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Experimental Analysis through Design Optimization for DFM and weight Reduction of Fork Diff-Lock Gear Shifter

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Abstract: ***During the process of Milling for v cut Slot minor Cracks are generated in fork Surface, which during induction hardening process converts in to deep cracks and can only be detected during the nondestructive testing like MPI. This leads to higher rejection ratio and incur financial and material losses. The aim of this research is to determine the contribution of v cut slot in forks. The model is designed in Creo 2.0 software package and analyzed using ANSYS. The second design is modeled without v slot and analyzed under the same boundary conditions and compared with previous design.***

Keywords: V slot , Gear shifter fork ,Structural Analysis, ANSYS , FEM

1. Introduction

The transmission is place between the engine and the drive shaft. The transmission is connected through the clutch. The figure 1 shows the basic concept where are the gearbox place in the car system. To understand the basic idea behind a standard transmission, the Figure 1 shows a very simple two-speed transmission in neutral

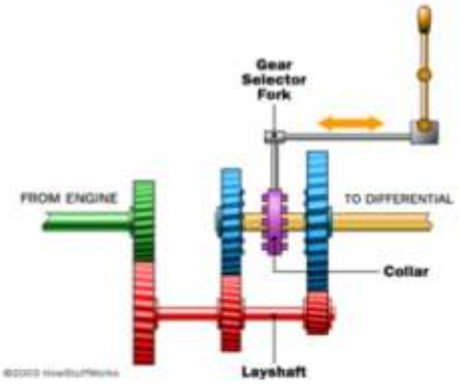

Figure 1: Simple 2 speed transmission

The green shaft comes from the engine through the clutch. The green shaft and green gear are connected as a single unit. (The clutch is a device that lets you connect and disconnect the engine and the transmission. When you push in the clutch pedal, the engine and the transmission are disconnected so the engine can run even if the car is standing still. When you release the clutch pedal, the engine and the green shaft are directly connected to one another. The green shaft and gear turn at the same rpm as the engine.) The red shaft and gears are called the layshaft. These are also connected as a single piece, so all of the gears on the layshaft and the layshaft itself spin as one unit. The green shaft and the red shaft are directly connected through their meshed gears so that if the green shaft is spinning, so is the red shaft. In this way, the layshaft receives its power directly from the engine whenever the clutch is engaged. The yellow shaft is a splined shaft that connects directly to the drive shaft through the differential to the drive wheels of the car. If the wheels are spinning, the yellow shaft is spinning. The blue gears ride on bearings, so they spin on the yellow shaft. If the engine is off but the car is coasting, the yellow shaft can turn inside the blue gears while the blue gears and the layshaft are motionless. The purpose of the collar is to connect one of the two blue gears to the yellow drive shaft. The collar is connected, through the splines, directly to the yellow shaft and spins with the yellow shaft. However, the collar can slide left or right along the yellow shaft to engage either of the blue gears. Teeth on the collar, called dog teeth, fit into holes on the sides of the blue gears to engage them. Figure 2 shows how, when shifted into first gear, the collar engages the blue gear on the right:

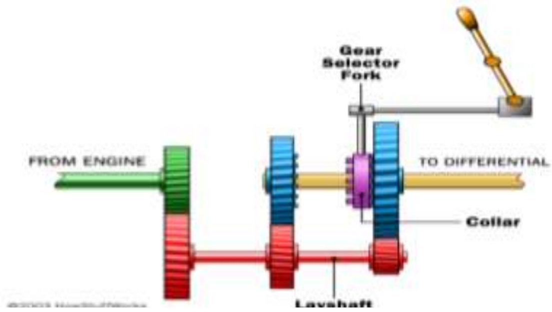

Figure 2: First gear collar engagement

In this picture, the green shaft from the engine turns the layshaft, which turns the blue gear on the right. This gear transmits its energy through the collar to drive the yellow drive shaft. Meanwhile, the blue gear on the left is turning, but it is freewheeling on its bearing so it has no effect on the yellow shaft. When the collar is between the two gears (as shown in the first figure), the transmission is in neutral. Both of the blue gears freewheel on the yellow shaft at the different rates controlled by their ratios to the layshaft.

The 5 speed manual transmission for standard on car today. The internally look like the figure 3. Gear knob inside the car is used to shift the gear. There are three forks controlled by three rods that are engaged by the shift lever. Looking at the shift rods from the top, they look like this in reverse, first and second gear:

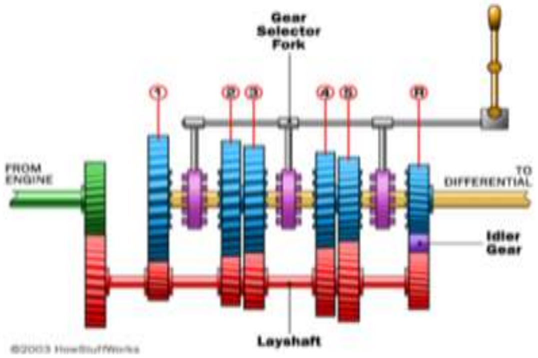

Figure 3:5 speed manual transmission

2. Literature Review

Dogan [1] have investigated the cause of rattling and clattering noise and concluded that torsional vibration is the main reason of vibration. For this analysis, Dogan has used simple gearbox geometry. This geometry consist of only transmission casing. The main advantage of Dogan study is that he has start simulating such a complex geometry of transmission gearbox. He has used EKM simulating program. Abouel has performed similar study on car gearbox. AbouelSeoud et al. [3] have used vibration response analysis method for the analytical analysis of car gearbox system. He has performed analytical and experimental analysis of a car transmission system. By using physical properties, he has calculated the radiation efficiency, and the vibration response was measured.

Vandi et al. [4] have presented the implementation of a simplified engine-driveline model to complete an existing vehicle dynamic model. The engine model is based on maps which are expressed as function of engine speed and load. Nacib et al. [5] have performed the failure analysis of heavy gearbox of helicopters. To prevent break down and accident in helicopters gear fault detection is important. Spectrum analysis and Cepstrum analysis method is used to identify damage gear.

Gordon et al.[6] have studied the source of vibration. A Sports Utility Vehicle with sensor and data acquisition system is used to find the vibration source. This study was focused on vehicle vibration response from road surface features.

kar et al. [7] have used motor current signature analysis (MCSA) and discrete wavelet transform (DWT) for studying the gear vibration. Transmission errors and internal excitation causes vibration and noise problem.

Czech [8] has described the vibroacoustic diagnostics of high-power toothed gears. The presented analysis is a experimental work done in a steel plant.

Singh [9] has done two case studies for the vibro-acoustic analysis of automotive structures. Analytical and experimental results are presented for brief description. In first case passive and adaptive hydraulic engine mounts and in second case welded joints and adhesives in vehicle bodies were considered.

3. Finite Element Analysis

The CAD model is developed in Creo 2.0 which is sketch based, feature based parametric 3d modeling software developed by PTC and is imported in ANSYS as shown in fig 4

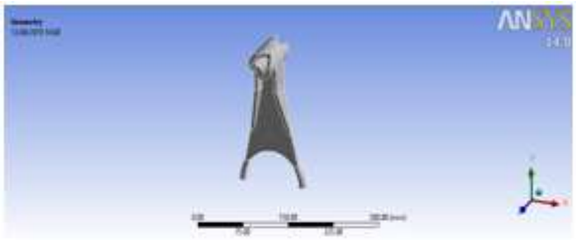

Figure 4: Imported CAD model in ANSYS

There are three main steps, namely: pre-processing, solution and post processing. In pre-processing (model definition) includes: define the geometric domain of the problem, the element type(s) to be used, the material properties of the elements, the geometric properties of the elements (length, area, and the like), the element connectivity (mesh the model), the physical constraints (boundary conditions) and the loadings. In solution phase, the governing algebraic equations in matrix form are assembled and the unknown values of the primary field variable(s) are computed. The computed results are then used by back substitution to determine additional, derived variables, such as reaction forces, element stresses and heat flow. Actually, the features in this step such as matrix manipulation, numerical integration and equation solving are carried out automatically by commercial software. In post processing, the analysis and evaluation of the result is conducted in this step.

Figure 5: Meshed model in ANSYS

Figure 6: Boundary conditions

Top cylindrical surface is provided with frictionless cylindrical support and forces of 500N,1000N,1500N respectively are applied on bottom face as shown in fig. 6 The results obtained are discussed in next section.

4. Result and Discussion

After analysis the contours of principal stresses, von-mises stresses are plotted for each loading condition. The stresses generated in first design with 500N load are given below.

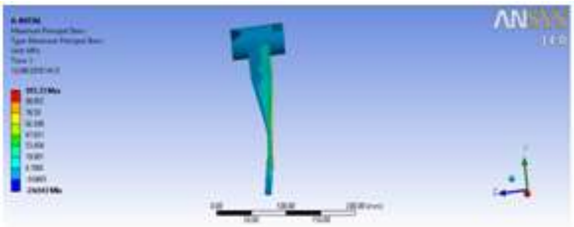

Figure 7: Maximum principal stresses at 500N

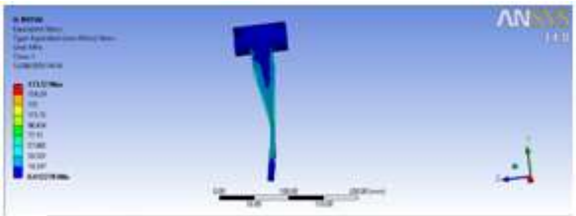

Figure 8: Maximum von-mises stresses at 500N

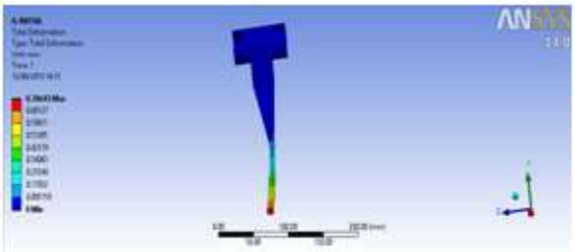

Figure 9: Total deformation at 500N

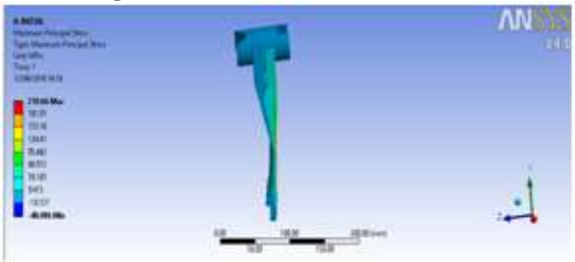

Figure 10: Maximum principal stresses at 1000N

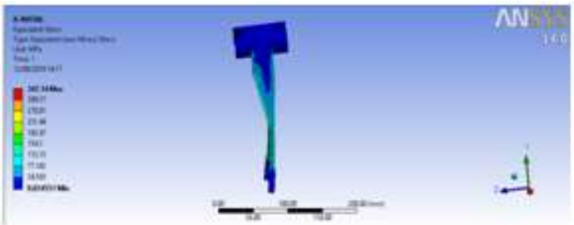

Figure 11: Maximum von-mises stresses at 1000N

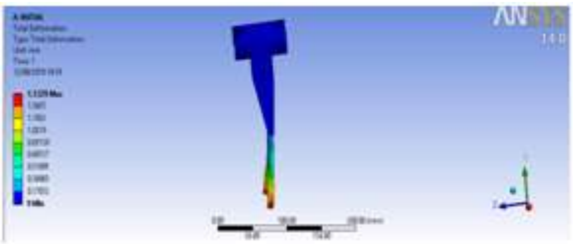

Figure 12: Total deformation at 1000N

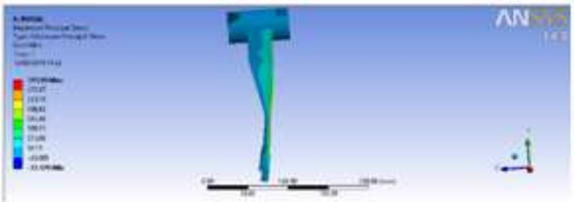

Figure 13: Maximum principal stresses at 1500N

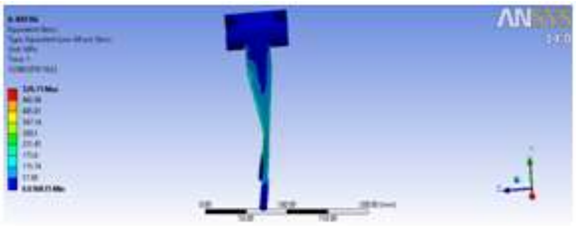

Figure 14: Maximum von-mises stresses at 1000N refrigerant equation

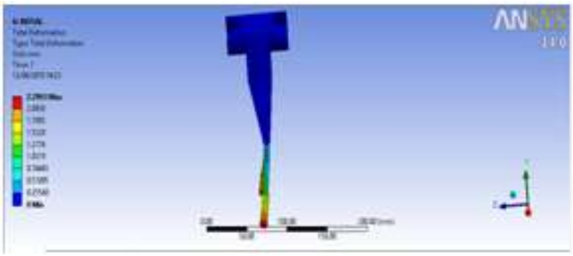

Figure 15: Total deformation at 1500N

The second design without v slot is analyzed under same loading conditions and force of 500N, 1000N, 1500N respectively

Figure 16: CAD model without v slot

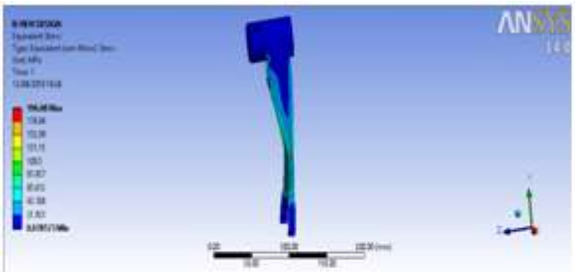

Figure 17: Maximum von-mises stresses at 500N

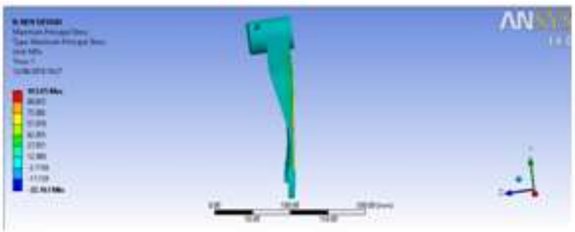

Figure 18: Maximum principal stresses at 500N

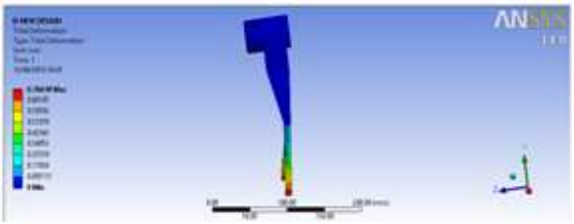

Figure 19: Maximum deformation at 500N

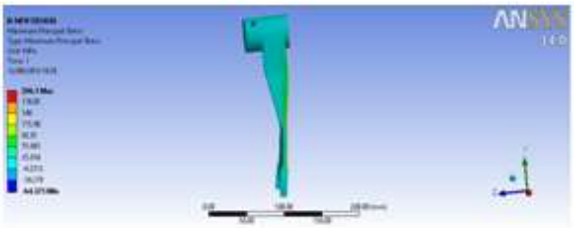

Figure 20: Maximum principal stresses at 1000N

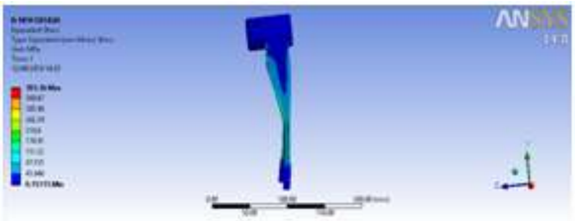

Figure 21: Maximum von-mises stresses at 1000N

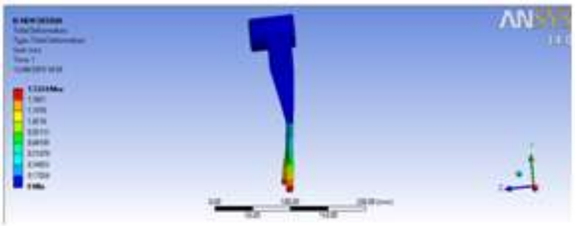

Figure 22: Maximum deformation at 1000N

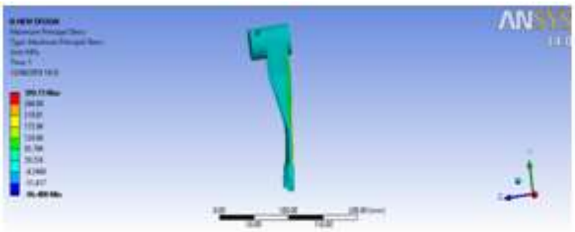

Figure 20: Maximum principal stresses at 1500N

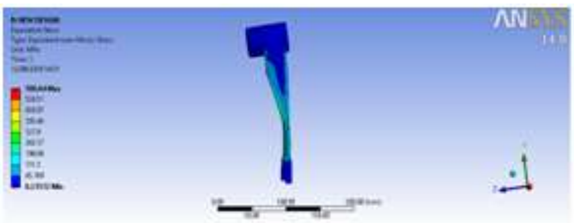

Figure 21: Maximum von-mises stresses at 1500N

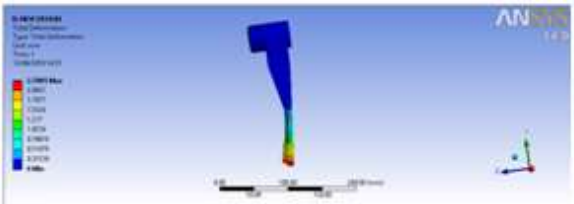

Figure 22: Deformation at 1500N

The analysis is performed and maximum values of principal stresses, von-mises stresses and deflection are shown in table 1for 500N, table 2 for 1000N loading and table 3 for 1500N loading below.

Table 1: Results under 500N loading

Output	Design 1	Design 2
Principal Stress(Mpa)	105.33	103.05
Von Mises Stress(Mpa)	173.57	196.68
Deflection(MM)	0.766	0.76

Table 2: Results under 1000N loading

Output	Design 1	Design 2
Principal Stress(Mpa)	105.33	103.05
Von Mises Stress(Mpa)	173.57	196.68
Deflection(MM)	0.766	0.76

Table 3: Results under 1500N loading

Output	Design 1	Design 2
Principal Stress(Mpa)	105.33	103.05
Von Mises Stress(Mpa)	173.57	196.68
Deflection(MM)	0.766	0.76

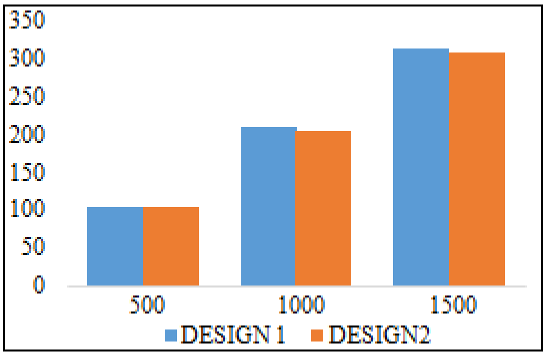

Figure 23: Principal stress comparison

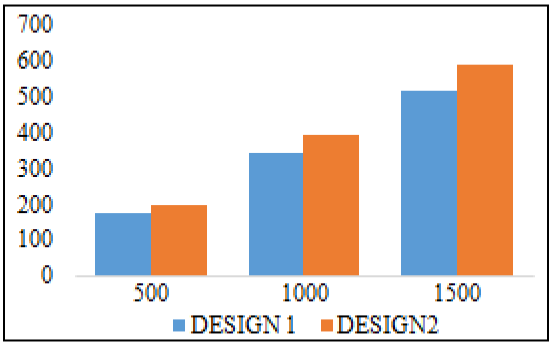

Figure 24: Von mises stress comparison

5. Conclusion

The results show that maximum principal stresses are generated on lower region of fork compared to upper region which is mounted on rail and deflection is also maximum on lower region. Equivalent stress is maximum on v joint on frontal portion of fork. On comparing principal stresses for both the design with and without v block the fork without v block shows lower stresses for all the loadings i.e. 500N, 1000N, 1500N while the deflection is almost equal in both the designs. Therefore from structural analysis we can conclude that fork without v block is better compared to fork with v block which causes higher rejection rate.

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