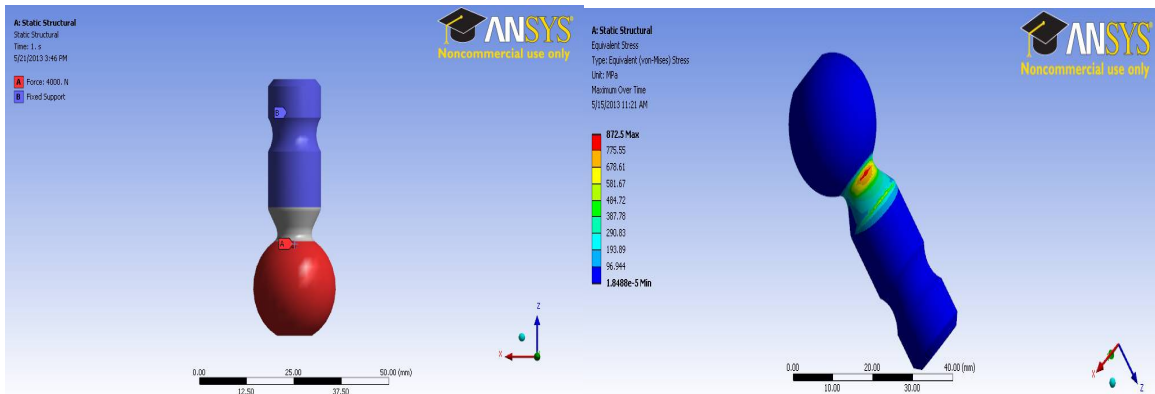


Fig-6 Loading condition Fig-7 Stress Distribution

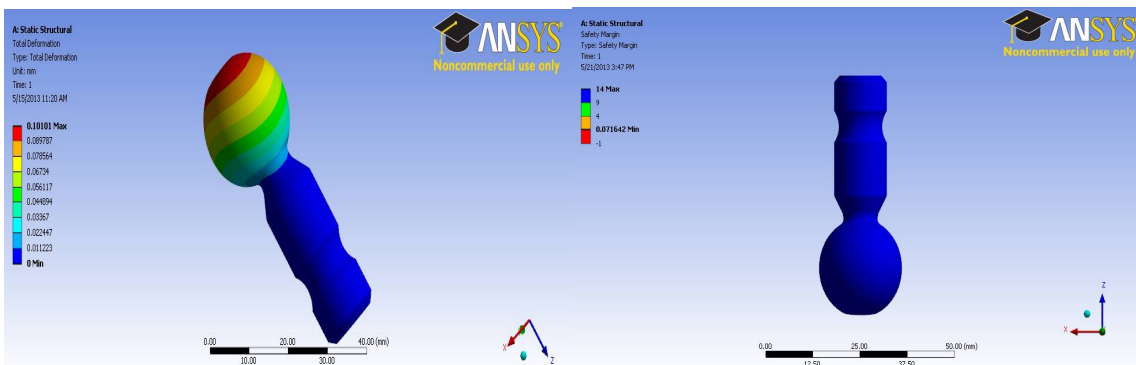


Fig-8 Total Deformation Fig-9 Safety margin

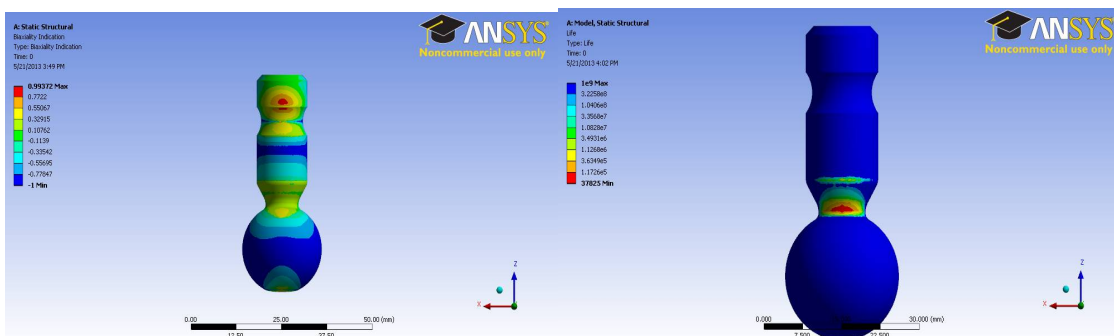


Fig-10 Biaxiality Indication Fig-11 Life Cycle

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Radius (R)	Angle (°)	Von-Mises Stress (Mpa)		Total deformation (mm)		S-N	ε-N		Loading Ratio
		Min	Max	Min	Max	Fatigue Life (Gerber)	Marrow-Fatigue life	SWT - Fatigue life	
R3	30	0	918.63	0	0.01065	8553	1.87E+05	2.91E+05	0.3
R3	31	0	611.87	0	0.064	35114	3.65E+06	3.89E+06	0.3
R3	32	0.0003	621.87	0	0.0644	33233	3.24E+06	3.47E+06	0.3
R3	33	7.55E-05	599.37	0	0.0683	37876	4.22E+06	4.46E+06	0.3
R3	35	0.00013	609.41	0	0.069	35635	3.68E+06	3.92E+06	0.3
R3	36	0.0002	604.26	0	0.0694	36763	3.94E+06	4.18E+06	0.3
R3	37	0.00015	608.39	0	0.0694	35855	3.73E+06	3.97E+06	0.3
R3	38	0.00015	865.2	0	0.106	10326	2.28E+05	3.21E+05	0.3
		Fatigue Strength factor -1		Type-full Reserved					

Table-2 - Fatigue life for σ-N and ε-N approaches were both obtained

Case study 2- Experimental Results Vs Software result

Specimen ID	Life	
	Experimental	Software Result
Ball 1-1	2.514E+05	4.222E+06
Ball 1-2	2.544E+05	4.222E+06
Ball 1-3	2.471E+05	4.222E+06
Ball 1-4	2.504E+05	4.222E+06
Ball 1-5	2.533E+05	4.222E+06

Table-3 Fatigue Life Predictions and Comparisons with Experimental Results

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