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Fatigue failure prediction of a rear axle housing prototype by using finite element analysis

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ABSTRACT

A premature failure that occurs prior to the expected load cycles during the vertical fatigue tests of a rear axle housing prototype is studied. In these tests, crack mainly originated from the same region on test samples. To determine the reason of the failure, a detailed CAD model of the housing was developed. Mechanical properties of the housing material were determined via tensile tests. Using these data, stress and fatigue analyses were performed by finite element method. Fatigue crack initiation locations and minimum number of load cycles before failure initiation were determined. Results provided from tests were compared with the analyses. Design enhancement solutions were proposed to increase the fatigue life of the housing.

1. Introduction

Due to their higher loading capacity, solid axles are typically used in the heavy commercial vehicles [1]. Structure of a solid axle can be seen in Fig. 1. During the vehicle service life, dynamic forces caused by the road surface roughness produce dynamic stresses and these forces 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. Before mass production, loading capacity and fatigue life of an axle housing prototype under dynamic vertical forces should be determined by vertical fatigue tests as shown in Fig. 2. In these tests, a predicted cyclic vertical load is applied by hydraulic actuators on specimens until fatigue crack initiation occurs. According to the acceptance criteria, a housing prototype has to resist $N = 5 \times 10^5$ load cycles without a fatigue failure. During the vertical fatigue tests of an asymmetric type axle housing as shown in Fig. 3, fatigue crack initiation occured on some of the prototypes before this load cycle limit. It was observed that the minimum load cycle before fatigue failure was about 3.7×10^5. In these tests, crack originated at the banjo transition regions E_1 and E_2. An example of the premature failure can be seen in Fig. 4.

To predict the reason of the premature failure, a detailed solid model of the housing was formed in CATIA® V5R15 commercial software. By using this model, finite element model was constructed. The stress and fatigue analyses were performed via ANSYS® Workbench V11.0 commercial finite element software. Housing material properties were obtained from tensile tests, which were utilized in FE analyses. Maximum dynamic axle loads, obtained from a vehicle dynamics simulation, performed via RecurDyn® a commercial CAE software were also used. By these analyses, the stress concentration regions were attained. To perform the fatigue analysis, an estimated S–N curve for housing material was constructed by taking the fatigue strength modifying factors into account. The results from the analysis were compared to the vertical

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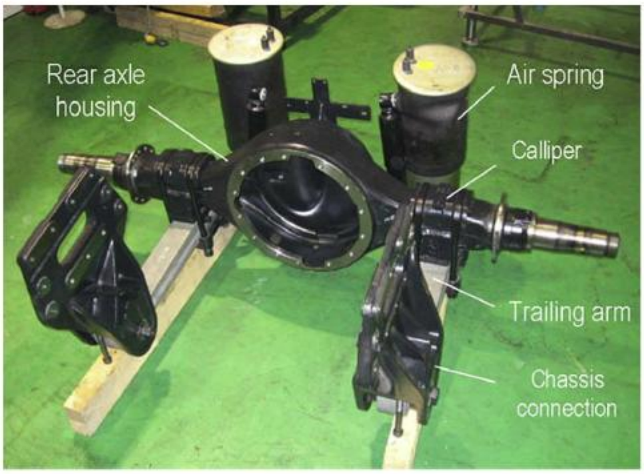

Fig. 1. Rear axle assembly of a commercial vehicle.

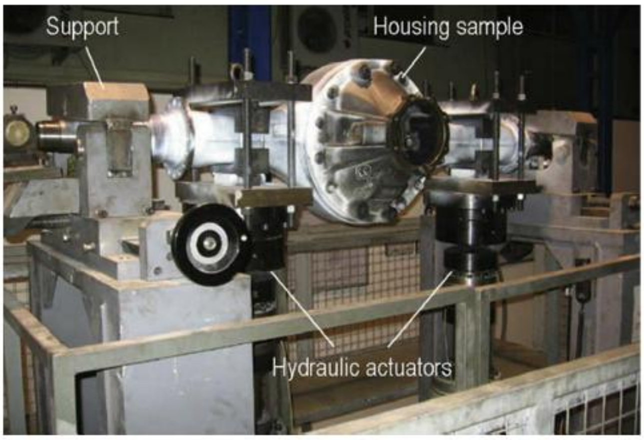

Fig. 2. Vertical fatigue test of housing prototype.

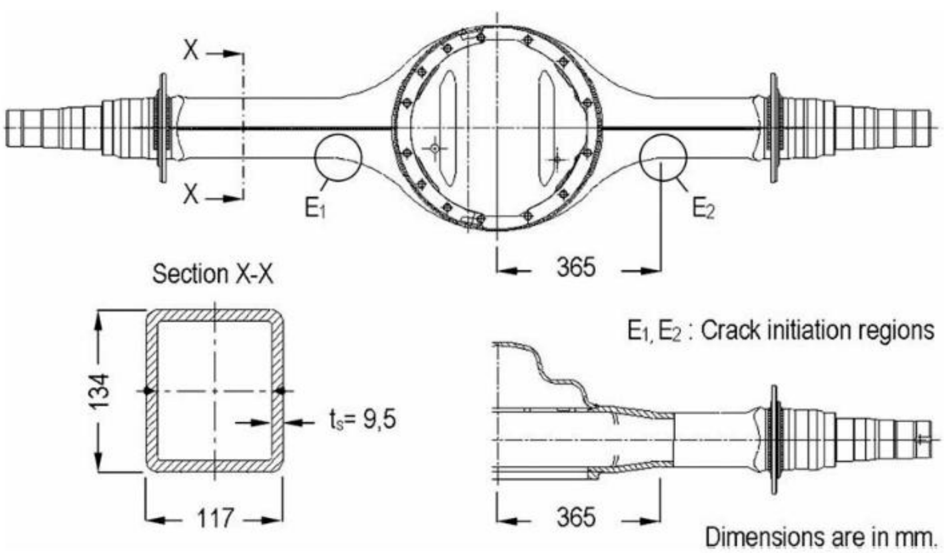

Fig. 3. Geometry of the axle housing.

fatigue test results. To prevent the premature failure and obtain an enhanced fatigue life, some design enhancement solutions were proposed.

Fig. 4. Fatigue crack on the lower half of a test sample.

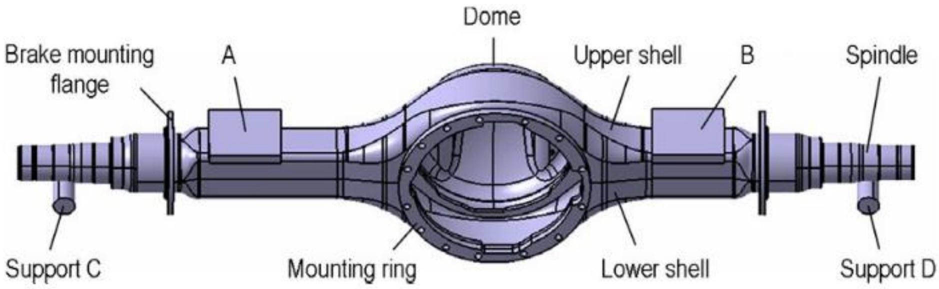

Fig. 5. Complete CAD model of the housing.

2. Finite element model

2.1. CAD and FE models

A full scaled CAD model of the housing was prepared for the analyses as shown in Fig. 5. The housing essentially consists of two equivalent thin walled shells, which have a uniform thickness of 9.5 mm and welded along the neutral axis of the rear axle. On the front side, a mounting ring is welded to the housing assembly to increase rigidity and the differential carrier is bolted on it. For sealing reasons, a dome is welded to the rear side. Here, elements A and B represent trailing arm–calliper connections. Supports C and D stand for the wheel–road contacts. The distance between the support-housing contact points is equal to the wheel track of the rear axle. The solid model of the housing was composed via CATIA® V5R15. CAD model of the complete housing was imported into ANSYS® Workbench V11.0 preproccessing environment to constitute the FE model required in the analyses. The FE model used in stress and fatigue analyses is shown in Fig. 6. To build the finite element model, housing was meshed using SOLID187, a higher order three-dimensional solid element, which has a quadratic displacement behavior and is well suited to model irregular meshes. The element is defined by 10 nodes having three translational DOF at each node [2]. To model the contact between the structural parts of the housing, CONTA174 and TARGE170 elements were used. Completely bonded contact was chosen as the contact condition for all welded surfaces. FE model consisted of 779,305 elements and 1,287,354 nodes.

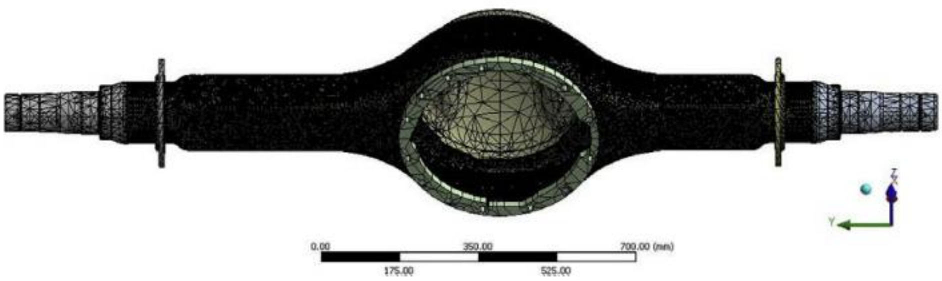

Fig. 6. Finite element model of the housing.

Table 1
Chemical properties of S460N (wt.%)

Standard	C (max.)	Si (max.)	Mn	P (max.)	S (max.)	Al (min.)	Cr (max.)	Cu (max.)	Mo (max.)	Nb (max.)	Ni (max.)	Ti (max.)	V (max.)	N (max.)
DIN EN 10025-3	0.19	0.60	1.40–1.70	0.020	0.010	0.02	0.10	0.10	0.08	0.05	0.40	0.03	0.15	0.012

Table 2
Results of tensile tests

Material	Modulus of elasticity, E (GPa)	Poisson's ratio, v	Yield strength, S_y (MPa)	Ultimate strength, S_{ut} (MPa)	Maximum elongation, ε_{max} (%)
S460N (1.8901)	208.5	0.3	497.5	629.9	26.8

2.2. Housing material

Shells are manufactured by the stamp-welding process from 9.5 mm thick sheets made from a micro alloyed fine grained, hot formable, normalized structural steel S460N (Material number 1.8901, equivalent to E460 according to ISO standard [3]). The chemical composition of the material which was obtained from the supplier is given in Table 1 [4]. The mechanical properties of non-processed S460N can be found in Ref. [5]. However, several processes are applied to the housing material during manufacturing, including annealing up to 800 °C and hot stamping at ca. 750 °C. In order to take the effects of the applied processes on mechanical properties into account in FE analysis and determine the exact properties of the processed material, five specimens were extracted from axle housing samples and tensile tests have been carried out. All of the tests were conducted at room temperature. Regions where the specimens were extracted from housing were beyond the heat affected zone. The results given in Table 2 are the minimum values obtained from five specimens and they were used in the finite element simulation. Behaviour of the material was defined as linear isotropic material model.

2.3. Loading conditions

Load applied to the FE model was chosen according to the loading range used during the vertical fatigue tests where premature failure was seen. Tests were performed on an 80 metric tons loading capacity test rig shown in Fig. 7. The device consists of two electro-hydraulic actuators with load cells and servovalves, which were positioned at the calliper connections

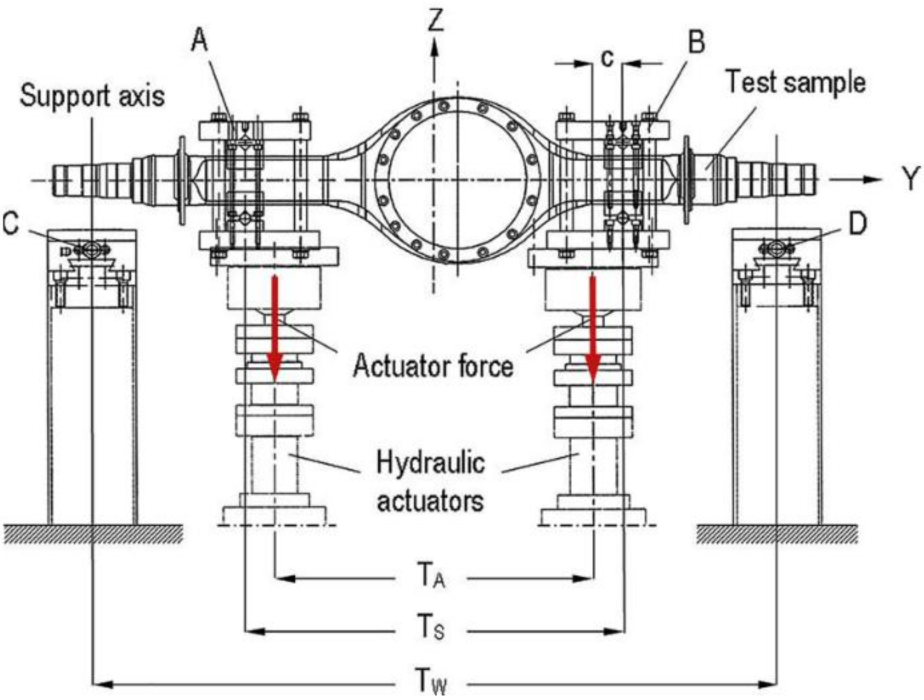

Fig. 7. Schematic for vertical fatigue test.

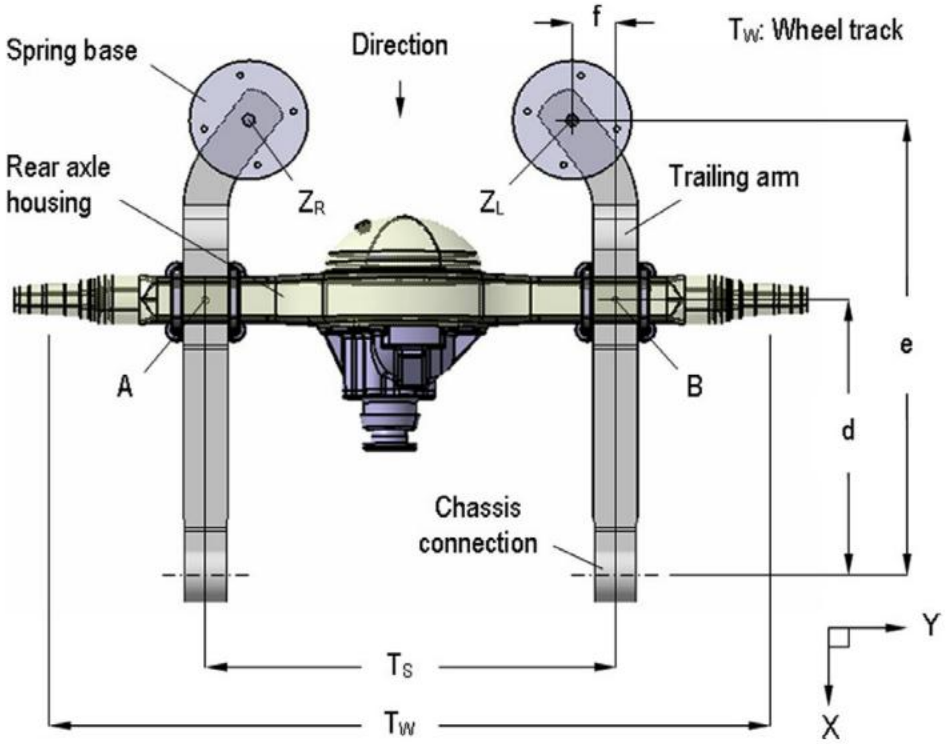

Fig. 8. Eccentric loading geometry of trailing arm.

A and B. Here, T_S represents the track between two callipers and the distance between C and D supports T_W, the wheel track of the actual rear axle. Housing prototype is designed for a rear axle, which is supported by two air springs shown in Fig. 8. Because of the eccentric loading geometry of the trailing arm, spring force also produces torsion, which causes an extra bending moment ΔM on housing. Extra bending effect was applied on test samples by means of an eccentricity c of hydraulic actuators as seen in Fig. 7. Maximum static design load per spring is given as $F = 2850$ kg. The load acts to the spring seats vertically at the points Z_R and Z_L. This causes a static reaction of $P = 4550$ kg on callipers A and B. Because of the vertical acceleration of lumped mass of the vehicle body due to the road surface roughness, maximum dynamic loading on each calliper was estimated about twice as much as P. Loading range was obtained as 182–9100 kg by means of a computer aided road simulation, performed by RecurDyn® a commercial CAE software. Characteristics of the test load used in the vertical fatigue tests can be seen in Fig. 9. FE analysis was performed considering the maximum dynamic load of 9100 kg along the effect of the extra bending moment ΔM. Vertical loading model of the housing, prepared with respect to Ref. [6] is given in Fig. 10.

3. Finite element analysis and results

FE analysis was used to predict the exact location of the regions where tensile stress concentrations were seen and fatigue life is relatively lower. P and ΔM were applied to the model at the calliper connections in accordance with Fig. 10. Stress

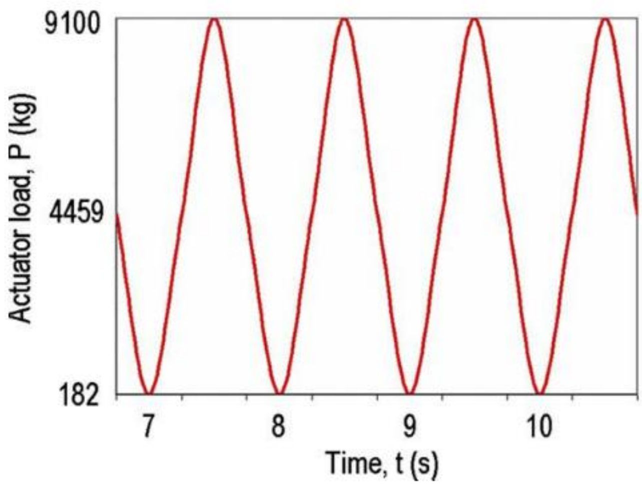

Fig. 9. Characteristic of actuator load used in fatigue tests.

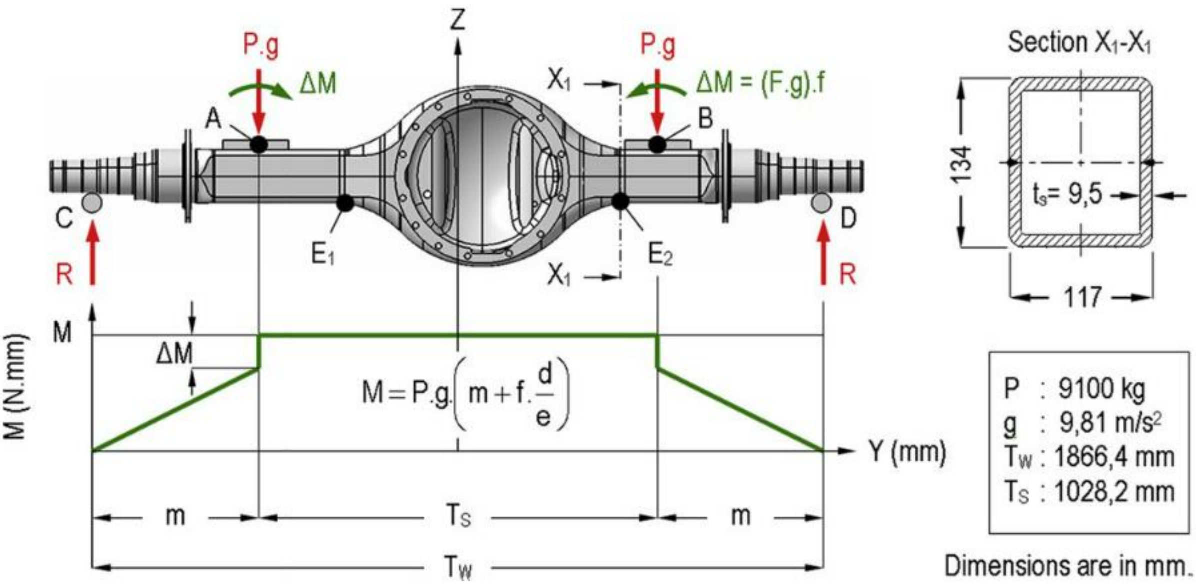

Fig. 10. Applied loads and bending moment diagram of housing.

analysis was carried out using ANSYS® Workbench V11.0 a commercial FEA software on a 1.86 GHz Intel quad-core Xeon Processor HP xw8400 Workstation. Fig. 11 shows equivalent von Mises stress distribution provided from the FE analysis. Results showed that there are tensile stress concentrated regions F_1 and F_2 at banjo transition area of the carrier mounting side of the lower shell. Locations of the critical regions and the premature fatigue failure are the same as seen in Fig. 12. The calculated maximum von Mises stress is σ_{max} = 388.7 MPa; 78.1% of the yielding point of material. This means that housing prototype satisfies the safety conditions for maximum loading if it is exerted statically.

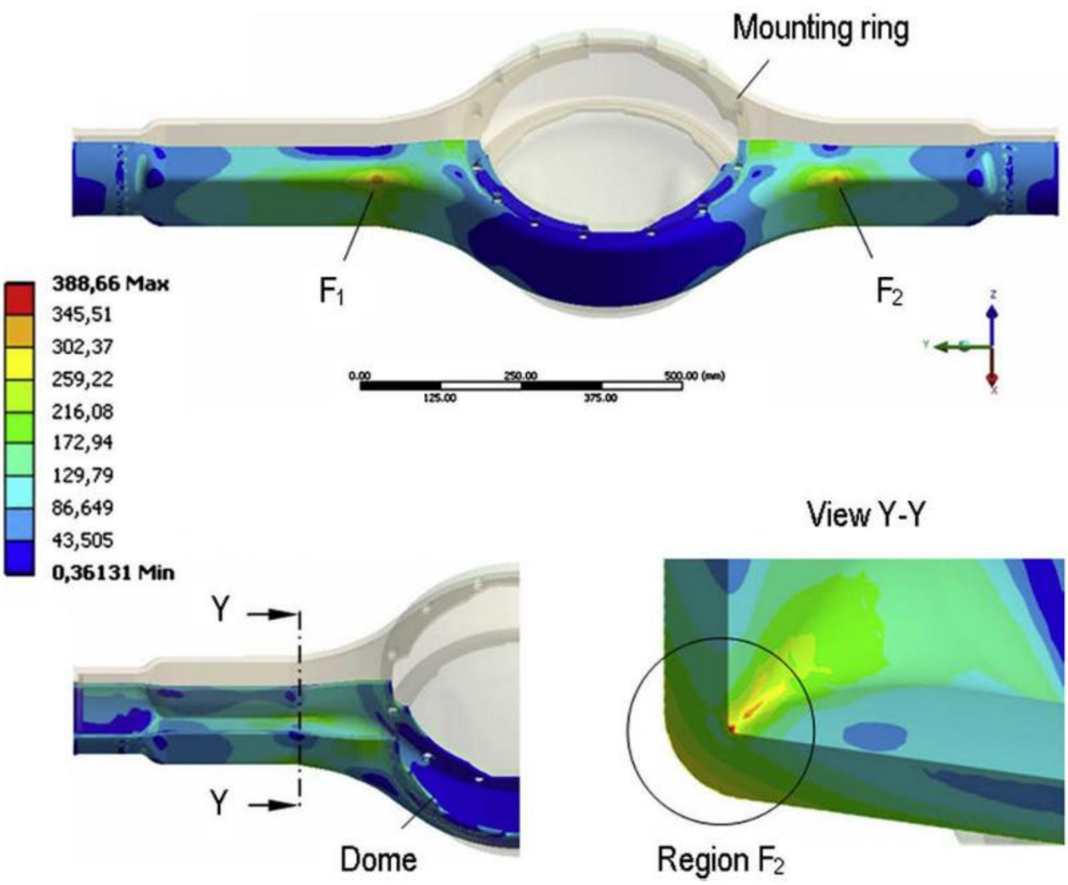

Fig. 11. Overall stress distribution on the lower shell.

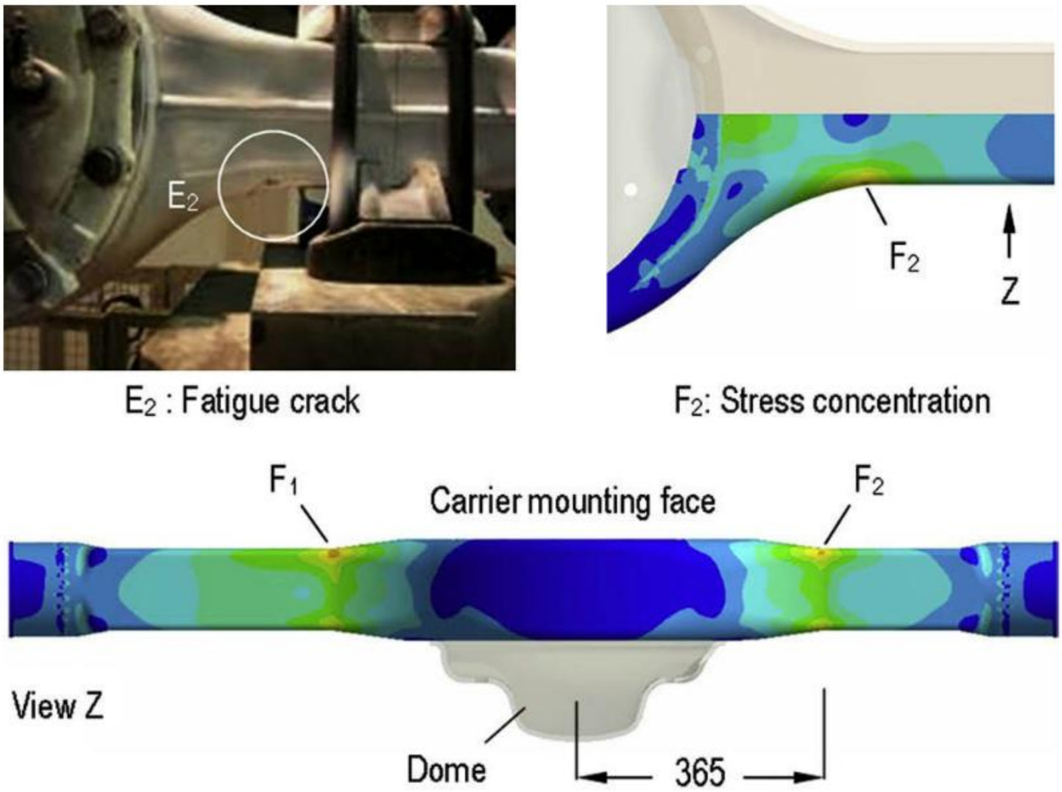

Fig. 12. Comparison of test and analysis results.

4. Fatigue life prediction

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

$$S'_e = 0.504 \cdot S_{ut} \tag{1}$$

for steels with ultimate strengths S_{ut} of less than 1400 MPa [7,8]. 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 estimated by means of a practical method given in Ref. [9] that uses the data obtained from the simple tensile tests.

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 [10]. S_e is given as

$$S_e = k_a k_b k_c k_d k_e S'_e \tag{2}$$

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

$$k_a = a S_{ut}^b \tag{3}$$

Since the roughness of the shell surfaces is similar to hot rolled sheet after hot stamping, recommended values are $a = 57.7$ and $b = -0.718$ [7]. The calculated value of $k_a = 0.564$ for $S_{ut} = 629.9$ MPa. In addition, shot peening, a well-known process to introduce favourable 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. In literature this increase is given by some as 70% [9]. 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 [11].

By means of static FE analysis, it is observed that there are stress concentrated regions on banjo 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 factor K_f. Hence k_e is calculated as

$$k_e = \frac{1}{K_f} \tag{4}$$

For safety reasons, K_f can be assumed as to be equal to K_t [7]. Because of the dimensions and shape complexity of the housing K_t cannot be derived from data in the standard literature. On the other hand K_t is defined as

$$K_{\mathrm{t}} = \frac{\sigma_{\mathrm{peak}}}{\sigma_{\mathrm{nominal}}} \tag{5}$$

where σ_{peak} is the peak stress and at the root of the notch and $\sigma_{\mathrm{nominal}}$ the nominal stress which would be present if a stress concentration did not occur [9,12] σ_{peak} was used as the value obtained from static FE analyses as $\sigma_{\mathrm{max}} = 388.7$ MPa. To calculate $\sigma_{\mathrm{nominal}}$, the rear axle was assumed as a simple beam which has a uniform box profile cross-section X_1–X_1 of critical region along the longitudinal axis Y and subjected to pure bending [6]. $\sigma_{\mathrm{nominal}}$ was computed by means of the model given in Fig. 10 as

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

where M is bending moment and Z is the section modulus of the critical cross-section. M was obtained as 41.9×10^6 N mm. Section modulus Z was calculated as 127507 mm^3. Hence $\sigma_{\mathrm{nominal}}$ was computed as 329 MPa. $K_t \approx K_f$ is found as 1.181 and $k_e = 0.846$. The S–N curve plotted regarding the modifying factors was defined in the ANSYS® Workbench V11.0 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, $\sigma_{\mathrm{m}} > 0$), modified Goodman approach given as

$$\frac{\sigma_a}{S_e} + \frac{\sigma_{\mathrm{m}}}{S_{ut}} = \frac{1}{n} \tag{7}$$

was used [9]. Here n stands for factor of safety. Stress amplitude σ_{a} is given as

$$\sigma_{\mathrm{a}} = \frac{\sigma_{\mathrm{max}} - \sigma_{\mathrm{min}}}{2} \tag{8}$$

and the mean stres σ_{m} can be expressed as

$$\sigma_{\mathrm{m}} = \frac{\sigma_{\mathrm{max}} + \sigma_{\mathrm{min}}}{2} \tag{9}$$

Here, both σ_{max} corresponding to maximum 9100 kg and and σ_{min} matching a minimum 182 kg of vertical load were obtained via FE analysis. Distribution of n at the lower shell can be seen in Fig. 13. In the light of the fatigue analysis results, it was estimated that crack initiation can occur at the region F_1 of the outer shell surface at ca. 3.6×10^5 cycles, which is lower than the expected minimum fatigue life of 5×10^5 cycles. Here the obtained minimum value of n is 0.93. On the inner surface of the shell, the minimum value of n was calculated as 0.767 at the region F_2 where the maximum stress concentration was observed. This means, at the two regions F_1 and F_2 fatigue crack can initiate before 5×10^5 load cycles as observed in the vertical fatigue tests.

5. Results and discussion

FE analyses showed that the regions, where fatigue failure was initiated during vertical fatigue tests, are subjected to stress concentration, which can cause a premature failure before the predicted 5×10^5 minimum cycles limit. The results are in agreement with the results of vertical fatigue tests. Enhancement of the fatigue life of the housing is dependent on the decrease of the stress concentration. The simplest way to reduce the stress concentration and improve the fatigue life is to increase the thickness of the sheet metal. However, except regions F_1 and F_2, the housing satisfies the infinite life criteria. An increase of sheet metal thickness causes an unnecessary weight increase. For example, a thickness increase of say 0.5 mm enhances the fatigue limit of the housing material at the critical regions up to more than 5.8×10^5 cycles which is higher than the desired limit. On the other hand, this also means a weight increase about 5% of unsprung mass of vehicle,

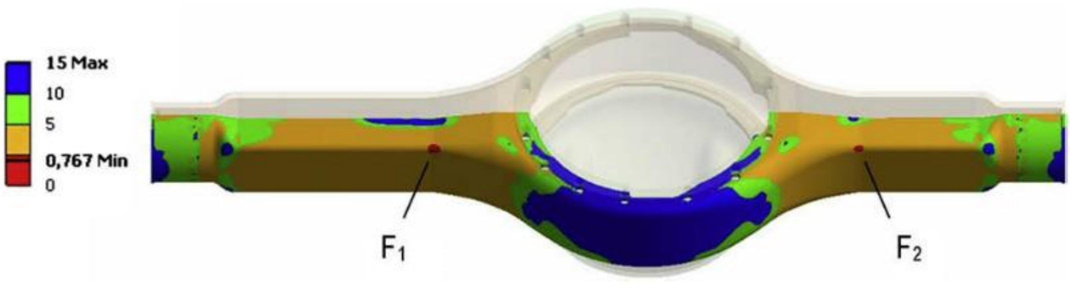

Fig. 13. Factor of safety distribution on the lower shell.

therefore it is not a practical solution. As an alternative, the transition geometry may be redesigned. Smoother transition geometry may offer an enhanced fatigue life without any weight increase.

In addition, shape of the reinforcement ring also affects the stress concentration. In the studied design, thickness of the ring is 20 mm. To predict the effects of the ring, FE analysis was repeated for the case without the ring. Maximum von Mises stress was obtained as 428 MPa at the critical region F_2. This means, the use of the ring decreases stress concentration about 10%. By increasing the thickness of the part, it is possible to obtain an enhanced rigidity. In this design, owing to the configuration of the drivetrain limit this increase to 5 mm. The static and fatigue analyses were composed according to this change in the ring shape. However, analyses pointed out that this increase on its own, enhances the fatigue life of the housing to a certain extent, which is not sufficient enough to obtain the desired minimum load cycles of 5×10^5. Therefore, ring thickness increase may be applied together with the redesign of the transition geometry.

6. Conclusion

Premature fatigue failure of a truck rear axle housing prototype was investigated by using finite element analysis. In the analyses, in which the vertical fatigue test procedure was simulated, stress concentrated regions were predicted at the banjo transition area. The regions in which the fatigue cracks originated were well-matched with the results of the analyses. By using FE analysis the location of the failure can be predicted.

Critical regions determined are subjected to a combined steady and cyclic tensile stress. The crack causing fracture is initiated at the stress concentrated regions of the housing. Although the housing prototype satisfies the static endurance condition for the maximum vertical load, analyses showed that premature fatigue failure can occur prior to the predicted 5×10^5 minimum cycles limit, if this load is applied in a cyclic manner. FE analyses also enable to provide an estimation of the number of cycles before fatigue failure initiation.

In order to solve the problem, increasing the thickness of sheet metal is not a practical solution because of the weight increase of housing. An application including both redesigning of the banjo transition area and increasing the thickness of the reinforcement ring may be a good alternative to obtain a longer fatigue life, which can satisfy minimum design criteria.

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References

[1] Reimpell J, Stoll H, Betzler JW. The automotive chassis: engineering principles. Butterworth-Heinemann; 2002. p. 39.
[2] ANSYS Theory Reference. ANSYS Release 10.0. ANSYS, Inc.; 2005.
[3] Jiang Y, Hertel O, Vormwald M. An experimental evaluation of three critical plane multiaxial fatigue criteria. Int J Fatigue 2007;29:1490–502.
[4] Erdemir Product Catalogue, Eregli Iron and Steel Works Co.; 2007. p.50 [in Turkish].
[5] Hoffmeyer J, Döring R, Seeger T, Vormwald M. Deformation behaviour, short crack growth and fatigue lives under multiaxial nonproportional loading. Int J Fatigue 2006;28:508–20.
[6] Gordon KW. Design, evaluation and selection of heavy-duty rear axles. SAE Trans 1955;63:5–34.
[7] Shigley JE, Mischke C. Mechanical engineering design. New York: McGraw-Hill; 1989. p. 286–8.
[8] Lee YL, Pan J, Hataway R, Barkey M. Fatigue testing and analysis. New York: Elsevier Butterworth-Heinemann; 2005. p. 162.
[9] Schijve J. Fatigue of structures and materials. Dordrecht, Netherlands: Kluwer Academic Publishers; 2001. p. 68–72.
[10] Smith M, Fisher F, Romios M, Es-Said OS. On the redesign of a shear pin under cyclic bending loads. Eng Fail Anal 2007;14:138–46.
[11] Shigley JE. Mechanical engineering design. Tokyo: McGraw-Hill Kogakusha; 1977. p. 196.
[12] Pilkey WD, Pilkey DF. Peterson's stress concentration factors. 3rd ed. New York: John Wiley & Sons; 2008. p. 159.