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MACHINE MAINTENANCE FOR CRITICAL DESIGN THROUGH CRANKSHAFT GEOMETRY CRITICAL

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ABSTRACT-Crankshaft geometry critical (ie, fault) positions are all on the fillet. Because of the high stress gradients at these locations, high stress concentration factors result. The use of rain flow cycle counting on critical stress histograms shows that only one peak is important throughout the cycle and may cause fatigue damage Components. Geometrical optimization reduces the weight of the forged crankshaft by 18%. This is achieved by changing the size and geometry of the crank arm. Keep the crankshaft statically balanced. This optimization phase does not require any changes to the engine block or connecting rods. Due to the geometric optimization of Phase II, the weight of the crankshaft was reduced by 26%. The need for crankshaft geometry changes during this optimization phase the main bearing in the engine is changed according to the optimized diameter, and the thrust bearing is used to reduce the increase of the axial displacement of the crankshaft.

KEYWORDS:- ***Crankshaft Machine Maintenance, Critical Machine, Longest Downtime, Highest Failure Rate.***

I. INTRODUCTION

The crankshaft experiences a complex loading due to the motion of the connecting rod, which transforms two sources of loading to the crankshaft. The purpose of this study was to optimization of the forged steel crankshaft which requires accurate magnitude of the loading on this component that consists of bending and torsion. The significance of torsion during a cyc. The importance of the revolutionary strength when the maximum amount of time to grow in the coming amount will be left to see if it is important to consider the product or not. Additionally, it is important to recognize the current situation in the charter that FEA required in the car.The main purpose of this section, the effect of connecting the board and the crankshaft and the magnitude and direction of the responsiveness between FEA that operates during the course of the cycle. Studies are used based on one specific meaning. MATLAB programs have been used to address the results. Analysis methods have been dissolved in the curvature mechanism for general branches, the resulting one can be used with veg radius comparison, the synthesis link quality, inertia for bindle sestina, engine speed, spread outfit, and diameter of the fisting, the fist with the corresponding block and cylinder diagrammed the internal pressure of a mechanical change. This search method helps ensure that the installation in the ADAMS View software is correct. However, due to the change in the analysis method of using the MATLAB code change is the most vulnerable result of the ADAMS View software as the quality of the solution analysis solution.

In summary, this paper describes the steps and methodologies that can be used in the MATLAB to speed up fast, rapid and fast-paced connection of piston and large companies between different variations in the system. It is shown that the results of the search by the product template are confirmed in ADAMS. What is the end of the research plan that FEA has discussed? It should be noted that in this study the crankshaft is considered to be rotatable straight, which means that angular angles are not connected in the search. However, the difference between the variation and the force of acceleration or without the immediate effect of less than 1%.

II.ANALYTICAL VECTOR APPROACH

In this section, a complete search procedure is described. The design objective is used with the only status of freedom to resolve motion launching, as shown in the next step to obtain different products for the product. This day represents a crankshaft angle, which is used as an overall degree of freedom in the machine, so, despite the other standing properties in this machine will be a function of this corner. Name some of the most important indicators, such as angular error, angular angles. They use in the matrix MATLAB, and get the angular speed and speed of the drive rod, with the immediate speed and speed of the gravity centre connecting the rod, and forcing the connecting rod crankshaft rod and providing the rod piston. The use of MATLAB programs is that each input and change can be implemented easily, and the information can be processed quickly, while marketing programs like ADMS require more time to process and enter data. When the value of optimization and / or others changes and takes time to make these changes in the commercial software, this use is used when making a backup of things. Complete the MATLAB program.

III. VERIFICATION OF ANALYTICAL APPROACH

The method of study used in this study is illustrated by the third-largest punctuation of a stick, consisting of a rod and a piston. The study is based on robust translators -The sloping path, AB connecting rod is crankshaft, connecting rod BC is the connection combination and designer is a piston assembly. In order to use the cast, the brain type is used with the rope and the transit calculation is used to increase the growth of the body's strength of strength and the actual source of stress. Because only what is in the piston group that can influence the work is a gathering, there is no need for a piston Gathering gender. The connecting unit uses a large range of materials of 2840 kg / m 3 (2.84E-6 kg / mm 3), which is the aluminium metallic unit used in the event.-The crankshaft AB rotational speed was taken to be the maximum operating speed of the engine, which is 3600 rev/min. It is shown in next Sections that the engine speed effect within the operating engine speed of 2000 rpm to 3600 rpm, is less than 15% on the force range applied to the crankshaft. This results in about 10% change in the stress range of the critically stressed location. The 3D model of slider crank mechanism used in ADAMS. Details of slider crank mechanism used in ADAMS are tabulated these details and any other information required were input in the MATLAB program. The loading on the piston which is the slider in the slider crank mechanism was taken to be zero in this part of the analysis since the purpose of this analysis was just compare the results from MATLAB programming and ADAMS software. Although the simulation was performed in a 3D space, the mechanism is a simple 2D linkage; therefore forces are expected to be in the plane of crankshaft motion. Therefore, forces in the longitudinal direction of the crankshaft would be zero in this case. Since the joints at different locations of this mechanism are pin joints, there would be no moment resistance. The results of the analytical approach for the slider crank mechanism using the MATLAB program are plotted through the results from ADAMS software are also included in these figures for comparison. Variation of angular velocity of link BC over one complete cycle of the engine, which is 720 degrees or two cycles of crank rotation. The two curves coincide perfectly indicating agreement of the results from MATLAB programming with results from ADAMS software. Similarly, the variation of angular acceleration over an entire. The variation of forces at joint B defined in the global/non-rotating coordinate system at the engine speed of 2800 rev/min, which is the mean operating speed of the engine, and the same forces at joint B projected in the local/rotating coordinate system. As could be seen in the figures, it can be concluded that the results from MATLAB programming are accurate and reliable.

IV. STATIC ANALYSIS FOR THE ACTUAL CRANKSHAFT

The engine configuration from which the crankshaft was taken The pressure versus crank angle of this specific engine was not available, so the pressure versus volume diagram of a similar engine was considered. This diagram was scaled between the minimum and maximum of pressure and volume of the engine. The four link mechanism was then solved by MATLAB programming to obtain the volume of the cylinder as a function of the crank angle. the scaled graph of pressure versus volume for this specific engine, and pressure versus crankshaft angle, which was used as the applied force on the piston during the Static analysis. It should be noted that the pressure versus volume of the cylinder graph changes as a function of engine speed, but the changes are not significant and the maximum pressure which is the critical loading situation does not change. Therefore, the same diagram was used for different engine speeds in this study. The results of the MATLAB programming are linear velocity and acceleration of the piston assembly, angular velocity and angular acceleration of the connecting rod, linear acceleration of center of gravity of the connecting rod, and forces that are being applied to the bearing between the crankshaft and the connecting rod. The program was run for different engine speeds in the operating engine speed range. Results from the MATLAB programming at the engine speed of 3600 rpm are plotted in though show the variation of linear velocity and linear acceleration of the piston assembly over 720 degrees, respectively. Angular velocity and angular acceleration of the connecting rod during a cycle. Note that variations of velocity and acceleration in both piston assembly and connecting rod from 0° to 360° are identical to their variation from 360° to 720°. The variation of the force at the journal bearing between crankshaft and connecting rod defined in the global/non-rotating coordinate system. Variation of the same force defined in the local/rotating coordinate system. F_x in force that causes bending during service life and F_y is the force that causes torsion on the crankshaft. As can be seen in this figure, the maximum loading happens at the angle of 355° where the combustion takes place. The only difference between these figures is their reference coordinate system, therefore the magnitude, which is not dependent on the coordinate system chosen, is the same in both plots.

As the Static loading on the component is a function of engine speed, the same analysis was performed for different engine speeds which were in the range of operating speed for this engine (i.e. 2000 rpm which is the minimum engine speed and 2800 rpm). The variation of forces defined in the local coordinate system at 2000 rpm and 2800 rpm engine speeds are respectively. Compares the magnitude of maximum torsional load and bending load at different engine speeds. Note from this figures that as the engine speed increases the maximum bending load decreases. The reason for this situation could be explained as follows. As mentioned previously, there are two load sources in the engine; combustion and inertia. It was pointed out that the maximum pressure in the cylinder does not change as the engine speed changes, therefore the load applied to the crankshaft at the moment of maximum pressure due to combustion does not change. This is a bending load since it passes thought the centre of the crank radius. On the other hand, the load caused by inertia varies as a function of engine speed. As the engine speed increases this force increases too. The load produced by combustion is greater than the load caused by inertia and is in the opposite direction, which means the sum of these two forces results in the bending force at the time of combustion. So as the engine speed increases the magnitude of the inertia force increases and this amount is deducted from the greater force which is caused by combustion, resulting in a decrease in total load magnitude. However, factors such as wear and lubrication are important at higher engine speeds. Further discussion of these issues is avoided since they are not of concern in this study and fatigue failure is the main focus.

In this specific engine with its Static loading, it is shown in the next chapter that torsional load has no effect on the range of von Mises stress at the critical location. The main reason for torsional load not having much effect on the stress range is that the maximums of bending and torsional loading happen at different times during the engine cycle. In addition, when the main peak of the bending takes place the magnitude of torsional load is zero. The Static analysis of this single cylinder crankshaft is very similar to an automotive crankshaft which consists of several cylinders. The only difference is the number of applied loads to the mechanism which could be projected to the rotating plane of the crankshaft. In a multi-cylinder crankshaft the effect of combustion of other cylinders on one cylinder results in high torsional load which must be included in the analysis. Since the studied crankshaft belonged to a single cylinder engine, there would be no such effect. Therefore, the analysis could be performed without considering torsional load. The noise and vibration analysis of single cylinder and multi-cylinder crankshafts are similar. The longitudinal and radial displacements of a single throw, which consists of two main bearings, two crank webs, and a crankpin, under service load is measured in order to define the noise and vibration level of the crankshaft.

V.FEA WITH STATIC LOADS

There are two different methods that can be used on the crankshaft to obtain the stress history. One approach is to run the finite element model during the engine cycle or at a selected time above 720° by applying the load in such a way that the load can define the component's stress history record. Another method of obtaining stress at different locations at different times during the cycle is by superimposing basic loading conditions. This involves applying a unit load under basic conditions and then scaling the stress from each unit load according to the static load. Then add similar stress components together. In this study, both methods were used at 3600 rpm engine speedThe results of verifying the two methods are the same. After verifying the results, the stacking method was used by writing the code in an Excel spreadsheet to perform the necessary calculations and obtaining stress results at different crankshaft angles. Based on the peak and trough of the load change, the reason for choosing some locations beyond 720° is justified. Three different graphs were used for selecting proper points to cover the entire cycle; bending, torsion, and total load magnitude.

VI.STRESS ANALYSIS AND FEA

Geometry generation used for finite element analysis, describes the accuracy of the model and explains the simplifications that were made to obtain an efficient FE model. Grid generation and its convergence are discussed. Using appropriate boundary conditions and loading types is very important because they strongly influence the results of finite element analysis. It also discusses the determination of appropriate boundary conditions and loading conditions. Finite element models for two components were analyzed; cast iron crankshafts and forged steel crankshafts. Since both crankshafts are from similar engines, both use the same boundary conditions and loads. This helps to correctly compare this part made by two different manufacturing processes. This chapter will discuss the finite element analysis results of these two crankshafts. According to the installation of the crankshaft in the engine, considering the boundary conditions, the above finite element model is used for static analysis.

In order to evaluate the FEA results, a component test was conducted with strain gages. FEA boundary conditions were changed according to the test setup. Strain gages were mounted on the forged steel crankshaft and results from FE analysis and experimental data were compared in order to show the accuracy of the FE model.

Finally, results from Static FE analysis, which consist of stress history at different locations, were used as the input to the optimization process discussed.

VII.FINITE ELEMENT MODELLING

The final product product of any important component includes mathematical translation, application of hardware, entry, distribution, and use of the form type correctly. These steps lead to violence and rotation in the event. In this study, such research findings are conducted for the production of iron and repair.

VIII.GENERATION OF THE GEOMETRY OF CRANKSHAFTS

The dimensions of the crankshafts were measured using a clipper and a CMM (coordinate measuring machine) with the accuracy of 0.025 mm (0.001 in) and 0.0025 (0.0001 in), respectively. Having accurate dimensions of both crankshafts solid models were generated using I-DEAS Master Modeler.

The solid model generated for the cast crankshaft is and a picture of the cast crankshaft from which the geometry was generated is As can be seen in the picture of the cast crankshaft the gear on the rear side of the main bearing is not included in the digitized model, instead the plane next to it is extruded to cover the section which the gear covers. This simplification is reasonable since the gear is press fit at this location and the gear tooth does not have any effect on stresses at fillet areas where high stress gradient exists. It should be mentioned that simplification on the gear will not affect the stiffness of the model because the boundary condition used on this side of the crankshaft is a sliding edge and moving its location along the main bearing shaft will not change the stress results. In addition, the significant effect of bending load was observed to be in the cross section of the crankpin bearing and crank web. Therefore, simplifications on the main bearings will not affect stresses at critical areas. Another simplification done on the cast iron crankshaft was neglecting the cap at the end of oil way on the bearing of the connecting rod. This cap is pressed in its place to seal the drilled hole to prevent oil to flow out. The absence of this cap does not affect stresses at any location; therefore the cap was not included in the model. The last simplification done on the cast iron crankshaft was neglecting the slight slope on the crank web from outside toward centerline of main bearings. The slope is a result of manufacturing process. The molds in casting are such that the molded part could easily slip out the molds; as a result, walls that are perpendicular to the mold movement must have a positive slope. Since the change in the web thickness was small, by averaging thickness over the entire crank web, the crank web was modeled with uniform thickness. This simplification is acceptable since stresses at this location are very low and the thickness does not change the results at the fillet areas, which are critical locations. The solid model of the forged steel crankshaft, and Figure 4.4 shows a picture from the same view of the received crankshaft. As can be seen the threaded front shaft is not threaded in the model, since this part is out of loading and boundary conditions and has no effect on stresses at different locations. Another simplification made to forged steel crankshaft model was in the crank web slope, similar to the cast iron crankshaft. In the manufacturing process of forging there is a need of the same slight slope as in casting process on the component in order to ease removing the forged component from the forging die. The web slope in the forged component is less than the web slope in the cast iron crankshaft. Again, an average value was used as the uniform thickness on the crank web in this model.

Drilled holes on the counter weights in order to balance the crankshafts and inside threads on holes at the back of the crankshafts were not included in the models since their presence makes the geometry complicated but they do not affect stresses at critical locations.

The cast iron crankshaft weight as measured on a weighing scale was 3.58 kg, and the weight of the forged steel crankshaft was 3.80 kg. The difference between the generated models and the actual crankshafts were about 7% and 2%, for the cast iron and forged steel crankshafts, respectively. The reason for 7% difference in the cast crankshaft is not including the gear tooth in the solid model, which if considered, the difference will reduce to only 3%. Material properties used in both models.

Another important characteristic for this component is the Static balance of the geometry; this property could be verified by the location of center of gravity. Examining the location of center of gravity of both digitized models showed a very close distance from the main bearings center line. By adding the drilled holes on the counter weight of the actual crankshafts, the center of gravity will coincide with the main bearing center line, indicating the component would be in proper Static balance.

This weight comparison between the actual crankshafts and corresponding models and Static balance are indications of accuracy of the generated models.

IX.MESH GENERATION

FEA analysis was performed on both crankshafts for the Static load analysis, as well as for the test setup. Since boundary conditions of Static FEA and test setup FEA are different, separate FE models were needed. In this section, meshing of both Static FEA and test setup FEA are presented for the forged steel and cast iron crankshafts.

X.STATIC FEA

Quadratic tetrahedral elements were used to mesh the crankshaft finite element geometry. Tetrahedral elements are the only option for meshing the imported complex geometries to the ABAQUS software. Using linear tetrahedral elements will result in a rigid model with less accuracy, whereas using quadratic tetrahedral elements will increase the accuracy and lessen the rigidity of the geometry. In order to mesh the geometry with this element type, the free meshing feature of ABAQUS software was used. In this feature, the global mesh size could be defined, while for critical locations free local meshing could be used to increase the number of elements for accurate stresses at locations with high stress gradients.

Convergence of stress at different locations was considered as the criterion for mesh size and number of elements selection. on Mises stress magnitude at six locations on fillet areas versus number of elements in the forged steel crankshaft geometry for a load of 20.3 kN applied in the F_x direction As could be seen from, with the increase of element numbers, especially

in the fillet areas, the stresses converge. Satisfactory results were obtained using 119,337 elements for the forged steel crankshaft and 121,138 elements for the cast iron crankshaft, corresponding to a global mesh size of 5.08 mm and a local mesh size of 0.762 mm in each model. This local mesh size results in having 5 elements in the radius of the fillet areas for both crankshafts.

In order to have more efficient models, different element sizes where used for the final models which increased the number of elements to 122,441 and 128,366 for the forged steel and the cast iron crankshafts, respectively. The selection of different sizes for elements was made to obtain a uniform growth of element size as the element size changed through the geometry. Shows element sizes used at different locations for the forged steel crankshaft and element sizes used for the cast iron crankshaft. the meshed geometry for forged and cast components, respectively, which were meshed using the mentioned mesh size growth.

XI.TEST ASSEMBLY FEA

FEA was also conducted for the bench test assembly of the forged steel crankshaft. According to the fixture of the component test assembly a separate model was created. The only difference in the mesh is the size of mesh at the part where the crankshaft is gripped in the fixture Local mesh size used in the test assembly FEA. The reason for using a fine mesh on the boundary of the gripped location is to measure stress around the oil hole which act as a stress concentration. Considering the above mentioned considerations, the crankshaft model resulted in 137,779 elements.

XII. Conclusion

Static load analysis of the crankshaft results in more realistic stress, while static analysis provides overestimated results. Accurate stress is the key input for fatigue. Analysis and optimization of the crankshaft. There are two different sources of load in the engine; inertia and combustion. These two sources of load can cause bending and torsional loads on the crankshaft. Maximum load Occurs at 355 degrees crank angle for this particular engine. At this angle, only the bending load is applied to the crankshaft. Considering torsional load in the overall Static loading conditions has no effect on von Mises stress at the critically stressed location. The effect of torsion on the stress range is also relatively small at other locations undergoing torsional load. Therefore, the crankshaft analysis could be simplified to applying only bending load. Overlay of finite element analysis results from two vertical loads is valid. A simple method obtains stress under different load conditions based on the force exerted on the crankshaft in the static analysis. Experimental stress and finite element analysis showed close agreement, within 7%. These results indicate the asymmetrical bending stress on the crank pin bearing, while using analytical methods to predict the bending stress is symmetrical at this position. The lack of symmetry is a geometric deformation effect, indicating that FEA modelling is required due to the relatively complex geometry of the crankshaft.

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