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Full Length Research Paper

Fatigue failure fault prediction of truck rear axle housing excited by random road roughness

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A premature fatigue failure that occurred prior to the expected load cycles during the vertical fatigue tests of a truck rear axle housing prototype was studied. A seven degree of freedom dynamic model of truck was presented and vibrating equations on automobile as a whole was also built based on the model. Random road roughness data in time history that is generated by random phase method were applied to vibration differential equations to acquire dynamic load imposed on the rear axle housing. Fatigue failure of the rear axle housing finite element model was predicted after the dynamic load was imposed on it. The simulation results showed that there were premature fatigue failure regions at Banjo transition of the rear axle housing. Results provided from tests were compared with the analyses. The simulation results were same as the fatigue tests. Design enhancement solutions were proposed to increase the fatigue life of the housing according to the simulation and test results.

Key words: Fatigue failure, finite element analysis, automobile dynamics, solid axle, rear axle housing.

INTRODUCTION

Due to their higher loading capacity, solid axles are typically used in the commercial vehicles. Solid axles are loaded with dynamic load when vehicle runs on road. Dynamic stress is generated in the solid axles. The dynamic load is the major cause leading to solid axle's fatigue failure when vehicle runs high-speed or carries large weight (Zheng et al., 2009). Load bearing of the solid axle accounts for 60% of the whole vehicle's load bearing. Thus the designed solid axle must have the ability to resisting fatigue failure.

In this paper, according to premature fatigue failure of a truck's rear axle housing, a seven degree of freedom dynamical model of vehicle was constructed, and the vehicle vibration differential equations based on the model were presented also. According to the road power spectrum data issued in national standard, a random phase method was used to generate random road roughness data of time history. Input the random road roughness data into the vibration differential equations, then the dynamic load generated in solid axle could be acquired. The dynamic load was applied into the rear axle housing finite element model and fatigue failure of the rear axle housing was analyzed. Results provided from tests were compared with the analyses. It showed that the simulation results were correct. In this paper, the reasonable solutions preventing the premature fatigue failure of the rear axle housing were presented also.

CONSTRUCTION OF VEHICLE DYNAMICAL MODEL

At present, when designers design the vehicle solid axle housing, the dynamic load generated in axle housing is regarded as static load or equivalent static load and the influence of random road roughness is not included generally. This processing method is applicable to a vehicle running slow and loading low. When vehicle runs high-speed or carries heavy load, the inertia load caused by vibration and impact load increase greatly. It can not reflect the actual load of vehicle caused by road roughness if still use static load model. For this reason, when design a new type solid axle, the influence of road roughness and variation of the loading must to be taken into consideration(Gao et al., 2008; Cong et al., 2009).

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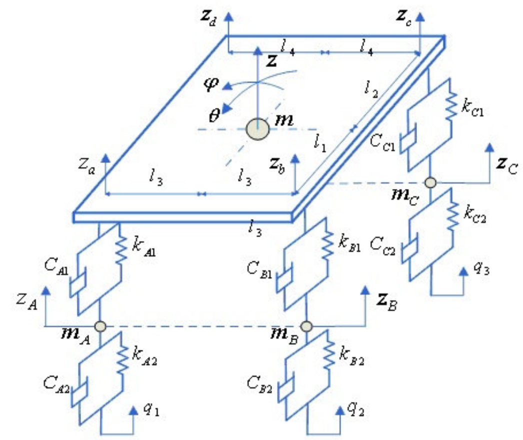

Figure 1. Seven degree of freedom of a truck.

In Figure 1, m is suspension mass. m_A and m_B are left and right front wheel mass. m_C and m_D are right and left rear wheel mass. l_1 and l_2 are distance between front axle and suspension center of mass and distance between rear axle and suspension center of gravity. $2l_3$ and $2l_4$ are front wheel-track and rear wheel-track. z is vertical displacement of suspension center of mass. z_A and z_B are vertical displacements of left and right front wheel. z_C and z_D are vertical displacements of right and left rear wheel. θ is pitch angle of suspension mass around center of gravity. φ is roll angle of suspension mass around center of gravity. k_{A1} and k_{B1} are stiffness coefficients of left and right front suspension. k_{C1} and k_{D1} are stiffness coefficients of right and left rear suspension. c_{A1} and c_{B1} are damping coefficients of left and right front suspension. c_{C1} and c_{D1} are damping coefficients of right and left rear suspension. k_{A2} and k_{B2} are stiffness coefficients of left and right front wheel. k_{C2} and k_{D2} are stiffness coefficients of right and left rear wheel. c_{A2} and c_{B2} are damping coefficients of left and right front wheel. c_{C2} and c_{D2} are damping coefficients of right and left rear wheel. q_1 and q_2 are road surface irregularity excitation of left and right front wheel. q_3 and q_4 are road surface irregularity excitation of right and left rear wheel.

DYNAMICAL MODEL OF VEHICLE

Three assumptions are given when vehicle vibration model is constructed.

Supposing that the body is a rigid body, the body weight and other weights supported by suspension are simplified as a rigid body which only has lumped mass.

The vehicle keeps constant speed to run in straight line and the wheels keep contact with road all the time.

Suspension stiffness and tyre stiffness are linear relationship with displacement. Suspension damping and tyre damping are linear with relative velocity.

Figure 1 shows the constructed seven degree of freedom model of vehicle. The seven freedom degrees are vertical body's jumping, front and back pitching, left and right rolling and four wheels' vertical jumping.

VIBRATION RESPONSE OF VEHICLE

In this paper, dynamics equations of the seven degree of freedom were deduced by using of Newton's law of motion equations.

If pitch angle θ and roll angle φ are small, the vertical displacements of body's four terminal vertexes have the following relationships.

$$\begin{aligned}
z_a &= z - l_1 \cdot \theta + l_3 \cdot \varphi \\
z_b &= z - l_1 \cdot \theta - l_3 \cdot \varphi \\
z_c &= z + l_2 \cdot \theta - l_4 \cdot \varphi \\
z_d &= z + l_2 \cdot \theta + l_4 \cdot \varphi
\end{aligned} \tag{1}$$

The dynamic load of Z axle caused by suspension mass m is :

$$\begin{aligned}
& m \cdot \ddot{z} = c_{A1} \cdot \left(\dot{Z}_A - Z_a\right) + k_{A1} \cdot (Z_A - Z_a) \\
& + c_{B1} \cdot \left(\dot{Z}_B - Z_b\right) + k_{B1} \cdot (Z_B - Z_b) + c_{C1} \cdot \left(\dot{Z}_C - Z_c\right) \\
& + k_{C1} \cdot (Z_C - Z_c) + c_{D1} \cdot \left(\dot{Z}_D - Z_d\right) + k_{D1} \cdot (Z_D - Z_d)
\end{aligned} \tag{2}$$

The Equation (3) is gotten from the Equations (1) and (2).

$$\begin{aligned}
& m \cdot \ddot{z} + (c_{A1} + c_{B1} + c_{C1} + c_{D1}) \cdot \dot{z} - c_{A1} \cdot \dot{z}_A \\
& - c_{B1} \cdot \dot{z}_B - c_{C1} \cdot \dot{z}_C - c_{D1} \cdot \dot{z}_D \\
& - [(c_{A1} + c_{B1}) \cdot l_1 - (c_{C1} + c_{D1}) \cdot l_2] \cdot \dot{\theta} \\
& + [(c_{A1} - c_{B1}) \cdot l_3 - (c_{C1} - c_{D1}) \cdot l_4] \cdot \dot{\varphi} \\
& + (k_{A1} + k_{B1} + k_{C1} + k_{D1}) \cdot z - k_{A1} \cdot z_A \\
& - k_{B1} \cdot z_B - k_{C1} \cdot z_C - k_{D1} \cdot z_D \\
& - [(k_{A1} + k_{B1}) \cdot l_1 - (k_{C1} + k_{D1}) \cdot l_2] \cdot \theta \\
& + [(k_{A1} - k_{B1}) \cdot l_3 - (k_{C1} - k_{D1}) \cdot l_4] \cdot \varphi = 0
\end{aligned} \tag{3}$$

The Equations (4) to (9) can be gotten from analyzing the body's pitching, rolling and four wheels' vertical jumping.

$$\begin{aligned}
& J_1 \ddot{\theta} - \left[(c_{A1} + c_{B1}) \cdot l_1 - (c_{C1} + c_{D1}) \cdot l_2\right] \cdot \dot{z} \\
& + c_{A1} \cdot l_1 \cdot \dot{z}_A + c_{B1} \cdot l_1 \cdot \dot{z}_B - c_{C1} \cdot l_2 \cdot \dot{z}_C - c_{D1} \cdot l_2 \cdot \dot{z}_D \\
& + \left[(c_{A1} + c_{B1}) \cdot l_1^2 + (c_{C1} + c_{D1}) \cdot l_2^2\right] \cdot \dot{\theta} - \\
& \left[(c_{A1} - c_{B1}) \cdot l_1 \cdot l_3 + (c_{C1} - c_{D1}) \cdot l_2 \cdot l_4\right] \cdot \dot{\varphi} \\
& - \left[(k_{A1} + k_{B1}) \cdot l_1 - (k_{C1} + k_{D1}) \cdot l_2\right] \cdot z \\
& + k_{A1} \cdot l_1 \cdot z_A + k_{B1} \cdot l_1 \cdot z_B - k_{C1} \cdot l_2 \cdot z_C - k_{D1} \cdot l_2 \cdot z_D \\
& + \left[(k_{A1} + k_{B1}) \cdot l_1^2 + (k_{C1} + k_{D1}) \cdot l_2^2\right] \cdot \theta \\
& - \left[(k_{A1} - k_{B1}) \cdot l_1 \cdot l_3 + (k_{C1} - k_{D1}) \cdot l_2 \cdot l_4\right] \cdot \varphi = 0
\end{aligned} \tag{4}$$

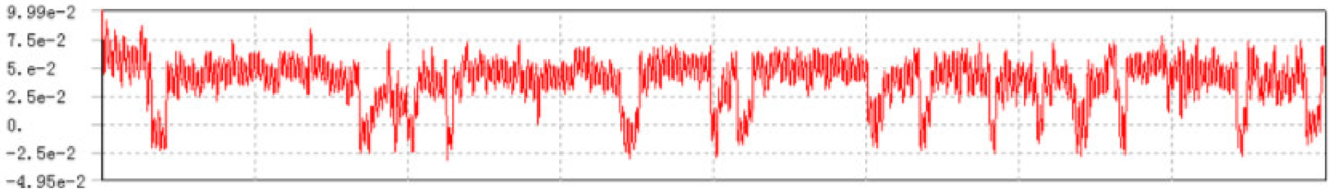

Figure 2. F level random road roughness sample.

$$\begin{aligned}&J_2\ddot{\varphi}+\left[\left(c_{A1}-c_{B1}\right)\cdot l_3-\left(c_{C1}-c_{D1}\right)\cdot l_4\right]\cdot\dot{z}\\&-c_{A1}\cdot l_3\cdot\dot{z}_A+c_{B1}\cdot l_3\cdot\dot{z}_B+c_{C1}\cdot l_4\cdot\dot{z}_C-c_{D1}\cdot l_4\cdot\dot{z}_D\\&-\left[\left(c_{A1}-c_{B1}\right)\cdot l_1\cdot l_3+\left(c_{C1}-c_{D1}\right)\cdot l_2\cdot l_4\right]\cdot\dot{\theta}\\&+\left[\left(c_{A1}+c_{B1}\right)\cdot l_3^2+\left(c_{C1}+c_{D1}\right)\cdot l_4^2\right]\cdot\dot{\varphi}\\&+\left[\left(k_{A1}-k_{B1}\right)\cdot l_3-\left(k_{C1}-k_{D1}\right)\cdot l_4\right]\cdot z\\&-k_{A1}\cdot l_3\cdot z_A+k_{B1}\cdot l_3\cdot z_B+k_{C1}\cdot l_4\cdot z_C-k_{D1}\cdot l_4\cdot z_D\\&-\left[\left(k_{A1}-k_{B1}\right)\cdot l_1\cdot l_3+\left(k_{C1}-k_{D1}\right)\cdot l_2\cdot l_4\right]\cdot\theta\\&+\left[\left(k_{A1}+k_{B1}\right)\cdot l_3^2+\left(k_{C1}+k_{D1}\right)\cdot l_4^2\right]\cdot\varphi=0\end{aligned}\tag{5}$$

$$\begin{aligned}&m_A\ddot{z}_A-c_{A1}\cdot\dot{z}+(c_{A1}+c_{A2})\cdot\dot{z}_A+c_{A1}\cdot l_1\cdot\dot{\theta}\\&-c_{A1}\cdot l_3\cdot\dot{\varphi}-k_{A1}\cdot z+(k_{A1}+k_{A2})\cdot z_A\\&+k_{A1}\cdot l_1\cdot\theta-k_{A1}\cdot l_3\cdot\varphi=c_{A2}\cdot\dot{q}_1+k_{A2}\cdot q_1\end{aligned}\tag{6}$$

$$\begin{aligned}&m_B\ddot{z}_B-c_{B1}\cdot\dot{z}+(c_{B1}+c_{B2})\cdot\dot{z}_B+c_{B1}\cdot l_1\cdot\dot{\theta}\\&+c_{B1}\cdot l_3\cdot\dot{\varphi}-k_{B1}\cdot z+(k_{B1}+k_{B2})\cdot z_B\\&+k_{B1}\cdot l_1\cdot\theta+k_{B1}\cdot l_3\cdot\varphi=c_{B2}\cdot\dot{q}_2+k_{B2}\cdot q_2\end{aligned}\tag{7}$$

$$\begin{aligned}&m_C\ddot{z}_C-c_{C1}\cdot\dot{z}+(c_{C1}+c_{C2})\cdot\dot{z}_C-c_{C1}\cdot l_2\cdot\dot{\theta}\\&+c_{C1}\cdot l_4\cdot\dot{\varphi}-k_{C1}\cdot z+(k_{C1}+k_{C2})\cdot z_C\\&-k_{C1}\cdot l_2\cdot\theta+k_{C1}\cdot l_4\cdot\varphi=c_{C2}\cdot\dot{q}_3+k_{C2}\cdot q_3\end{aligned}\tag{8}$$

$$\begin{aligned}&m_D\ddot{z}_D-c_{D1}\cdot\dot{z}+(c_{D1}+c_{D2})\cdot\dot{z}_D-c_{D1}\cdot l_2\cdot\dot{\theta}\\&-c_{D1}\cdot l_4\cdot\dot{\varphi}-k_{D1}\cdot z+(k_{D1}+k_{D2})\cdot z_D\\&-k_{D1}\cdot l_2\cdot\theta-k_{D1}\cdot l_4\cdot\varphi=c_{D2}\cdot\dot{q}_4+k_{D2}\cdot q_4\end{aligned}\tag{9}$$

ROAD ROUGHNESS

According to GB/T 7031-2005 / ISO 8608 1995, mechanical vibration spectrum of road surface roughness measurement data, power spectrum density of road $G_q(n)$ can be generated using the following formula.

$$G_q(n)=G_q(n_0)\left(\frac{n}{n_0}\right)^{-w}\tag{10}$$

In the formula, n is space frequency (reciprocal of wave(m^{-1}). n_0 is reference space frequency (n_0 = 0.1 m^{-1}). w is exponent of fit power spectrum density. $G_q(n_0)$ is road spectrum of reference frequency n_0.

According to the given road spectrum in GB/T 7031-2005 / ISO 8608, phase random method was used to generate random road roughness in time-history. The method uses the formula to generate random road roughness.

$$q(t)=\sum_{i=1}^{m}\sqrt{4G_q(n_i)\Delta n}\cdot\cos(n_i vt-\theta_i)\tag{11}$$

In this equation, n_i is space frequency. Δn is increment of space frequency. θ_i is the random number which distributes uniformly in $[0,2\pi]$. v is speed of vehicle. t is vehicle travel time.

The software Matlab is used to simulate the road roughness when vehicle travels under 50 km/h. The F level road roughness got from the simulation is showed in Figure 2.

BUILDING FINITE ELEMENT MODEL OF AXLE HOUSING

3D Solid model of axle housing

The housing essentially consists of two equivalent thin walled shells, which have a uniform thickness of 8 mm and welded along the neutral axis of the rear axle. An erection loop is welded on the housing's front side to increase stiffness. Differential is fixed in the housing by bolts. For sealing reasons, a dome is welded to the rear side. The CAD model of the axle housing was imported into ANSYS Workbench V11.0 preproccessing environment to constitute the finite element model. To building 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. Finite element model consisted of 448,003 elements.

Because of the vertical acceleration of lumped mass of the vehicle body due to the road surface roughness, maximum dynamic loading on each coil spring seat is estimated about twice as much as static loading. According to the dynamics analysis aforementioned, vertical loading range was obtained as 126 to 1670

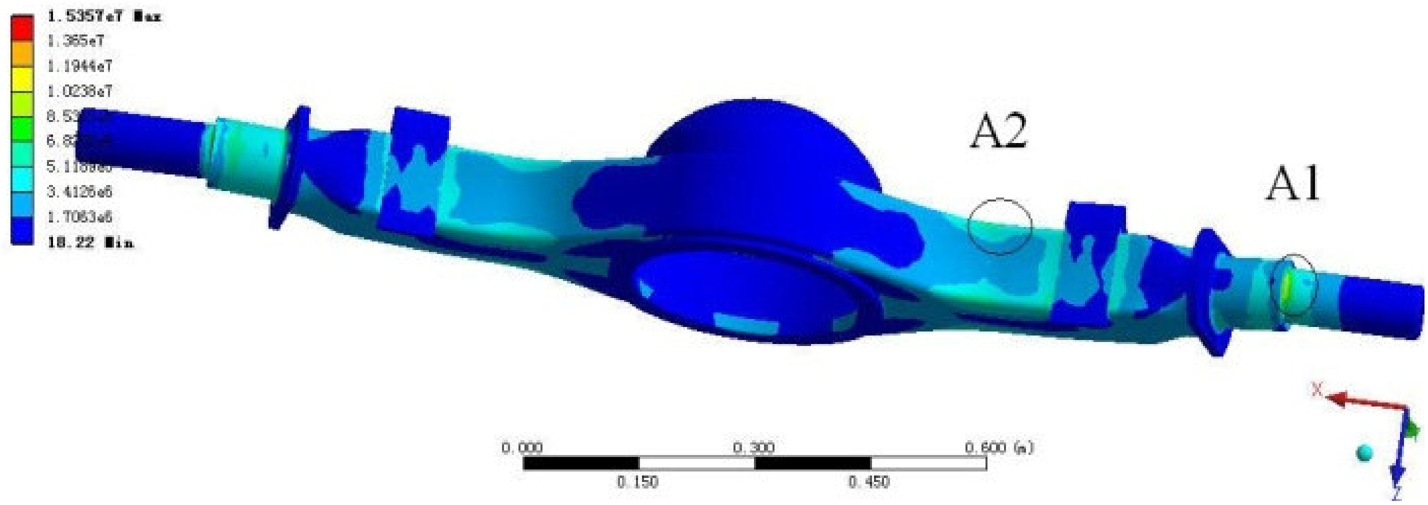

Figure 3. Equivalent von Mises stress distribution of the housing prototype.

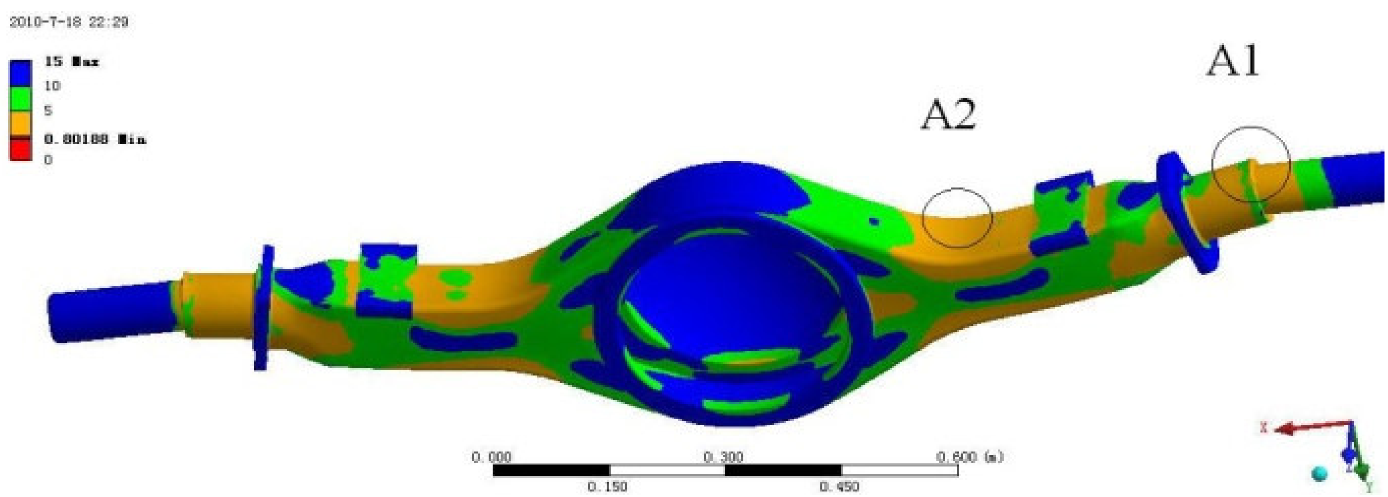

Figure 4. Factor of safety distribution of the housing prototype.

kg on each coil spring seat under F level road roughness. The finite element model of the real axle housing could be analyzed using ANSYS software after constraints and loading were applied on the model.

Finite element analysis and results

Figure 3 shows equivalent von Mises stress distribution provided from the FE analysis. Results show that there are tensile stress concentrated regions A1 and A2 at transition area of axle spindle mountings and 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 Figure 4. The calculated maximum von Mises stress is 15.3 MPa far below the material's yielding stress. This means that housing prototype satisfies the safety conditions for maximum loading if it is exerted statically.

FATIGUE LIFE PREDICTION

When vehicle is running on the road, the axle housing is inspired by dynamic loading. Thus fatigue analysis is important to axle housing. Fatigue endurance limit is given as (Topac et al., 2009)

$$S_e' = 0.504 S_{ut} \tag{12}$$

Because the ultimate tensile strength of steel S_{ut} is less

Figure 5. Fatigue failure of a test sample.

than 400 Mpa, the fatigue endurance limit of steel is about 10^6 cycles. $S_e^{'}$ stands for the stress life endurance limit of ideal laboratory samples. In order 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 (Smith M et al., 2007). S_e is given as

$$S_e = k_a k_b k_c k_d k_e S_e^{'} \quad (13)$$

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

$$k_a = aS_{ut}^b \quad (14)$$

Size factor k_b can be assumed as 0.75. Load factor k_c is given as 1 for bending. Temperature factor k_d is 1 for the range of the ambient temperature of 0 to 250℃ (Shigley, 1977). Because there are stress concentrated regions, in addition to the modifying factors aforementioned, a fatigue strength modifying factor k_e must be taken into account that is related to fatigue stress concentration factor k_f . Hence k_e is calculated as

$$k_e = 1/k_f \quad (15)$$

For safety reason, k_f can be assumed as to be equal to factor of static stress concentration K_t (Shigley et al., 1989). K_t cannot be derived from the standard literature Because of the dimensions and shape complexity of the axle housing, however, it is defined as

$$K_t = \frac{\sigma_{peak}}{\sigma_{nominal}} \quad (16)$$

wdhere σ_{peak} is the peak stress and $\sigma_{nominal}$ is the nominal stress.

To calculate $\sigma_{nominal}$, the rear axle housing is assumed as a simple beam which has a uniform box profile cross-section along the longitudinal axis X and subjected to pure bending moment.

$$\sigma_{\text{nominal}} = \frac{M}{Z} \quad (17)$$

where M is bending moment and Z is the section modulus of the critical cross-section.

The fatigue safe coefficient distribution is showed in Figure 4 derived from finite element analysis. It is estimated that crack initiation can occur at the region A2 of the outer shell surface and the symmetric region at about 2.8×10^5 cycles, which is lower than the expected minimum fatigue life of 5×10^5 cycles. Here the obtained minimum safe factor value is 0.80. This means, at the two regions A2 and its symmetric region fatigue crack can initiate before 5×10^5 load cycles as observed in the vertical fatigue tests. The results are in agreement with the results of vertical fatigue tests. The fatigue crack regions derived from vertical fatigue tests are showed in Figure 5.

Enhancement of the fatigue life of the axle 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 A2 and its symmetric region, the housing satisfies the infinite life criteria. An increase of sheet metal thickness causes an unnecessary weight increase. For example, a thickness increase of 0.5 mm enhances the fatigue limit of the housing material at the critical regions up to more than 5×10^5 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 and can influence vehicle fuel economy and dynamic characteristic. Therefore it is not a practical solution. Smoother transition geometry may offer an enhanced fatigue life without any weight increase. In addition, the use of the ring on the critical regions can decrease stress concentration to enhance fatigue life.

Conclusion

In order to analyze premature fatigue failure of a truck's rear axle housing, a seven degree of freedom dynamic model of the truck was presented. Based on the model, vibration differential equations were given. Random road roughness data of time history that is generated by random phase method were applied to vibration differential equations to acquire dynamic load imposed on the drive axle housing according to road power spectrum in the national standard. The acquired dynamic load was applied to the finite element model of axle housing to predict fatigue failure. In the analyses, in which the vertical fatigue test procedure was simulated, stress concentrated regions were predicted at the Banjo transition area and transition area of axle spindle mountings. The regions in which the fatigue cracks originated were well-matched with the results of the analyses. By using the method in this paper, the location of the failure can be predicted.

In order to solve the problem of enhancing fatigue life of axle housing, 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 enhance the fatigue life, which can satisfy minimum design criteria.

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