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Stress Distribution Analysis of Rear Axle Housing by using Finite Elements Analysis

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--ABSTRACT--

A premature failure that occurred due to the higher loading capacity of the heavy vehicle is studied. To determine the reason of the failure, a CAD model of the housing was developed. The mechanical properties of the housing material were determined from the manufacturer manual. By using the given data, stress distribution analysis performed by using finite element software and fatigue life are predicted. Design enhancement solutions were proposed to increase the fatigue life of the housing.

Keywords *- Rear axle housing, Stress Distribution, Finite Element Analysis and Fatigue.*

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I. INTRODUCTION

The automobile was considered merely a mode of transportation. Today, the automobile is considered more and more as one of life's necessities. Many countries around the world have immersed interest and investment in the automobile industry which affects their industries a great deal. Also, they are focusing on the development of vehicles with improve performance, increased stability, and enhanced driver pleasure [1]. A premature failure that occurs prior to the expected load cycles during life of rear axle housing as shown in figure 1 is studied. Due to their higher loading capacity, solid axles are typically used in the heavy commercial vehicles [2]. During the service life, dynamic forces caused by the road surface roughness produce dynamic stresses and these force lead to fatigue failure of axle housing, which is the main load carrying part of the assembly. Therefore it is vital that the axle housing resist against the fatigue failure for a predicted service life. In automobiles, axle shafts are used to connect wheel and differential at their ends for the purpose of transmitting power and rotational motion. In operation, axle shafts are generally subjected to torsional stress and bending stress due to self-weight or weights of components or possible misalignment between journal bearings. Thus, these rotating components are susceptible to fatigue by the nature of their operation and the fatigue failures are generally of the torsional, rotating-bending, and reversed (two-way) bending type [3].

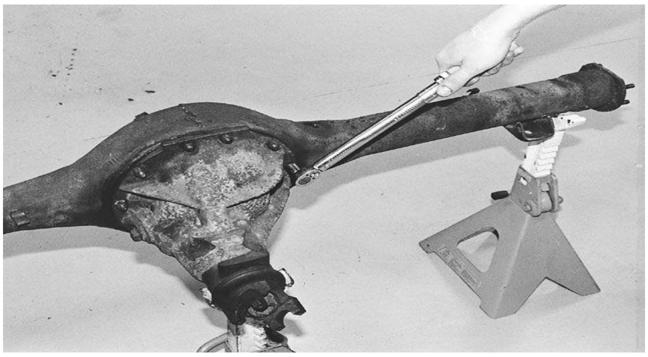


Figure 1: Rear axle housing

II. LITERATURE REVIEW

A. Introduction

Stress distribution analysis of rear axle housing is analysis to determine the stress distribute around the axle housing after load applied to the housing. In order to determine the location of the stress distribute or the failure point on the axle housing, the axle housing dimension was measured and convert to CAD software and this design was imported to FE analysis software to analyze the housing by given loading on it. The loading point is referring to the real loading condition on the vehicle and other condition of axle housing is considered to reach this aim.

B. Stress Concentration

In reality, bars often have holes, grooves, notches, keyways, shoulders, threads, or other abrupt changes in geometry that create a disruption in the otherwise uniform stress pattern. These discontinuities in geometry cause high stresses in very small regions of the bars, and these high stresses are known as stress concentration which is shown in figure 2. The discontinuities themselves are known as stress raisers [4].

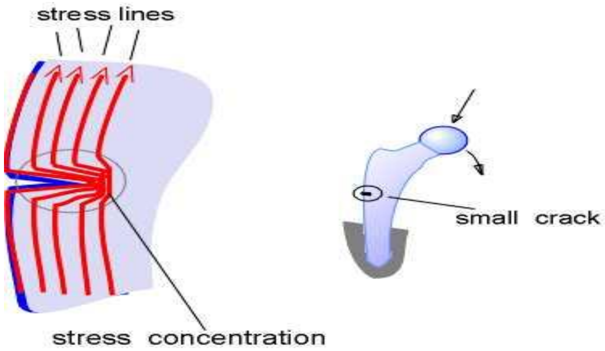


Figure 2: Stress Concentration Region

An object is strongest when force is evenly distributed over its area, so a reduction in area, e.g. caused by a crack, results in a localized increase in stress. The crack was occurred due to overload load carried by the vehicle. Which is the vehicle carry a load more than the axle can support. A material can fail, via a propagating crack, when a concentrated stress exceeds the material's theoretical cohesive strength. The real fracture strength of a material is always lower than the theoretical value because most materials contain small cracks that concentrate stress. Fatigue cracks always start at stress raisers, so removing such defects increases the fatigue strength. In this test, crack originated at the high stress concentration region.

C. Fatigue Failure

Fatigue is the progressive and localized structural damage that occurs when a material is subjected to cyclic loading [5]. The nominal maximum stress values are less than the ultimate tensile stress limit, and may be below the yield stress limit of the material. Fatigue occurs when a material is subjected to repeat loading and unloading. If the loads are above a certain threshold, microscopic cracks will begin to form at the surface. Eventually a crack will reach a critical size, and the structure will suddenly fracture [6]. The shape of the structure will significantly affect the fatigue life; square holes or sharp corners will lead to elevated local stresses where fatigue cracks can initiate as shown in figure 3. Round holes and smooth transitions or fillets are therefore important to increase the fatigue strength of the structure.

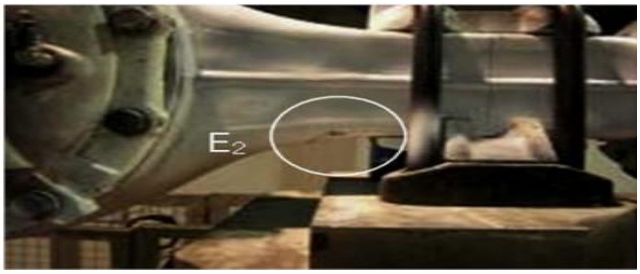


Figure 3: Crack initiation

III. METHODOLOGY

A. Finite Element Analysis

i. CAD models

A full scale of rear axle housing model was developed by using Catia software as shown in figure 4 below. There are two main part body was develop which is consist of main part of the housing and dome. The housing length in X-direction is 1180 mm, in Y-direction is 207.5 mm and in Z-direction is 264.5 mm. The housing essentially consists of thin walled shells, which is having uniform thickness of 5 mm. Mass body for the housing is 61.189 kg and it volume is 7.7948 x 10^{-3} mm². On the front side, a mounting ring is welded with cap to increase rigidity of the housing. This cap was replacing drive shaft position where the drive shaft transmits the rotation to make the rear tires rotate while the vehicle moves. In the other side, a dome is welded to the rear side for sealing reasons. Element A and B represent for the force acting on the axle. The distance between element A and B from the mounting ring is the same [7]. Support 1 and 2 stand for the wheel road contact. The distance between the support housing contact points is equal to the wheel road contact.

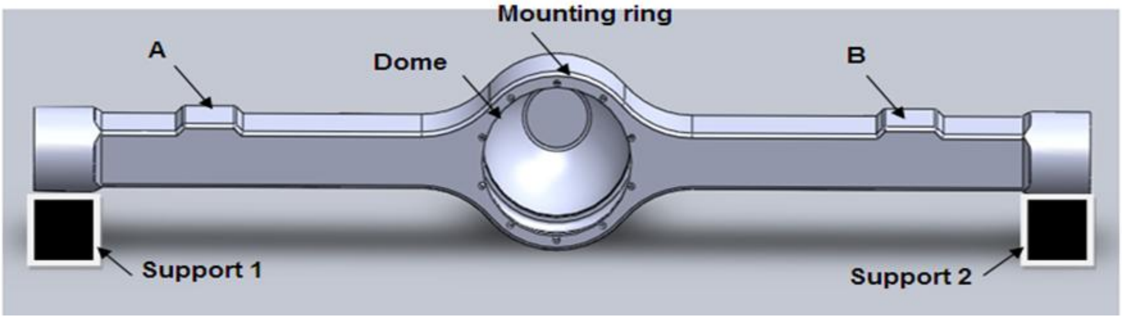


Figure 4: Complete CAD model of housing

ii. Finite Element Model

The FE model used in stress and fatigue analyses is shown in figure 5. From the Catia file, the model was converted to IGES file before run the simulation on FE software. In order to run the simulation, part must be converted to IGES file. This is the rule while the part was developed from the separate software. To build the finite element model, housing was meshed a higher order three-dimensional solid element, which has a quadratic displacement behavior and is well suited to model irregular meshes. The element size is defined by 10 mm completely bonded contact was chosen as the contact condition for all welded surfaces. By using small size for the element mesh, the result shown have more accurate than using a greater size of element. As a result for this housing, the total FE model consisted is 53245 nodes which are consist of 30118 element.

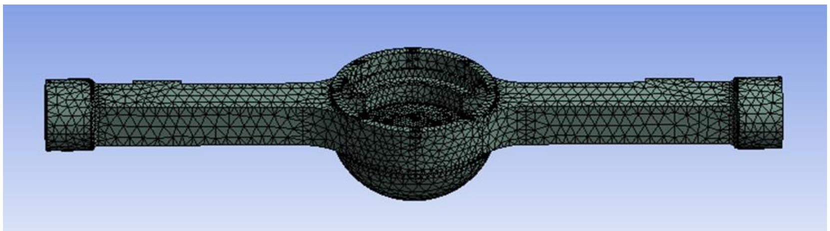


Figure 5: Finite Element model of the housing

B. Housing Material

Housing are manufactured by the stamp welding process from 7 mm thick sheets made from a micro alloyed fine grained, hot formable normalized structural steel S460N. Material number is 1.8901, EN 10025 – 3: 2004 technical delivery conditions for normalized / normalized rolled weld able fine grain structural steels. Weld able fine grain structural steels are in compliance with the EN 10025-3:2004 standard, which defines four levels of mechanical properties. Each grade can be obtained with guaranteed impact properties at -20°C (N grades), or for low-temperature applications at -50°C (NL grades) [8]. Weld able fine grain structural steels are either normalized or obtained by normalizing co-lamination and thus keep their mechanical properties if the welded parts are normalized. These steels are used for welded parts that have to withstand high levels of strain. In order to take effects on the applied process on mechanical properties into FE analysis, each value of mechanical properties of this material was defined before simulation. The material was defined as structural steel with constant variable. The mechanical properties of this material as shown in figure 6**.**

Material	Modulus of elasticity, E (GPa)	Poisson's ratio, v	Yield strength, Sy (Mpa)	Ultimate strength, Sut (Mpa)
S460N (1.8901)	208.5	0.3	497.5	643

Figure 6: Mechanical Properties

C. Loading Condition

Load applied to the FE model was chosen according to the loading range used in real vehicle condition. Tests were performed on a 5 metric tons loading capacity. Housing for a rear axle, which is supported by two air springs in real life but in FE analysis support was defined as in fixed condition. The load pressed on the element A and B was divided with same loading condition which is each element was given starting 0 tons up to 2.5 tons load in 12 seconds as shown in figure 7 below. The load given was used to determine the maximum loading condition can be supported by the housing before failure. During the loading press, the bending moment occurs before the housing failure. The maximum loading can be supported by the housing before failure determined on this analysis is while loading at 4224.755 kg.

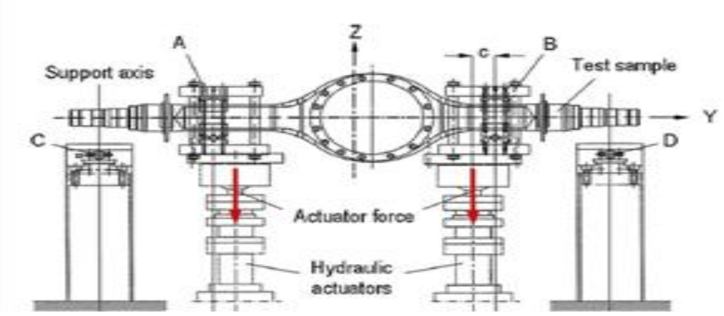


Figure 7: Schematic for loading condition

IV. RESULT AND DISCUSSION

A. Equivalent von-Mises Stress

This result based on the equivalent tensile stress known as von Mises theory where this theory state that the failure at a particular location occurs when the energy of distortion reaches the same energy for failure in tension. In this case, yielding occurs when the equivalent stress reaches the yield strength of the material in simple tension. By applying the force data before in 12 seconds, the result of the stress was shown in figure 9 below.

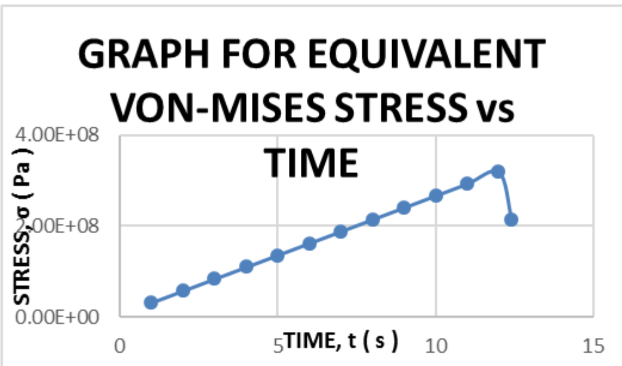


Fig. 8: Graph for Equivalent von-Mises Stress vs Time

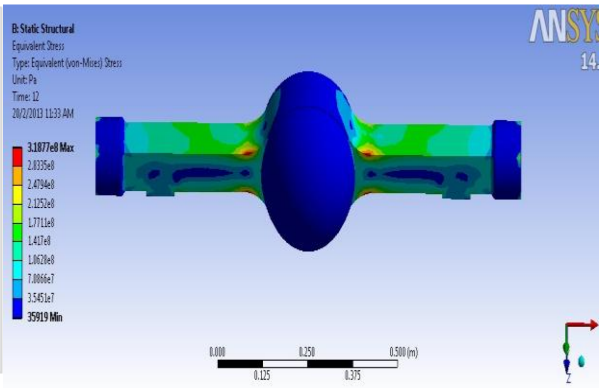
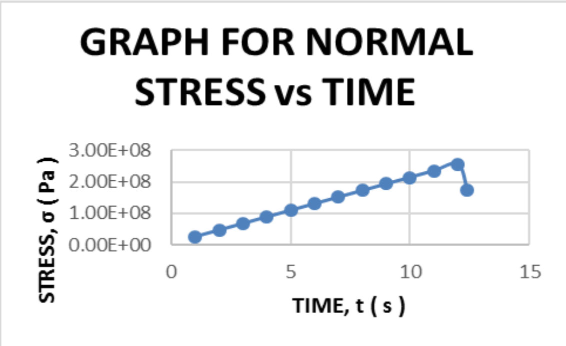


Fig. 9: Equivalent von-Mises Stress

From the result, the graph for equivalent von-Mises stress versus time was constructed. Starting from 0 second until 12 second, the stress was increase based on the load applied on it. The stress increase because of the loading applied is increase. This is the relationship between the load applied and stress occurs. Based on the graph shown in figure 8, the result exposed that the minimum equivalent von-Mises stress is 0.0036 MPa when time taken reaches 1 second. The minimum values of this stress means that the load applied still do not achieve the maximum load applied. When time is increase, the stress also increase directly proportional to the time until reach the maximum stress before the axle housing failure. After 12 second, the stress reach the maximum stress where the stress obtained from this analysis is 318.77 MPa. This means after 318.77 MPa the failure is initiated and these maximum stresses occur on the expected failure region where the failure initiated.

B. Normal Stress



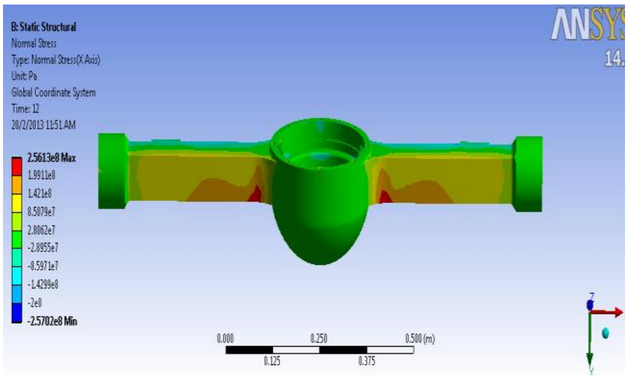


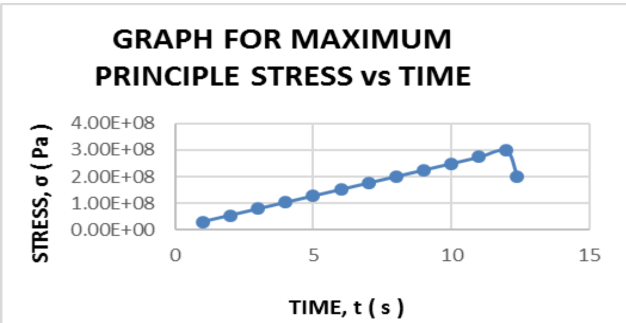
Fig.10: Graph for Normal Stress versus Time

Fig 11: Normal Stress

From the result as shown in figure 11 above, the graph for normal stress versus time was constructed. Initial value from 0 second until 12 second, the stress was raise based on the load acting on it. The stress increase because of the loading applied is increase. The relationship between the normal stress and time is directly relative where stress is increasing when time is increase. Based on the graph shown in figure 10, the result exposed that the minimum normal stress is 0.082 MPa when time initially at 1 second. The minimum values of this stress means that the load applied still do not achieve the maximum load applied. The stress increase proportional to the time until reach the maximum stress before the axle housing failure. After 12 second, the stress reach the maximum stress where the stress obtained from this analysis based on normal stress is 25.613 MPa. This means after 25.613 MPa the failure is initiated and these maximum normal stresses occur on the expected failure region where the failure initiated.

C. Principal Stresses

The principal stresses are defined as the algebraically maximum and minimum values of the normal stresses and the planes on which stress act. On this stress component analysis, the stress on rear axle housing was determined by using finite element analysis. There are two types of result for principal stress which is divided by maximum and the minimum principal stress. The maximum shear stress or maximum principal shear stress is equal to one-half the difference between the largest and smallest principal stresses, and acts on the plane that bisects the angle between the directions of the largest and smallest principal stresses.



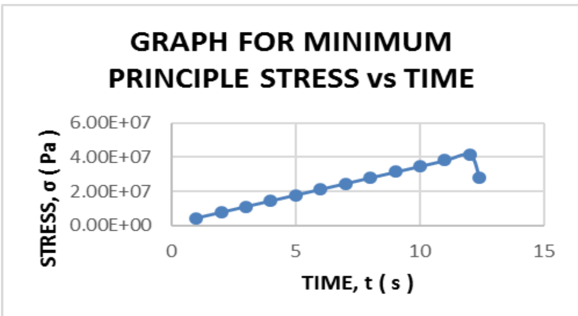


Figure 12: Graph for Maximum Principal Stress versus Time

Figure 13: Graph for Minimum Principal Stress versus Time

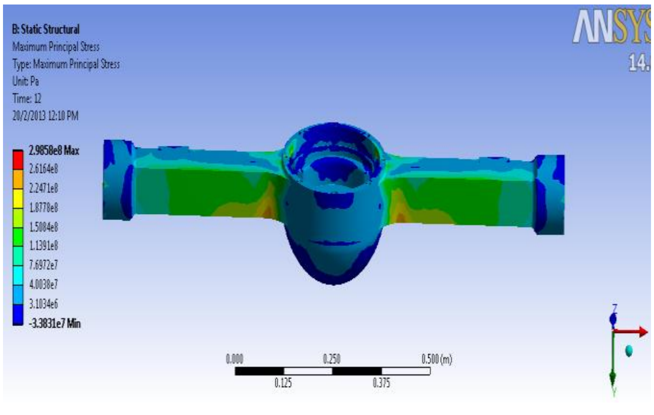


Figure 14: Maximum Principal Stress

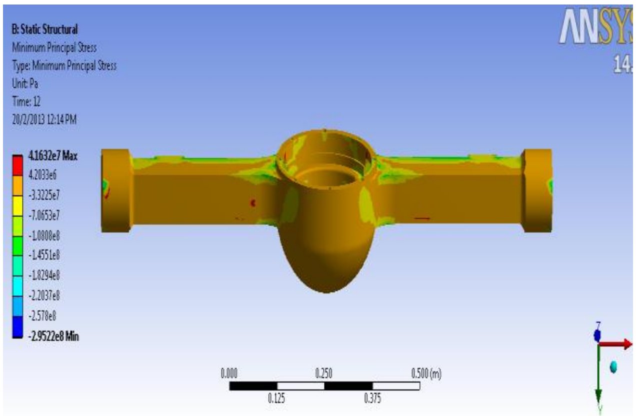


Figure 15: Minimum Principal Stress

Based on these two types of principal stress (maximum and minimum) as shown in figure 14 and 15 above, the value of the principal stress obtained from the analysis result shown that the maximum principal stress has the higher stress compare to the minimum principal stress. The value obtain fulfill the theory statement which is the maximum must be higher than the minimum value. These two graph which are shown in figure 12 and 13 were had a similar trend line where the stress is directly proportional to the time. From the analysis, the maximum principal stress obtained is 298MPa and the minimum principal stress is 42 MPa. The maximum value for principal stress obtained at time 12 second. The stress occurs at the critical region of the axle housing where the failure initiated.

D. Reason of the Premature Failure and Maximum Loading Capacity

Premature failure occurs when the axle housing reach the limit of fatigue strength. The axle is no longer can prevent the failure if the stress acting is higher than the true fatigue strength. From this analysis, while the force acting on the axle is 5 tons (5000 kg), the axle became failure. In order to predict the maximum loading capacity, the axle was tested in 12 second where starting from 1 second until 12 second; the value of load adding is increased by 5000 N. So, at time 12 second the total load acting on axle housing is 50000 N ≈ 5 tons. From the analysis result show that the maximum loading capacity is 4224.755 kg ≈ 42000 N where σ_{max} for equivalent von-Mises stress is 318.77 MPa.

E. S-N Diagram and Fatigue Life Prediction

By using Finite element software the S-N curve for axle housing material was developed directly after the mechanical properties for the housing was defined. The cycles developed for this diagram is up to 10^6 cycles. According to the theoretical, when the alternating stress is decrease, the fatigue life for the material is increase. As a result, the curve as shown in figure 16 below fulfills the requirement for the structural steel curve behaviour. Fatigue life from this experiment is 1 x 10^9 cycles. The true fatigue strength from the calculation is 361.0290 MPa. At this point the obtained minimum value of n from this analysis is 1.45. The minimum value of n from calculated is 1.56 where the maximum stress concentration was observed. This mean at critical region, fatigue crack can initiate before 1 x 10^9 cycles as observed in this analysis.

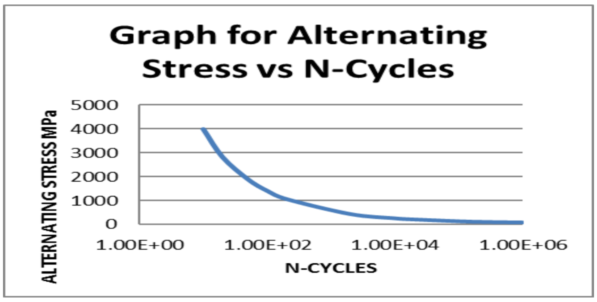


Figure 16: Graph S-N curve for S460N material

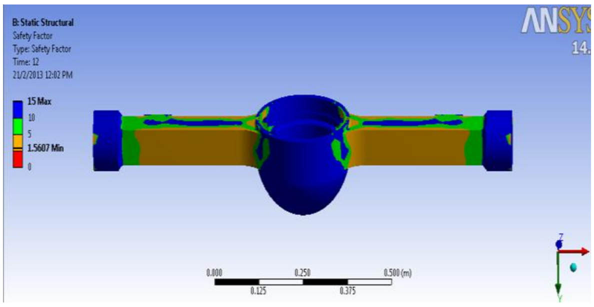


Figure 17: Factor of Safety

i. Fatigue Strength Calculation

Since the rear axle housing is actually loaded with dynamic forces during the service, fatigue analysis was also performed. An estimation of the stress life endurance limit S'_e is given as:

$$S'_e = 0.504 \, . \, S_{ut} \quad (1)$$

Where,

S'_e = Endurance limit

S_{ut} = Ultimate strength

for steels with ultimate strengths S_{ut} of less than 1400 MPa [12]. This represents the fatigue strength at 10^6 cycles and more. For the fatigue life prediction of the part in the range of 10^5–10^6 cycles, the S–N curve of the housing material was predictable by means of a practical method given by FE software. S'_e stands for the stress life endurance limit of ideal laboratory samples. To predict the true fatigue strength S_e for a mechanical component S'_e has to be multiplied by several modifying factors which represent various design, manufacturing, and environmental influences on the fatigue strength. S_e is given as:

$$S_e = k_a k_b k_c k_d k_e S'_e \quad (2)$$

Where,

k_a = surface factor

k_b = size factor

k_c = load factor

k_d = temperature factor

k_e = fatigue strength modifying factor

where k_a a surface factor which depends on surface finish is given as:

$$k_a = aS_{ut}^b \quad (3)$$

Where,

a = factor

b = exponent

Since the roughness of the shell surfaces is similar to hot rolled sheet after hot stamping, recommended values are a = 57.7 and b = - 0.718. The calculated value of k_a = 0.5557 for S_{ut} = 643 MPa. In addition, shot peening, a well-known process to introduce favorable residual stresses in the material surface of a component is also applied on the housing surfaces after hot stamping to increase the fatigue life of the part. Hence k_a is used in the fatigue analysis as 0.959. For non-round sections, size factor k_b can be assumed as 0.75 for the values of the depth of the cross-section h, which is greater than 50 mm. Load factor k_c is given as 1 for bending and temperature factor k_d is 1 for the range of the ambient temperature of T = 0− 250 °C. By means of static FE analysis, it is observed that there are stress concentrated regions on critical area and arm transition areas. Therefore, in addition to the modifying factors mentioned, a fatigue strength modifying factor k_e must be taken into account by means of the static stress concentration factor k_t that is related to fatigue stress concentration factorc k_f. Hence k_e is calculated as:

$$k_e = \frac{1}{k_f} \quad (4)$$

For safety reasons, k_f can be assumed as to be equal to k_t. Because of the dimensions and shape complexity of the housing k_t cannot be derived from data in the standard literature [2]. On the other hand k_t is defined as:

$$k_t = \frac{\sigma_{peak}}{\sigma_{nominal}} \quad (5)$$

where σ_{peak} is the peak stress and at the root of the notch and $\sigma_{nominal}$ the nominal stress which would be present if a stress concentration did not occur σ_{peak} was used as the value obtained from static FE analyses as σ_{max}= 318.77 MPa. To calculate $\sigma_{nominal}$, the rear axle was assumed as a simple beam which has a uniform box profile cross-section X–X of critical region along the longitudinal axis Y and subjected to pure bending . $\sigma_{nominal}$ was computed by means of the model given as:

$$\sigma_{nominal} = \frac{M}{Z} \tag{6}$$

$$Z = \frac{bd^2 - b_1\, d_1^2}{6d} \tag{7}$$

where M is bending moment and Z is the section modulus of the critical cross-section. M was obtained as 5724.6 kN.mm. Section modulus Z was calculated as 39543.96 mm³. Hence $\sigma_{nominal}$ was computed as 144.765 MPa. $k_t \approx k_f$ is found as 2.202 and k_e = 0.454. The S–N curve plotted regarding the modifying factors was defined in the FE software user interface. Stress-life approach was used to determine the fatigue life of the housing material. All fatigue analyses were performed according to infinite life criteria (N $=10^6$ cycles). Von Mises stresses obtained from finite element analyses are utilized in fatigue life calculations. Since the loading has a sinusoidal fluctuating characteristic (mean stress, σ_m > 0), modified Good- man approach given as:

$$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n} \tag{8}$$

was used. Here n stands for factor of safety (as shown in figure 17). Stress amplitude σ_a is given as:

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

(9)

and the mean stres σ_m can be expressed as:

$$\sigma_a = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{10}$$

Here, both σ_{max} corresponding to maximum 381.13 MPa and and σ_{min} matching a minimum 0.0035924 MPa of vertical load were obtained via FE analysis. Distribution of n at the lower shell can be seen in factor of safety distribution region. In the light of the fatigue analysis results, it was estimated that crack initiation can occur at the region E1 and E2 of the outer shell surface. The value of n is obtained.

ii. Factor of Safety

Factor of safety is used to provide a design margin over the theoretical design capacity to allow for uncertainty in the design process. Factor of safety to be used in this design is essentially a compromise the benefit of increased safety and reliability. From the analysis, the value of safety factor of the axle is 1.56. By using manual calculation, the value of safety factor is 1.45. So, the error between both result is 7% still can be accepted because less than 10% error.

F. Rear Axle Model Solution

By increasing the thickness sheet metal from 5 mm to 7 mm at critical region as shown in figure 18 below, the rigidity and safety of the axle housing can be improved. From the theoretical, when the area is increase, the value of stress is decrease. When the stress is decrease, the axle can prevent from failure. So, in order to reduce the value of stress acting on the axle, the new design with more thickness than before can be used to solve the axle problem and the axle can be used longer than before.

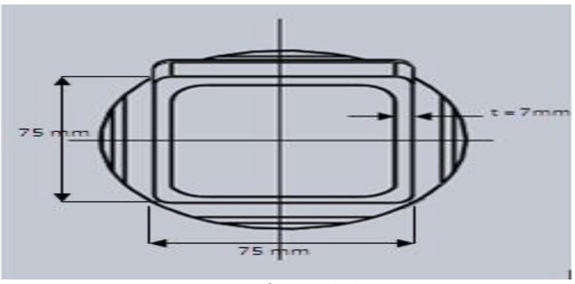


Fig. 18: Geometry of rear axle housing

V. CONCLUSION

Stress distribution analysis of rear axle housing was investigated by using finite element analysis. The model of the axle was developed by using CAD software. The dimension of the axle housing follow the real dimension based on the data collected. In order to run the simulation in the FE software, the model must be meshed. By using small values of the element size, the result given will more accurate. All the parameter and boundary condition for the rear axle was defined in the FE analysis before run the simulation. In the analysis, in which the stress are distribute, stress concentration from the load given to the axle housing make the axle housing failure. From the analysis, the result for stress distribution for equivalent von-Mises stress, normal stress, maximum principal stress, and minimum principle stress was analyzed and determined.The reason failure occurs is because the axle housing no longer can prevent the load given onto it. From the several load given, the maximum load for the housing can stand was determined by using FE analysis. The result show that the maximum load can be carried by this rear axle housing is 4224.755 kg ≈ 42000 N.

In this FE analysis, the fatigue life cycle for the axle material also was computed and the result shown that the axle material can stand up to 1x 10^6 cycles before the crack initiated on the critical region. The factor of safety for the axle housing was determined from the analysis and manual calculation. The values from the analysis is n = 1.56 and from the calculation is 1.45 can be accepted because only has 7% of the error. In order to increase the fatigue life of the housing, new design of rear axle model was developed with increasing the thickness of sheet metal to 7 mm used to make the housing can stand longer and increase the rigidity of the axle housing. But other consideration such as material changing and redesigning also can be used to improve the problem

REFERENCES

[1] Woongsang Jeong, Jinhee Jang, Changsoo Han. Modeling & Dynamic Analysis for Four Wheel Steering Vehicle.
[2] M.M. Topac, H. Gunal, N.S. Kuralay (2008). Fatigue failure prediction of a rear axle housing prototype by using finiteElement analysis.
[3] Osman Asi (2006). Fatigue failure of a rear axle shaft of an automobile.
[4] Carol Francois (2011). What is an axle, http://www.wisegeek.com/what-is-an-axle.htm, 21 March 2011.
[5] James M. Gere (2002). *Mechanics of Materials.* 5th SI Edition. Nelson Thornes Ltd, pp 139.
[6] Richard G. Budynas & J. Keith Nisbett (2008). *Fatigue Failure Resulting from Variable Loading*, 8th Edition In SI Units, pp 257-319.
[7] Higgins, Raymond Aurelius (2001). The Properties of Engineering Material, 2nd Edition. Raymond A. Higgins, pp. 384-407.
[8] Yung-Li Lee, Jwo Pan, Richard Hathaway, Mark Barkey (2005). *Fatigue Testing and Analysis*. Elsevier Butterworth-Heinemann, pp 107-176.
[9] Yasushi Ito, Alan M. Shih and Bharat K. Soni. Reliable Isotropic Tetrahedral Mesh Generation Based on an Advancing Front Method.
[10] ANSYS Theory References. ANSYS Release 14.0 ANSYS, Inc; 2011.
[11] Shigley JE, Mischke C (1989). *Mechanical Engineering Design.* 8th Edition. New York, McGraw-Hill, pp 286.
[12] Li-Ping Lei, Jeong Kim, Beom-Soo Kang (2000). Analysis and design of hydroforming process for automobile rear axle housing by FEM.
[13] Hong SU, Ph.D. Automotive CAE Durability Analysis Using Random Vibration Approach
[14] M. Davis, Y.S. Mohammed, A.A. Elmustafa, RF. Martin, C. Ritinski (2010). Designing for Static and Dynamic Loading of a Gear Reducer Housing with FEA.
[15] E. Makevet, I. Roman (2001). Failure analysis of a final drive transmission in off-road vehicles.
[16] K.J. Anusavice and B. Hojjatie (1987). Stress Distribution in Metal-Ceramic Crowns with a Facial Porcelain Margin.
[17] Chris Dolling (2006). Material Selection
[18] F.P. Brennan and W.D. Dover (1995). Stress Intensity Factors for Threaded Connections.
[19] Pilkey WD, Pilkey DF. Peterson's (2008). *Stress Concentration Factor*. 3rd Edition. John Wiley & Sons, pp 159.
[20] Collins, J.A. (Jack A.) (1993). *Failure of Materials in Mechanical Design*, pp 34-37.
[21] Walter D. Pilkey (2005). *Formula for Stress, Strain and Structural Matrices*. 2nd Edition. John Wiley & Sons, Inc, pp 137-344.
[22] Schijve J (2001). *Fatigue of structures and materials*. Dordrecht, Netherlands: Kluwer Academic Publishers, pp. 68-72.