ANALYSIS OF MINI TRACTOR CRANKSHAFT SUBJECT TO DYNAMIC LOADING

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ABSTRACT

The main aim of this research work is to analyzed and validate existing engine crankshaft von misses stress and strain and deformation which subjected under dynamic loading. To complete this parameters here in this work CATIA a design software is used to prepared model and HYPERMESH is used to meshing and analyzing the existing crankshaft.so ultimately this will used as a tool for analysis and optimization of crankshaft. From that dynamic analysis, fatigue life of the crankshaft and dynamic analysis using modal analysis to find total deformation and frequency of the crankshaft. While the converting the reciprocating motion into rotary motion by the crankshaft, it is subjected to cyclic compressive and tensile loading in vertical direction as well as with some amount of vibratory motion. The study to be carried out to check the load carrying capacity of the crank shaft subjected to both vibration and rotation. This dynamic analysis is conducted on the forged steel crankshaft, single cylinder four stroke engines of mini tractor which is used in agriculture activities.

Keywords: Dynamic Analysis, FEA, Crankshaft, CATIA, HYPERMESH

I. INTRODUCTION

Now a day's maximum affiance, minimum production cost and fine work is major important phenomenon in automobile industries. Vast products like connecting rod, crankshaft, and engine block are wide researchable parts of automobile field. Crank shaft is a major effectible and also complex geometry in the engine, which converts the reciprocating displacement of the piston to a rotary motion with a four bar link mechanism. The Shaft parts which revolve in the main bearings, the crank pins to which the big end of the connecting rod are connected, the crank arms or webs which connect the crank pins and shaft parts. In addition, the linear displacement of an engine is not smooth; as the displacement is caused by the combustion chamber therefore the displacement has sudden shocks.

The concept of using crankshaft is to change these sudden displacements to as smooth rotary output, which is the input to many devices such as generators, pumps and compressors. It should also be stated that the use of a flywheel helps in smoothing the shocks. Crankshaft experiences large forces from gas combustion. This force is applied to the top of the piston and since the connecting rod connects the piston to the crank shaft, the force will be transmitted to the crankshaft.

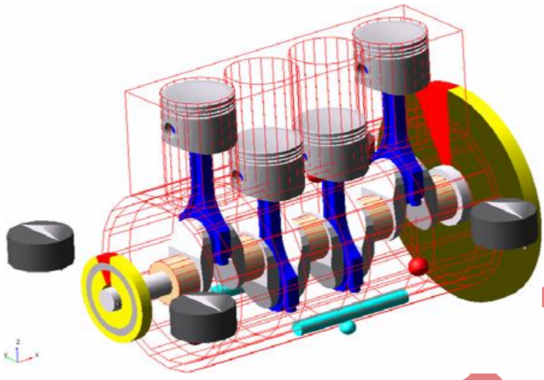


Fig 1. Engine Inside Block Diagram

The magnitude of the forces depends on many factors which consist of crank radius, connecting rod dimensions, and weight of the connecting rod, piston, piston rings, and pin. Due to combustion gas force and inertia force there act mainly two type of forces on crankshaft via connecting rod is torsional load and bending load. A material of crankshaft must be strong enough to sustain the downward force of the power stroke without excessive bending so the reliability and life of the internal combustion engine depend on the strength of the crankshaft largely. The crank pin is like a built in beam with a distributed load along its length that varies with crank positions. Each web is like a cantilever beam subjected to bending and twisting. 1. Bending moment which causes tensile and compressive stresses. 2. Twisting moment causes shear stress. There are many sources of failure in the engine one of the most common crankshaft failure is fatigue at the fillet areas due to the bending load causes by the combustion. The moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crank initiation leading to fracture. For this experimental work all the theoretical equations and some of the assumption and some fundamental design data arereferd form machine design book by R.S.Kurmi. [6]

II. LITERATURE REVIEW

R. J. Deshbhratar, and Y.R Supple. [1] have been analyzed 4- cylinder crankshaft and model of the crankshaft were created by Pro/E Software and then imported to ANSYS software The maximum deformation appears at the centre of crankshaft surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point. The edge of main journal is high stress area. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. So this area prones to appear the bending fatigue crack.

K. Thriveni et al [2] was work on analysis of connecting rod and they are concluded from their study is that The maximum deformation appears at the centre of the crankpin neck surface. The maximum stress appears at the fillet areas between the crankshaft journal and crank cheeks and near the central point journal. The value of von-misses stresses that comes out from the analysis is far less than material yield stress so our design is safe.

Jianmeng et al [3] have been work on 480 diesel engine crankshaft block and they found relationship between the frequency and the vibration modal is explained by the modal analysis of crankshaft. By using Pro-E and Ansys software for modeling and analysis appropriately. They found that results would provide a valuable theoretical foundation for the optimization and improvement of engine design. also they concluded from their research work is that maximum deformation appears at the center of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point. The edge of main journal is high stress area.

S. Bhagya Lakshmi, Sudheer Kumar V, Ch. Nagaraju [4] was work on Dynamic Analysis of Honda Engine Crank Shaft and they find out fine result regarding dynamic analysis like Dynamic analysis is used to find out the motion of the bodies with represents to applied loads with the time frame of crank angle. The crank angle is calculated with respect to seconds as it takes 0.008 sec for each stroke with to revolution. This value re calculated by using spread sheet. The results of the analysis indicate the forces diagram of given connections at different crank angle. The piston acceleration is calculated to find out the theoretical and experimental results values of graph shown in results.

R.ravi et al [5] work regarding analysis and optimization of connecting rod.desining and modification of any other engine parts are ultimately affected on design parameters of crankshaft.inn that experimental work they explore weight and cost reduction opportunities for a production forged steel connecting rod. This has entailed performing a detailed dynamic load stress analysis of the connecting rod. In the first part of the study, the static load analysis and the selection of material and the production method of the connecting rod are considered. Then they go for design of connecting rod in “Catia” Then component was imported to the Ansy’s work bench and analysis is done. Then the results obtained in Ansy’s are compared with Experimental values. Also they found that Dynamic load should be incorporated directly during design and optimization as the design loads, rather than using static loads.

III. MODELING, ANALYSIS AND RESULTS

Methodology of this research work is starting from Modeling of crankshaft and other parts of engine then follow by meshing, applying load magnitude and direction and last dynamic analysis and find out results of Von-misses stress, strain and total deformation.

Configuration of engine from which crankshaft design taken

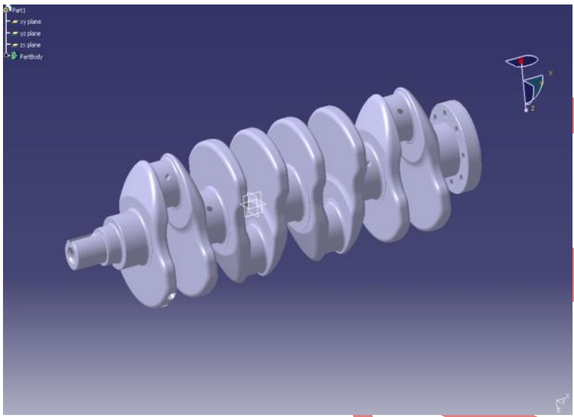
Crank shaft radius	47.5 mm
Piston diameter	0.085m

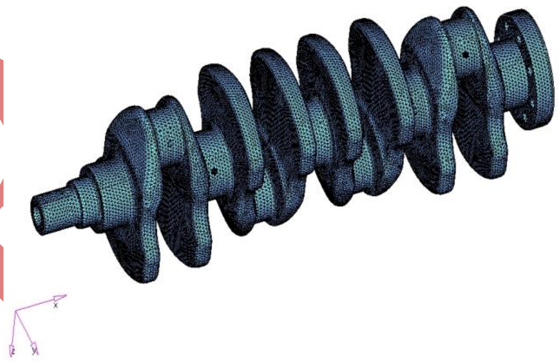
Mass of the connecting rod	0.856kg
Mass of the piston assembly	0.550 kg
Connecting rod length	150mm
Izz of connecting rod about the center of gravity	.004 kg-m^2
Distance of C.G .of connecting rod from crank end center	41mm
Maximum gas pressure	125 bar

Material for Crankshaft is S53C of which modulas of elasticity is 210 GPA

Yield Strength is 584Mpa and Poisson's Ratio=0.3

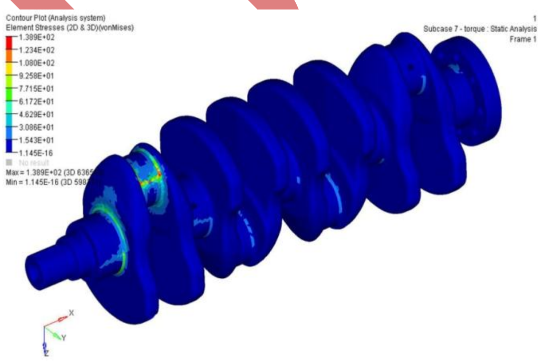
For analysis purpose it will mesh by tetrahedral element by 1mm size so number of elements getting are 353539.





Fg-3 Meshed Geometry of Crankshaft **Fig-2 Model of Crankshaft for Analysis**

Now for analysis of this meshing geometry there are two major approaches for stress calculation: (a) Based on entire crank & (b) Based on single throw.



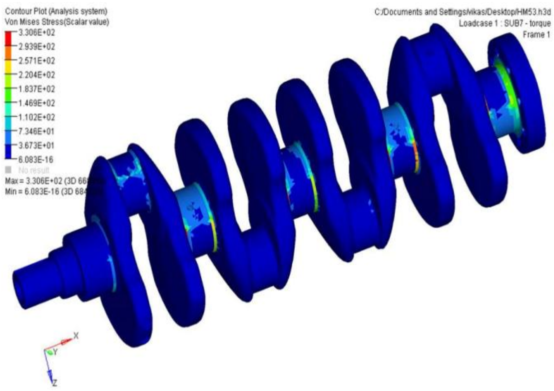


Fig-4 Maximum Bending Load Result **Fig-5 Maximum Torsional Load Result**

Run full crank reduced model (dynamic) to calculate main bearing reactions and torques. Model entire crank shaft with FEM. Constrain the model at the fly wheel end. Run analysis applying all possible loads (at the pin and main bearing locations) (pressure distributed over bearing area) one at a time. Another approach is published can be described as follows: Run dynamic analysis on a reduced model. Cut out one throw of the crank through the main journal middle cross-sections (detailed FE).Constrain one cross-section and apply the forces i.e. bending as well as torsional forces and obtain corresponding stress states. Another approach is to constrain the main bearings for all degree of freedom & applying the bending & torsional force at the crankpin end.

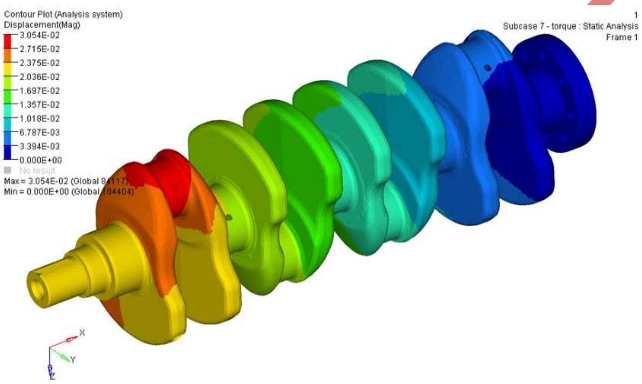


Fig-6 Maximum Displacement Result

Result of that analysis on base of stress and displacement phenomenon shows that the crankshaft shows more stresses in area of fillet sand in the p in journal oil holes. Section changes in the crank shaft geometry result in stress concentrations at inter sections where different sections connect together. Although edges of these sections are filleted in order to decrease the stress level, these fillet areas are highly stresses locations over the geometry of crank shaft. Therefore stresses were traced over these areas. The maximum Bending stress acting on the crank shaft is 330 Mpa by taking both maximum tensional & bending together. Yield strength of the material of crank shaft is 584. So by cross verifying a FOS of this object is coming about 1.76.so its safe design right now. Show figure 4, 5 and 6 indicates results of bending, torsional and displacement respectively.

IV. CONCLUSION

The main target of this research work is upgrade the engine performance in form of breathing capacity by 0.2 L. The following conclusions can be drawn from the dynamic analysis conducted in this study: 1) Dynamic analysis gives more sophisticate results as compare to static analysis.2) the maximum load occurs at the crank angle of 360 degrees for this specific engine. At this angle only bending load is applied to the crankshaft.3) Torsional force is maximum when crank is at $25^{o}$from the top dead centre.

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