FATIGUE ANALYSIS FOR A STEERING KNUCKLE OF A BAJA SAE VEHICLE

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Abstract. Fatigue in automotive components is of great interest for both automakers and consumers. Due to high costs and time involved in experiments and tests of these vehicles, engineers often resort to analytical and computational models during the design project phase. This work presents the fatigue analysis and expected lifetime calculus for a steering knuckle used in an offroad BAJA vehicle. Different models were tested to choose one which could most accurately represent known vehicle behavior, as experimentally verified by the Tchê Baja team. Using loads obtained through a ride simulation in MSC Adams/Car, critical fatigue points were determined for the component analysed. Cycle-counting was done through the rainflow algorithm, and correction for mean stresses through the FKM directive and Goodman methods. The results show the differences in loads in both critical points, as well as differences in the mean stress correction methods.

Keywords: High-cycle fatigue, offroad vehicles, systems modelling.

1 Introduction

Fatigue in automotive components is one of the main concerns for engineers during the design and maintenance of these products, and is a widely studied topic in mechanical engineering, with several works concerning methodology (such as Takahashi [1], McKelvey, Lee and Barkey [2] or Miranda [3]) or practical applications (Lee, Han and Kim [4], for instance). Despite the large number of articles for standard road vehicles, the literature is rather scarce when it comes to off road vehicles. One example of a class of such off road vehicles is the competition vehicle used in BAJA SAE competitions; in such competitions, not only are these vehicles subjected to dirt roads but also obstacles, such as ramps, logs and rocks on the track.

In this article, we seek to apply a predictive fatigue analysis for the lifetime estimation of a frontal steering knuckle of a BAJA vehicle. Section 2 presents the theoretical foundations for this work, while section 3 contains a description of the applied methodology to model the steering knuckle and its loads. Finally, section 4 presents the results and section 5 the conclusions obtained in this work.

2 Theoretical Basis

2.1 Road Profiles and Signals

Despite the many advances of mechanical systems modeling, the best and most accurate way to obtain the load histories for different components is still through experimental methods. For vehicular systems, this is usually done through either proving grounds, on which the vehicle is subjected to extreme conditions, or through in-lab test rigs. The high degree of accuracy these methods provide for vehicle analysis is however offset by their high cost and time expense. As such, computational methods are usually more suitable for the preliminary analysis which occurs during the design phase of new products, and their low cost is also attractive for small engineering projects, such as those undertaken by student competitions.

For on-road vehicles, ride analyses are usually based on either displacement or velocity power spectral densities (henceforth PSDs) representative of a road class, ranked from A to H (from least to most severe) as prescribed in ISO 8608 [5]. As these PSDs were created based on conditions for paved roads, it would be incorrect to say that they represent adequately the profile of unpaved or faulty roads. However, considering that the BAJA vehicle analyzed in this work rides not only on very rough terrain, but must also traverse obstacles (such as jumps, rocks, potholes and others), and due to our lack of experimental data, we have decided to use the H class of displacement PSDs to model the road conditions faced by this vehicle during competition.

2.2 High-Cycle Fatigue Analysis of Aluminum Alloys

Fatigue analysis for mechanical components usually requires a very large quantity of experimental data in order to determine the service life of any material under different conditions and varied loads. Naturally, as it is impractical to construct Wöhler curves for a large number of geometry/material/load combinations, most investigations in this area of research focus on obtaining curves for fully-reversed loading conditions. According to Norton [6], different conditions are accounted for through multipliers (for instance, notch fatigue multipliers) which adequate the material curve to the analyzed component.

It is well-known that aluminum alloys do not have an endurance limit, which means that there is no stress small enough below which infinite life is guaranteed. With this in mind, mechanical engineers usually design aluminum components for a set, economically viable finite lifetime. Considering that the analyzed steering knuckle is made from machined Aluminum 7021 without any heat treatments, we have researched the available literature for S-N curves, with no avail. As such, we have resorted to using the untreated data for the 7020 alloy, presented in Bloem et al. [7], which we considered to be "close enough" to our alloy. Figure 1 presents the S-N curve for AA 7020, plotted from the exponential-regression model presented in their paper, alongside the relevant material properties for untreated AA 7021 in Table 1, as sourced from the supplier and used during the design of the steering knuckle.

Table 1. Material properties of the aluminum 7021 alloy used in the steering knuckle and in this work.

Material Properties	Young's Modulus E [GPa]	Poisson's Ratio v	Yield Strength S_y [MPa]	Ultimate Tensile Strength S_{uts} [MPa]
Value	71	0.3	315	350

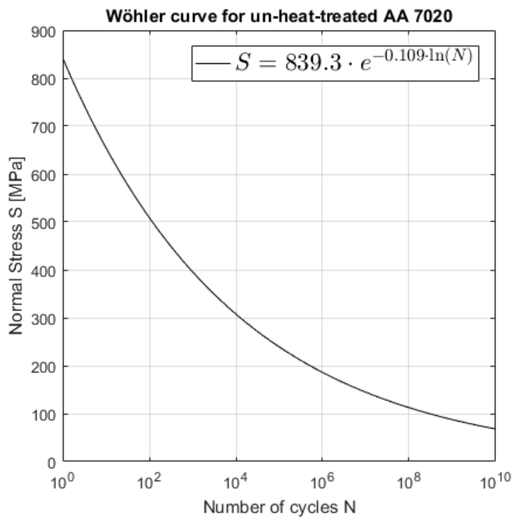

Figure 1. Wöhler curve for AA7020. The exponential characteristic of this S-N curve is representative of aluminum alloys.

In order to use such diagrams, it is necessary in first place to obtain the stress-time histories from the input load-time histories, which can be done in the case of linear elasticity through simple linear algebra. After obtaining the time-dependent stress tensor, it is then necessary to determine whether the analyzed critical point is subjected to a uniaxial or a multiaxial stress state. For the uniaxial case, it then becomes a matter of determining what should be the fatigue effective stress; according to Pedersen [8], common examples are the maximum shear stress, principal stress and the signed von Mises stress, which will be used here due to its simplicity.

After obtaining the time-history of the fatigue effective stress, the next step in fatigue analysis becomes to count the total number of equivalent fully-reversed stress cycles and their amplitudes, which can be done in a multitude of ways. The ASTM Standard E1049-85 [9] compiles different acceptable methods for this, such as level-crossing or peaks counting, and among these is the widely-used Rainflow method, which was used on this work wide range of applications and ease of use. After the total number of (half-)cycles is extracted, the next step is to realize the mean stress correction of the obtained cycles, which can be done through various different methods. In this work, we have chosen to use both the well-known Goodman and the FKM-Guideline [10] methods for mean stress correction. Finally, through the well-known Palmgren-Miner rule for constant damage, we can estimate the total life of the analyzed component.

3 Mechanical System Modelling

The steering knuckle analyzed in this work is machined from un-heat-treated Aluminum 7021, set up on a double wishbone suspension and connected to both the wishbones and steering rod through universal joints. The support for the steering joint is a separate piece machined from SAE 4340 steel, connected by a bolt to the steering knuckle. Figure (2), below, shows the steering knuckle (light blue) and the support (grey) on the left, while the right part of the figure presents the suspension assembly connected to the steering knuckle.

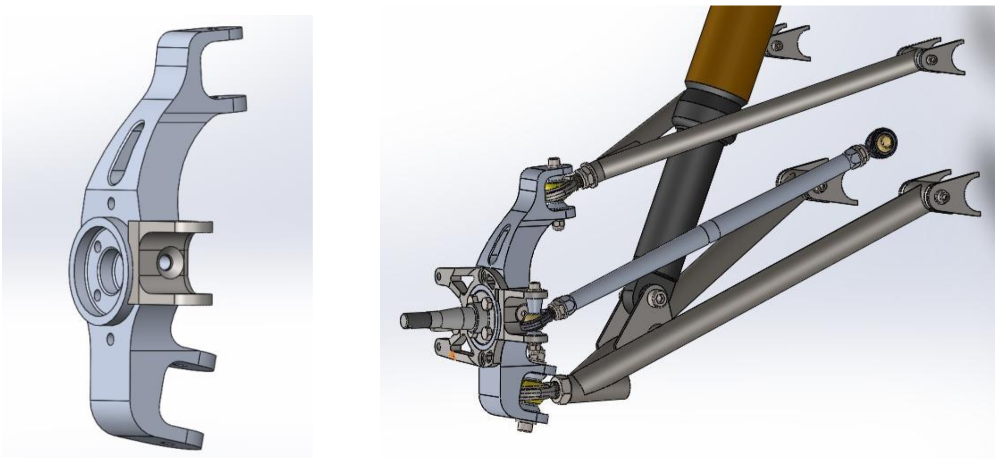

Figure 2. On the left: steering knuckle with support. On the right: frontal-right suspension assembly.

In order to simplify the fatigue and finite element analyses, we have decided to model the steering knuckle by merging the knuckle itself with the steering support as if it were a single component; it is expected that such an idealization results in more severe fatigue, as it imposes a hard constraint of no relative movement between these parts. To determine the critical points, two different models were tested. The first model considered the steering knuckle as clamped on the hub and subjected to point loads acting on the three universal joints, which were obtained from a rigid body analysis in MSC Adams/Car for 5 seconds of ride. The second model accounted for the entire quarter-car suspension assembly, subjected to gravity loads and connected to a spring (representing the wheel's vertical stiffness) subjected to a vertical displacement loading, acting on the stub axle.

By running these models through the same track, we found that the first model predicted more accurately the stress state and loads faced by the steering knuckle and which was observed by the Tchê BAJA team. Furthermore, we determined the positions of the critical points for this component, which lay on the 0,4 mm radius around the hub (point "A")and on the inner side of the steering knuckle (point "B"). These points are illustrated on the FE mesh on the left and right sides of Figure (3), respectively.

With this in mind, we then ran a 120 ride analysis on the H class PSD track at a constant vehicle speed of 50 km/h, with sampling rate of 200 Hz and measured the reaction forces acting on the steering knuckle. After that, we ran a total of 9 different analyses with unit loads (3 force components on each joint) to determine a force-to-stress "transfer matrix" for each critical point. By multiplying said matrix with the load history for each force component we then calculated the stress history for both critical points, without the burden of re-running an entire FE analysis for each time-step. With these stresses, we then calculated the equivalent von Mises stress history and finally, using the Rainflow cycle counting method and Palmgren-Miner damage model we obtained the expected fatigue life for each point, considering either Goodman or FKM mean stress correction. Figure (4) shows the obtained Rainflow cycle components for both points.

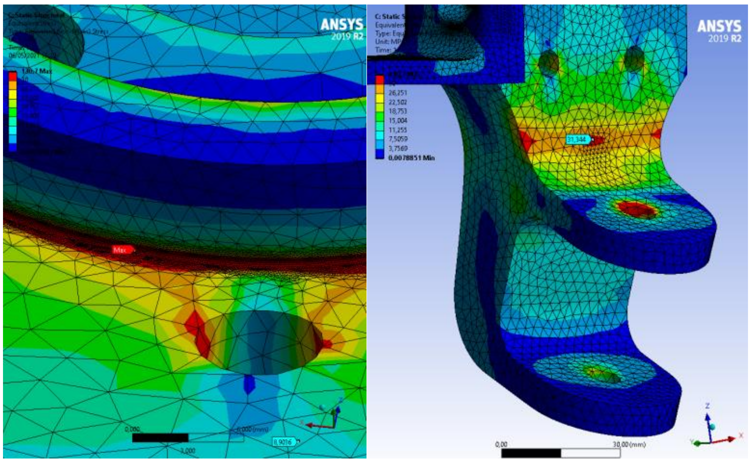

Figure 3. Detected critical points for the steering knuckle. Point "A" on the left, point "B" on the right. Mesh refinement:

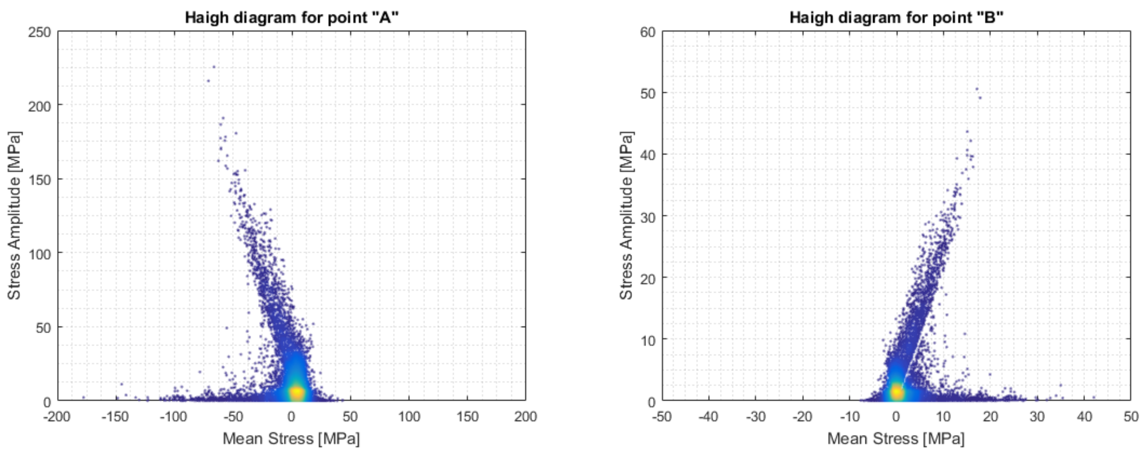

Figure 4. Scatter plots of load cycles on Haigh diagrams for points "A" and "B", colored by density (brighter colors indicate regions with a higher number of Rainflow cycles).

4 Results

Following the FKM-Guideline, we have considered the damage limit for the component as 0.3 (as it is made out of a cast Aluminum alloy), and considered the calculated damage as representative of the conditions faced by the vehicle during competition. With a confidence of 97.5%, Table 2 presents the fatigue life for both points as calculated by both criteria, as well as damage per second and total number of cycles until failure.

Table 2. Damage per second, number of cycles until failure and fatigue life, as calculated through Goodman and FKM mean stress correction methods for both points.

Result\Analysis	Point A: FKM	Point A: Goodman	Point B: FKM	Point B: Goodman
Damage per second[1/s]	5.1964E-7	2.0354E-6	1.2833E-10	3.2273E-11
Number of cycles until failure	4810.991	1228.289	19481405.838	77464669.547
Fatigue Life [hours]	160.366	40.943	649380.194	2582155.651

5 Conclusions

As the results show, it is possible to determine that point A, despite bearing mostly compressive stresses, will fail first due to the high intensity of these stresses; meanwhile, despite the presence of tractive mean stresses, Point B likely will not be a crack initiation point for this component. This is because stresses on point A are of overall much higher intensity when compared to point B stresses, such that the benefits of compressive stresses are not enough to offset their high intensity.

One interesting thing to note is that the FKM four-region mean stress correction results in a longer fatigue life for point A, but a shorter life for point B; this can be explained due to the distribution of the mean stresses on these points, as can be seen on Figure 4. For point A, as the high-intensity stresses lie between $R=-\infty$ and $R=-1$, the FKM method results in a longer fatigue life, in contrast to the Goodman method, which doesn’t consider compressive stresses as beneficial. For point B, the opposite happens; this is due to the AA7021 mechanical properties, which result in a higher slope for the FKM method, and therefore a shorter fatigue life when compared to the Goodman method. In either case, even for the most severe and conservative approach, the results show that the steering knuckle is more than resistant enough to stand the loads faced during one competition cycle, and as such is good enough to be used in different years.

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