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Static and dynamic analysis of front axle housing of tractor using finite element methods

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Abstract

MT250D Mitsubishi Tractor with 25hp power is a small agriculture tractor which is used to do light agricultural operations. In order to do off road operations, a front mounted mechanical shovel with total weight 400kg that including hydraulic equipment for example the weight of hydraulic cylinder was added on the front of the tractor and thus, the static weight on the front axle housing was increased. In this study, to do modeling of housing, Solid Works software (Version2010) was used. In order to use finite element method for static and dynamic analysis, Cosmos Works Software (Version 2010) was used. Finite element analysis results showed that the maximum stress of 238.84MPa is applied on the upper housing. According to Von- Misses theory, the value of maximum applied stress and allowable stress, the safety factor of 1.05 was obtained which is less than the required value. The first four natural frequencies of housing were found as 678.54, 720.29, 908.78 and 1877 Hz, respectively. The obtained factor of safety is very low and obviously this value decreases under dynamic loading conditions of field operation. The present study clearly indicates that the front axle housing of MT250D Mitsubishi tractor is not strong enough to be mounted on a tractor. There is a need to optimize the existing design of the front axle housing, if we want to use a mechanical shovel.

Keyword: Stress analysis, modeling, finite element, differential, Mitsubishi tractor

Abbreviations: σ _stress (MPa); $\sigma_1,\sigma_2,\sigma_3$ _principal stresses (MPa); f_ frequencies (Hz); V_i_ tractor speed (m/s); g _Gravity(m^2/s); h_ fall height (m); T_r_ rise time (s); h' _ tire compression (m); F_{max}_ maximum value of dynamic load (N); M_f_ applied static mass (kg); Δv_i_ tractor speed (m/s); Δt_ rise time (s).

Introduction

Helping with developed technologies and design software which integrated in new generation of computers, designs are getting easier and reliable. Designers can design own products in virtual screen and they can evaluate working condition of the products by simulating techniques using the computers. Yaghoubi et al. (2010) for design and simulation a new universal joint used the CAD softwares (Comosworks , Visual NASTRAN and Autodesk Inventor).Today three-dimensional (3D) modeling and finite elements method applications are getting so widespread in the industry. Many of 3D modeling and finite elements application samples can be seen on different engineering disciplines (Gunay 1993).

Especially in the small- and middle-scale agricultural machinery industry, insufficient technical knowledge, use of new technology, and new design features can cause problems such as breakdowns and failures during field operations. Failure of machinery devices is one of the major problems in engineering (Yılmaz et al., 2001). There are a few publications on the failure of agricultural machinery, and some reports on failure of agricultural machinery have been published. For example, Akinci et al. (2005) studied the failure of a rotary tiller spur gear. Nanaware and Pable (2003) investigated fatigue cracks on the rear axle of a tractor. Generally, before any additional load on the tractor, the main parts of that must be resistant. Each machine is designed for a specific purpose. If you use the machine over its ability, some of its component may face additional stress. This additional stresses may cause the parts to failure or face permanent deformation. Tractor is one of the multifunctional machines that is used for different agricultural operations. Because the tractors work on difficult conditions than other machines, components of that should have high safety of factor. If for an special order, a new mechanism should be installed on the tractor, its main components must be resistant to tolerate additional stress and loads. The front axle housing of a tractor is one of the major and very important components and needs to be designated carefully, science this part experiences the worst load condition such as static and dynamic loads.

Leon et al. (2000) used experimental and numerical methods, for the stress analysis of a frontal truck axle beam. The results obtained by finite element method were verified experimentally using photo stress. Mahanty et al. (2001) performed an experimental and numerical analysis of a tractor's front axle. Based on finite element analysis results redesign was carried out for the front axle for weight optimization and easy manufacturability. Five different models were proposed based on ease of manufacture and weight reduction. Maly and Bazzaz (2003) used experimental and numerical methods, for design change from casting to welding for an axle casing. Mehmet topakci (2008) used finite element method to obtain the stress distribution and factor of safety on transmission gear of rotary tiller. After evaluating of simulation results, stress distributions on gears showed that gears working without failure according to yield stress of gear's materials. Topac et al. (2009) analyzed premature fatigue failure of a truck rear axle housing prototype using finite element method. In the analysis, in which the vertical fatigue test procedure was simulated, stress concentrated regions were predicted at the banjo transition area.

The main objective of this research was to analyze the front axle housing of MT250D four-wheel drive tractor under static and dynamic loading conditions and to find the factor of safety and maximum stress, whether the housing can tolerate the additional stresses of installation of a mechanical shovel on the front of tractor or not.

Material and methods

The CAD model of the front axle housing of MT250D tractor was created. The housing essentially consists of two equivalent thin walled shells, which have a uniform thickness of 6 mm. Two sides of the housing are connected to each other with 8 bolts. For modeling and stress analysis of the housing, the commercial 3D modeling and finite element package Solidworks and Cosmos Works V2010 were used. To build the finite element model, the housing was meshed using SOLID45, 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 29 nodes having three translational DOF at each node (Cosmos works help, V2010) completely bonded contact was chosen as the contact condition for all connective surfaces. Finite element model consisted of 40,241elements and 73,214nodes. Fig .1 and fig .2 show the Front axle housing of commercial tractor and 3D-model, respectively.

Fig 1. Front axle housing of Mitsubishi tractor

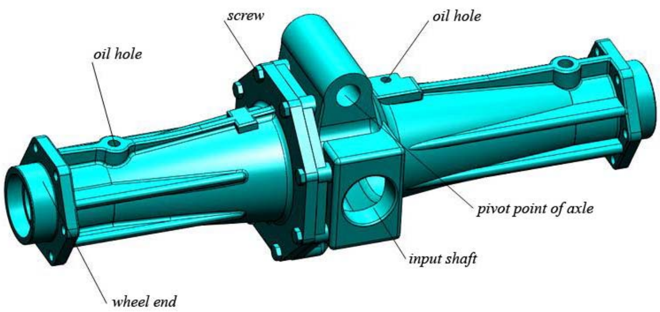

Fig 2. Complete CAD model of the housing

Housing material

Shells are manufactured by the casting process and their thickness is 6 mm. The mechanical properties of the material which were obtained from the supplier are given in Table 1.

Loads on front axle beam

All vehicles are subjected to both static and dynamic loads. In this study, static loads, drop test and also the first four natural frequencies were studied on the housing. Front axle housing of tractor is considered as a support for front wheels, bodywork or super-structure etc. Static loads that applied on the front axle housing are as follows: vertical bodywork loads, weight of the bucket, weight of the boom, weight of hydraulic cylinders of loader and etc. These forces were determined by experimental and theoretical methods.

Analysis

The commercial finite element package Solidworks version 2010 was used for the solution of the problem. The geometric model for the front axle was created based on the drawings provided by tractor manufacture. The front axle housing is modeled with two hexahedral three dimensional elements. The individual components have been coupled together so that there is no free motion between components. The next step was to define the boundary conditions. All degrees of freedom are constrained at the connection plates of the model. The next step was the definition of the loads that have to be previously described and defined.

The static analysis of front axle was carried out and elements with average size of 9 mm were used. The meshed model of front axle housing of MT250D Mitsubishi tractor is shown in Fig. 3. After obtaining the solution, the results of analysis can be reviewed using post processing to determine minimum value of factor of safety and maximum induced stress and its location.

FE analysis was used to predict the exact location of the regions where compressive stress concentrations were seen. Stress analysis was carried out using CosmosWorks commercial FEA software on a 2.56 GHz 2 Duo-core Intel Processor SONY VPCCW Workstation.

Calculating safety factor

In designing parts to resist failure, it is assumed that the internal stresses do not exceed the strength of the material. If the material to be used is brittle, then it is the yield strength that designer is usually interested in, because a little deformation would constitute failure. The distortion-energy theory is also called the Von-Mises theory, which is the most suitable theory to be used in materials (Shigley and Mischke 1989) .Von-Mises stress, is calculated by using the formula Von-Mises stress as follows:

$$\sigma=\left(\frac{(\sigma_1-\sigma_2)^2-(\sigma_2-\sigma_3)^2-(\sigma_1-\sigma_3)^2}{2}\right)^{\frac{1}{2}} \quad (1)$$

Where $\sigma_1, \sigma_2, \sigma_3$ are principal stresses associated with the three principal directions.

Table 1. Mechanical properties of front axle housing of MT250D 4WD tractor.

material	Modulus of elasticity E	Poisson's ratio ϑ	Yield strength σ_y	Tensional strength σ_t	Density ρ
ASTM A-220	162 GPa	0.26	280 MPa	448 MPa	7300 kg.m^{-3}

Modal analysis

The eligibility of the static strength of front axle housing cannot prove that it will never break. In reality, the front axle housing is loaded with some kinds of stimulations, which result in breakages such as resonance, fatigue and etc. It is very significant to analyze dynamic characteristic for the chosen modified design of front axle housing. Modal analysis of front axle housing is performed by Block Lanczos method of finite element software CosmosWorks V 2010.

Considering the typical row spacing range from 150 to 400 mm, the field ground profile is modeled with a sinusoidal wave with a wavelength of 0.3 m and amplitude of 0.15 m. considering the tractor speed ranges from 2 to 11 km/h, the maximum value of corresponding frequency is calculated as below:

$$f = \frac{11}{3.6 \times 0.3} = 10.18\ Hz \quad (2)$$

Therefore, a frequency range from zero to 11 Hz is considered for frequency range of harmonic loads.

Transient analysis

In transient analysis, front axle housing of tractor is considered as a beam that was subjected to one bodyworks loads (Fig. 4). A dynamic loading condition consists of two load steps (Fig. 5). In the first load step, the force increased during the rise time and the value of force at the end of the first step, reached the maximum value. During the second load step, value of load remained constant at maximum value. Rise time and maximum value of dynamic load applied on the housing were calculated by considering some assumptions. Assuming the tractor falls from 0.3 m height, its speed when reaching the ground, was calculated as follows:

$$V_i = \sqrt{2gh} = \sqrt{2 \times 9.81 \times 0.3} = 2.42\,m/s \quad (3)$$

Where, V_i is the tractor speed when reaching the ground in meters per second and h is the fall height in meters. Assuming that the tractor tires are compressed by 0.05 m, the rise time can be calculated using following equation:

$$T_r = \frac{h}{V} = \frac{2h'}{V} = 0.041\ s \quad (4)$$

Where T_r is the rise time in second; h' is the tire compression in meter and is the average speed during rise time in meter per second. Above equation includes the assumption that the final speed of the tractor during rise time is zero.

To calculate the maximum value of dynamic load, the following equation can be used:

$$F_{max} = m_f \times \frac{\Delta v_i}{\Delta t} = m_f \times \frac{2.42}{0.041} = 59.024 m_f = 6.01 m_f.g \quad (5)$$

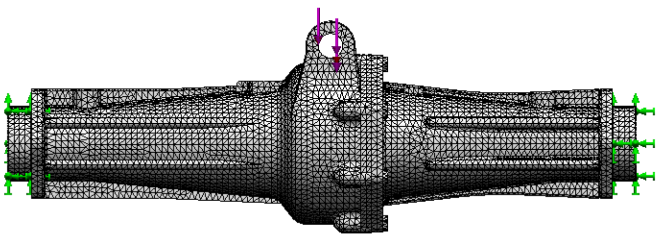

Fig 3. Finite element model of the housing

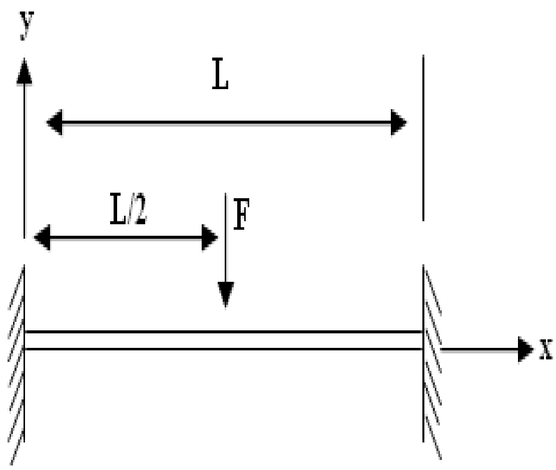

Fig 4. Beam model of front axle subjected to the transient loads

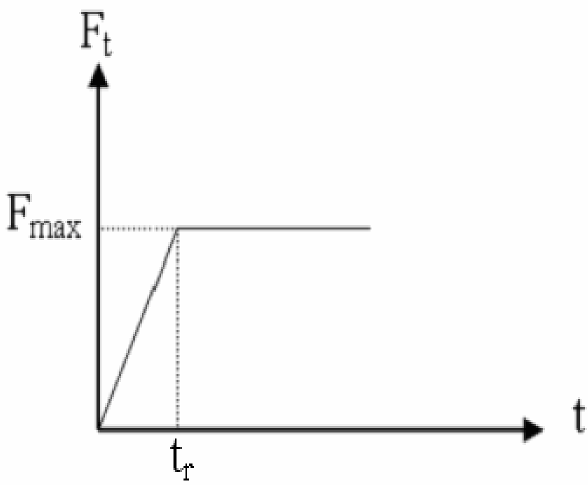

Fig 5. Load versus time curve in dynamic loading

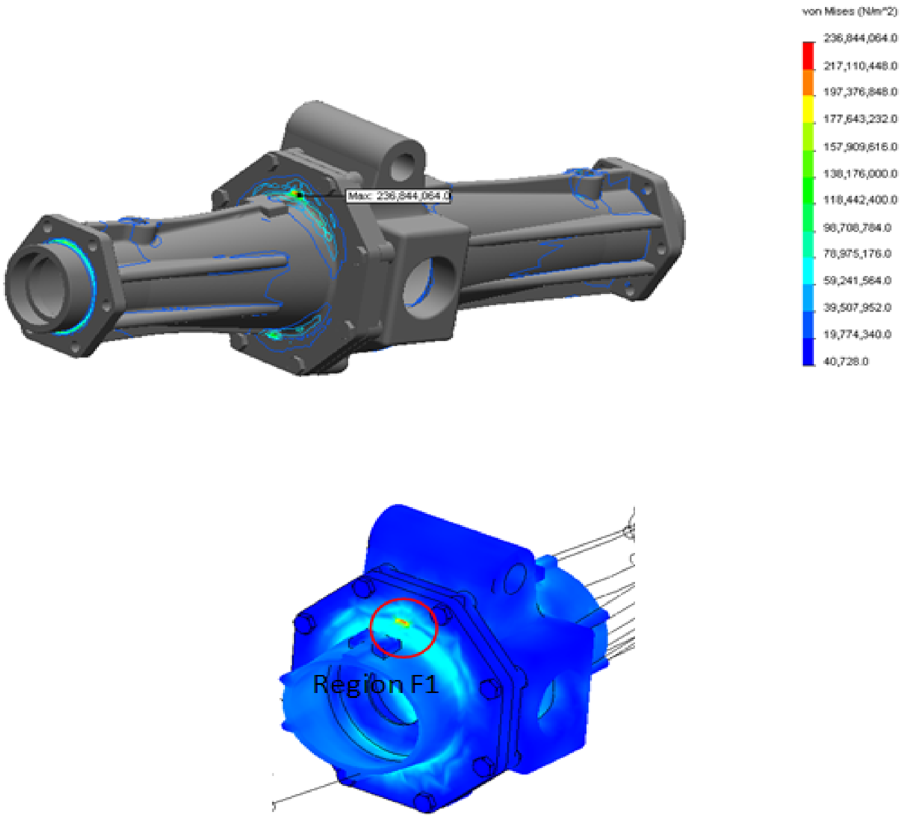

Fig 6. Overall stress distribution on the housing.

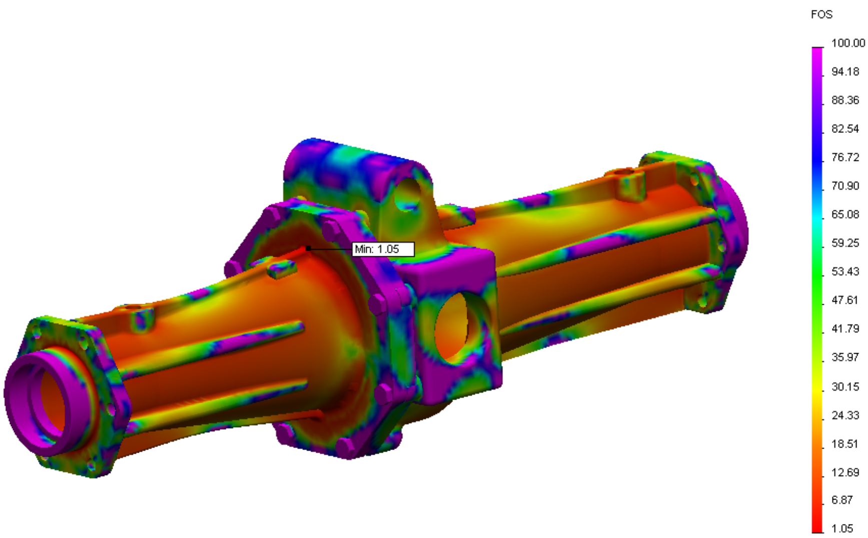

Fig 7. Distribution of factor of safety on housing

Where F_{max} is the maximum value of dynamic load, m_f is the applied static mass on the front axle and g is the acceleration due to gravity. This equation means that the dynamic load is 6.01 times as the static load.

In dynamic loading conditions, the vertical acceleration of lumped mass of the vehicle body due to the road surface roughness can be estimated as six times as much as the acceleration of gravity. This means that the maximum dynamic loads can be increased to six times as much as the corresponding loads in static loading conditions. The amount of dynamic acceleration for tractors and agricultural machinery was considered as 6g. (Mirehea 2005). This issue approximately verifies the mentioned assumptions and calculations for obtaining the rise time and maximum value of dynamic loads applied on the front axle. In this study to increase the design factor of safety, the maximum value of dynamic load was considered as six times as much as the applied static load. Subsequently, the defined boundary conditions applied to the model were analyzed using post processing. The results of analysis were used to identify the value of maximum induced stress.

Fig 8. The first four mode shapes of housing

Results and discussion

The results of the modal analysis that was meshed using elements with average size of 9 mm are given in Fig. 3. As seen, the maximum Von-Mises stress appears on the upper box and near to the left connection plate as shown by the arrow in the Fig. 6. Fig.6 shows equivalent von Mises stress distribution provided from the FE analysis. Results showed that there is a compressive concentrated region F_1 at banjo transition area of the carrier mounting side of the up shell. The calculated maximum von Mises stress is: σ_{max} = 236.84 MPa. According to distortion-energy theory, allowable stress in order to avoid fracture is equal to yield stress strength. Factor of safety can be calculated by dividing yield stress to maximum Von-Mises stress. Fig.7 shows safety factor distribution and the place of the minimum factor of safety on the upper of shell. The first four natural frequencies of the housing are showed in Fig.8.The dynamic characteristics are much better, because the natural frequencies are all greater than the excitation frequency range of 0.33 ~ 28.3Hz from the ground. Therefore the condition of resonance was not encountered (Wei 2001).

From the value of maximum induced stress and allowable stress, a factor of safety value was calculated 1.05 and found to be less than required value. Calculated value of factor of safety is very low and obviously this value decreases under dynamic loading conditions of field operation. The present study clearly indicates that the front axle housing of MT250D Mitsubishi tractor is not strong enough to be installed on the tractor in order to install a mechanical shovel on front of tractor to do road-way operation. There is a need to optimize the existing design of the front axle housing of MT250D Mitsubishi tractor.

Suggested modifications to increase strength and reliability are as follow:

1. Increase the thickness of upper box
2. Design a lightweight mechanical shovel with low capacity of bucket
3. Increasing the shell thickness in areas where stress concentration occurs

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