



Finite Element Analyses of Automobile Crankshaft Using ANSYS

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Author's contribution

The sole author designed, analysed, interpreted and prepared the manuscript.

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ABSTRACT

Finite Element Method (FEM) of failure analysis was developed on automobile crankshaft, to determine the stress distribution and the fatigue life, by using ANSYS software. Further, an analytical analysis is applied, Measure the crankshaft stress life. A study was performed on some of the Honda CR-V engine components, specifically are crankshaft, the connecting rod, and the piston. Upon the finite element analysis, it was found that the fillet areas of the crankshaft are the most critical locations where high stresses were generated in these areas. Moreover, with/without considering torsional force acting on the crankshaft does not appear to have any major effects on the stress experience by the crankshaft. In addition, the location where the crack initiated, and fatigue failure starts is located at one of the crankpin journal fillet areas. Indeed, the crankshaft critical areas are mostly affected by uniaxial stress. Moreover, the prediction of the crankshaft fatigue life by using the strain-life theory gives the overall most conservative fatigue life results.

Keywords: Finite element method; ANSYS; fatigue life; crankshaft.

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NOMENCLATURE

k_a	: Surface condition
k_b	: Size part or the cross-section differs from that of the test specimens
k_c	: Load factor between the actual part and the test specimens
k_d	: Temperature
k_e	: Reliability factor of scatter of test data
k_f	: Miscellaneous factor is accounts for reductions from all other effects, including residual stresses, corrosion, plating, metal spraying, fretting, and others
dm	: Mass element
r_m, r_c	: The distance to the center of mass of the entire rigid body
θ	: is the crankshaft angle of rotation
ω	: The angular velocity
α	: acceleration
l_1	: The length of the crankshaft
l_2	: The connecting rod
V_{rod}	: The linear velocity of the center of mass of the connecting rod
$l_{g,x}, l_{g,y}$: The distance from rotating axis of the crankshaft to the center of mass of the connecting rod in X and Y directions.
a_{rod}	: The linear accelerations of the center of mass of the connecting rod
V_{pis}	: The linear velocity of the piston in X and Y direction
$l_{p,x}$: The distance from rotating axis of the crankshaft to the piston in X direction
a_{pis}	: Lastly, the linear accelerations of the piston
r_1, r_2, r_0	: constant
\bar{p}	: The position vector for the center of mass of the connecting rod
γ	: The distance from point A to the center of mass of the connecting rod
D_{pis}	: The diameter of the piston
m_{crank}	: The mass of the crankshaft (for single throw)
m_{rod}	: The mass of the connecting rod
m_{pis}	: The mass of the piston
σ_v	: where σ_v is the equivalent Von Mises stress and σ_1, σ_1 and σ_1 are the principle stresses

1. INTRODUCTION

Crankshaft is one of the complex geometries as well as a large component in the engine. It's

main function is, converting the motion of the piston from reciprocating displacement to a rotary motion with a four-link mechanism. Even though, the crankshaft has shown a huge number of load cycles through its service life, the component fatigue performance and durability should be considered in the design process [1]. Design developments of crankshaft production became an important issue in the industry, to reduce the cost of the manufacturing process of component with the minimum weight possible, an appropriate fatigue strength and other functional requirements. Indeed, all these enhancements finding in terms of better fuel efficiency and higher power output are suited only the lighter and smaller engines [2].

All the engine components are subjected to constant to varying load which also varies in direction and due to these, components may fail. Bending and shear stress due to twisting are common stresses acting on crankshaft. The crankshaft failure such as the cracks form in fillet area it is because of the repeated bending and twisting effect. Therefore, fatigue plays an important role in crankshaft development. The prediction accuracy of fatigue life is very important in order to ensure the components and its reliability safety [3].

The sudden failure of crankshaft made researchers and academia to investigate the problem. Because of complicated loading and geometry problem, general method to predict fatigue life is still not evolved. Since crankshaft is subjected to several forces which vary in magnitude, direction (multiaxial) and connecting rod transmitting gas pressure from cylinder to crankpin. Where, stresses acting in the crankshaft vary with respect to time. Most of time crankshaft fails due to fatigue at fillet areas due to bending load [2]. Several methods are developed by researchers and some are available.

The most three used methods to predict fatigue life are as following:

- i) stress life(S-N),
- ii) strain Life (E-N) and
- iii) Linear Elastic Fracture Mechanics (LEFM).

The 1st one is based on nominal stress life which is uses rain flow cycle counting. In addition, this method it could be helpful to test fatigue life. In contrary, the main disadvantages of this method that plasticity effect is not considered, and it was shown a poor accuracy

for low cycle fatigue [4]. The 2nd method provides more analysis details which is involving plastic deformation at a localized region and it is useful for low cycle fatigue. Lastly, the 3rd method is estimated that, the availability of crack is already detected and predicted the crack growth by considering stress intensity factor. As comparison among these methods, stress life is most commonly utilized method where S-N curve data is commonly available for particular material.

Another method is so called Damaged Tolerance Analysis (DTA) where uses LEFM to predict the crack stability, crack growth, and hence minimal time between the two inspection to avoid a crack reaching critical size. The function of this method is to assess the effect of cracks in the structure. Damage tolerance is the ability to resist the fracture from pre-existing crack in the given period of time to avoid catastrophic failure [5]. Fatigue Life could be determined also by the Principal Stress Criterion (PSC) by considering largest principal stress [6]. Principal strain criterion considers largest principal strain, and Von misses equivalent strain criterion provides an estimate of onset yielding behaviour of material can also be used to identify the estimation of the fatigue life. Nevertheless, surface treatment of an engine crankshaft is significantly on deep rolling of the fillet area where the fatigue life of a crankshaft is increased by developing a compressive residual stress in fillet area [7].

2. FAILURE OF CRANKSHAFT

Various factors in mechanical fatigue failure of crankshaft play important role such as, misalignment of crankshaft on assembly, improper journal bearing, and vibration due to some problem with main bearing or due to high stress concentration caused by incorrect fillet size, oil leakage, overloading, and high operating oil temperature [8]. Fillet rolling it may increase fatigue life [9]. The geometry and mechanism of an engine have great influence on fillet that experiences a large stress cycle during its operation [1]. Small crack initiates at this fillet location due to stress concentration which will propagate subsequently, and failure of crankshaft takes place. Most of the crankshafts that failed in fatigue were due to bending fatigue Design and Development mostly has an essential issue in the industry for crankshaft production to manufacture less expensive crankshaft with high fatigue strength [10]. Another, factors of Heat and stresses during

contact bearing with repetition of heating and cooling would eventually create thermal fatigue cracks and it will propagate with time [11]. An investigation was made by Jung et al. [12], they have used ductile crankshaft which is commonly used for SUV and lightweight truck. A crankshaft is mounted on fixture and monotonic bending load and frequency was applied to front main bearing. After investigating the sample cracks are found initiates around fillet area and propagating rapidly causing failure.

Crankshaft is one of the main rotary parts of the engine that translates the linear motion into rotating motion of the piston [13]. another research work has been conducted to examine the fatigue fracture mechanism and microstructure of crankshaft for a heavy-duty truck diesel engine. The failure of crankshaft has begun in the third crankpin during the travelled distance of 955,000 km [14]. A research group have performed failure analysis on crankshafts of two diesel engine for mini backhoe and the automobile vehicle. However, the crack was grown in both crankshafts, the crankpin–web fillet has extended, and the cracks forms were semi-elliptical and symmetrical. The effect of mode I had approved the same fact [15].

3. APPLICATION FEM IN FATIGUE LIFE

The use of numerical method such as Finite Element Method now a day commonly used to give detail information about structure or component. This method predicts the behaviour which is otherwise difficult to find out by theoretical calculation, as great number of degrees of freedom are engaged. FEM can be used as excellent tool to analyses and find out the approximation of the crankshaft fatigue life by computer simulation and therefore it can help to reduce time and costs required for prototyping and to avoid numerous test series when laboratory testing is not available. Since loading on crankshaft is complex in nature, sophisticated analysis of crankshaft is required. Various Finite Element analysis tool such as MSC-Fatigue, ANSYS, FEMFAT etc. are commonly used now a days by automobile companies to check durability of their products.

In last two decades, Renault Company has developed a new crankshaft durability assessment tool based on 3D mesh, to improve the fatigue analysis. These approaches calculate fatigue factor of safety through external load calculation. Where, mesh generation and FEM load distribution were employed to calculate the

stresses. In fact, the crankshaft fillet stresses are highly localized and then, the distribution of stress is very complex inside of the automotive engine crankshaft [16].

Another research group conducted dynamic simulation using Finite Element Analysis (FEA) on single cylinder of four stroke engine, to compute the stress magnitude at critical location. The dynamic simulation analysis was analytically solved and verified in ADAMS software. FE model was accessed in ABAQUAS and boundary condition was applied according to engine mounting conditions. It was found that torsional load was very small as compared to other loads and hence neglected and analysis was made simpler by applying inertia and gas forces [2].

4. DYNAMIC LOAD ANALYSIS

The applied loads dynamically changed on the crankshaft at different rotation angle, because of the connecting rod motion; it means that a complex loading was occurred and transforms two sources of loading to the crankshaft. The main goals of this Finite Element Analysis (FEA) are to evaluate the fatigue life performance and the stress distribution of the crankshaft using ANSYS stimulation software [2]. Further, the software requires an accurate application of the crankshaft under the influence of magnitude of bending and torsion loading. Moreover, the implication of torsion under cycle loading and its maximum should be investigated and compared to the total loading magnitude.

The main objectives of this study of FEA, are to evaluate the load application of its magnitude and orientation on the bearing which is located between connecting rod and crankshaft. Therefore, in this analysis, the crankshaft is assumed to be rotates at a steady-state-velocity, thus means the angular acceleration was inclusive in the analysis. However, as per a comparison of forces with or without acceleration consideration, the differences founded to be minority. The equations obtained were then programmed in Microsoft Excel's Visual Basic for Applications (VBA) in order to perform calculations of the loading at different angle of rotation. Figs. (1, 2 and 3) below shown the engine the main components of Honda CR-V Crankshaft, these specifically are; crankshaft, connecting rod, and piston.

5. DYNAMIC FORCE RESULTS

For the result of dynamic loading, the two main forces conditions have been applied on the

crankpin bearing surface. These forces are oriented as F_x and F_y and are perpendicular to each other. Forasmuch-as, there is no tension occurred between the contact surface of connecting rod and crankpin bearing, the loading of bending, F_x and torsional F_y could also be applied in the opposite direction as shown in in Fig. 4 [2].



Fig. 1. Crankshaft

$l_1 = 0.0443 \text{ m}$, mass of the crank, $m_{crank} = 3.9 \text{ kg}$



Fig. 2. Connecting rod

Connecting rod, $l_2 = 0.137 \text{ m}$, $m_{rod} = 0.52 \text{ kg}$

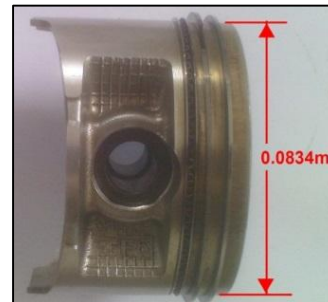


Fig. 3. Piston

Piston, $D_{pis} = 0.083 \text{ m}$, $m_{pis} = 0.4 \text{ kg}$

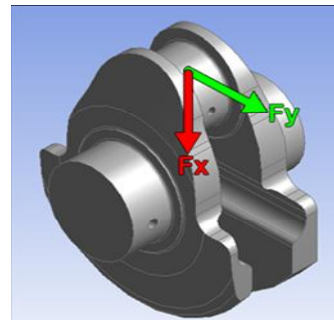


Fig. 4. Components of force acting on crankshaft

The dynamic loads at crankshaft rotating at 2000rpm from both methods (analytical and vector methods) were plotted in the graphs below (Fig.5 and Fig.6) over the crankshaft orientation at 720 degrees., it can be noticed from the Fig. 4, that the maximum load occurs at the crankshaft angle of 365° with the load in x-direction (bending load).

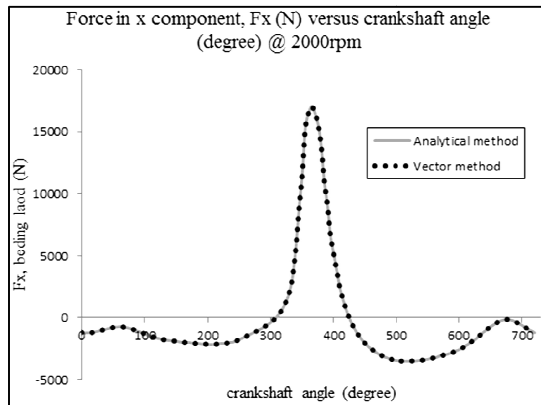


Fig. 5. Graph of bending load versus crankshaft angle

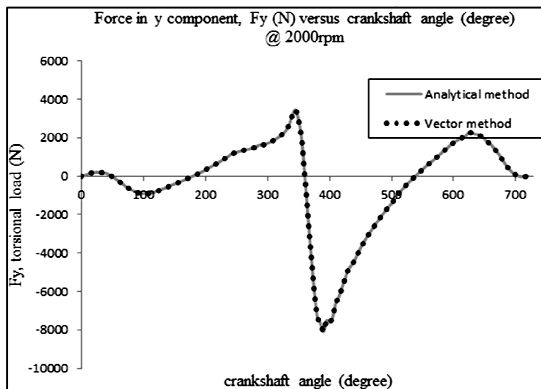


Fig. 6. Graph of torsional load versus crankshaft angle

The results of crankshaft angle of the dynamic force were compared for both methods of kinematics velocity/acceleration analysis. From the graphs (Fig. 5 and Fig. 6), both methods have shown similarity and comparable results with minimal differences. Upon, the obtained results obviously show that the dynamic force results from either method are accurate and reliable to be used.

6. STRESS ANALYSIS

The preparation procedures of finite element analysis are employed through the applied

simulation on the single throw of the crankshaft [17]. This preparation includes the required material properties for cast iron Which is crankshaft material and the real dimensions of the crankshaft geometry. Accordingly, the mesh generation of the single crankshaft throw should be on 3-D geometry and setting the boundary conditions (fix supports and force loading) [2]. Further, Von Misses is used to compute the distribution of stress. Regarding, the stress distribution, the critical locations of the crankshaft can be found. Furthermore, comparison of the stress will be presented with/ without including the torsion load effect on the stress as well as the thermal stress effect on the overall stress of the crankshaft.

William and his group [18] have done some intensive research and experiments on and forged steel for crankshafts in their research paper. The needed properties of particular material for finite element simulation are chooses from the published literature [19]. The cast iron material properties are listed in Table 1.

Table 1. Cast iron material properties.

Material	Cast iron
Mass density	7197 kg/m ³
Poisson's ratio	0.30
Modulus of elasticity	178 GPa
Yield strength	412 MPa
Coefficient of thermal expansion	1.1 * 10 ⁻⁵ °C ⁻¹

6.1 Geometry Generation

The introductions of the geometry constructions and generations by obtaining geometric solutions details along the help of descriptive geometry play an important role in spatial investigation and object measurement generally, in the engineering and specially in mechanical engineering design [20].

In order to model the crankshaft for finite element analysis, the measurements of the required crankshaft dimensions have been obtained using a vernier caliper with the accuracy of 0.005 cm [2]. With the complete lengths and dimensions acquired, the 3-dimensional geometry of the single throw of the crankshaft are then modeled in ANSYS software. Fig. 7 and Fig. 8 shows the different views of the geometry for the single crankshaft throw.

6.2 Mesh Generation

Finite Mesh generation is a very important step in FEA for the accuracy of the simulation result.

Basically, the finer the meshing (more meshing elements), the more accurate the final result will be. In fact, the mesh size can be also affecting the usage of computer hardware resource such as the processor. In order to generate an efficient mesh, smaller meshing size could be done on critical areas where high stresses are occurred under certain loading. From the research history, it was found that the critical locations for crankshaft are mostly, around the fillet areas. Further, the mesh refinement could be done on these locations whereas these stress results from these locations are more important than other locations. The advantage of refinement is that a smaller number of mesh elements will be used on locations of less interest (non-critical locations). Therefore, it significantly reduces the overall number of mesh elements and of course will be beneficial in the usage of computer hardware resource. Furthermore, mesh refinement on these locations was performed on ANSYS with a refinement factor of 1 (1 being lowest, 3 being highest refinement). Overall, the mesh generated in a total of 55855 elements and 84757 nodes. Fig. 9 shows the meshing generation model of the crankshaft.

slider crank mechanisms as shown in Fig. 10. Accordingly, it could be noticed that from one side, the crankshaft mechanism has limitation with a ball bearing and in another side with the journal bearing. The crankshaft mechanism is allowed to rotate only about its main central axis and it hasn't any motion other than rotation.

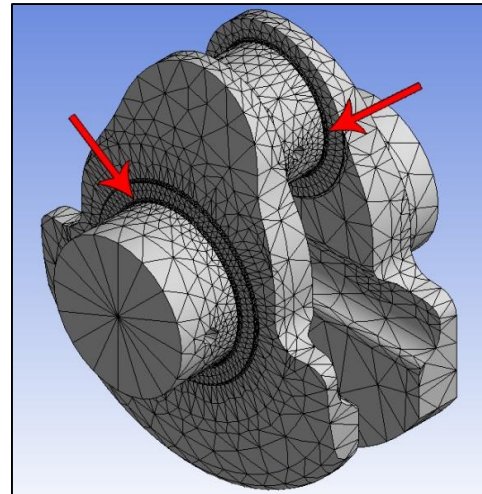


Fig. 9. Meshing refinements on critical locations

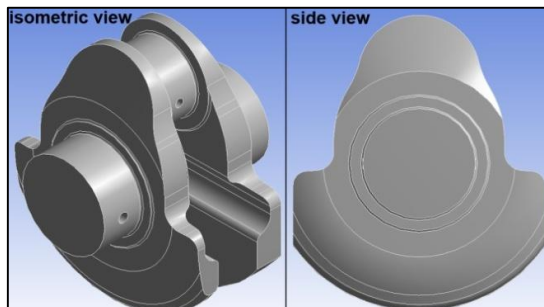


Fig. 7. The crankshaft isometric and side views



Fig. 10. Crankshaft mounted on engine

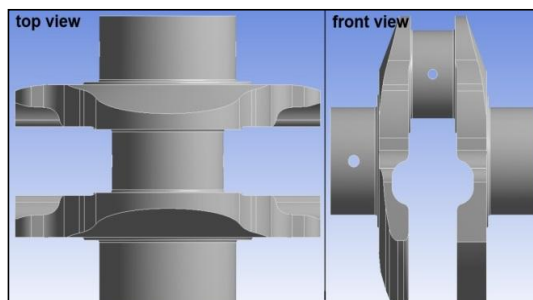


Fig. 8. The crankshaft top and front views

6.3 Boundary Conditions

The appropriate boundary conditions are determined from the nature of the crankshaft's

Therefore, the fix constraints were defined at both side of the single throw of the crankshaft as shown in Fig. 11. the fixed edge in a journal bearing is defined based on the degrees of freedom. In addition, thus allows the crankshaft to have displacement along its central axis [2]. With these fix constraints, it allows the force to be acting on the crankshaft at different directions. In other words, force acting at different directions simulates the force acting on the rotating nature of the crankshaft. The boundary condition for the force is shown in Fig. 11.

6.4 Von Mises Stress

There are several failure theories available in ANSYS to display the stress distribution. The

theorem of maximum principal stress is the most conservative of failure theories and is used for brittle materials. Regarding the used material of crankshaft in this study is ductile cast iron, and in this case the applied theory for ductile failure is “von Mises theory” which is also known as Von Mises-Hencky criterion. It is commonly used for ductile materials and is seen most often when evaluating stresses, both static and dynamic. There are three "Principal Stresses that could be computed at any point, applying in the x, y, and z orientations. (The x, y, and z orientations are the "principal axes"). The formula Von Mises criterion is combining these 3 principal stresses into an equivalent stress, and then is compared with the material yield stress. (the failure criteria of ductile materials are mostly considered the yield stress). The equivalent stress is often called the Von Mises stress [20]. If the Von Mises stress exceeds the yield stress, then the material will be under the failure condition. The equation of Von Mises stress is given as follows:

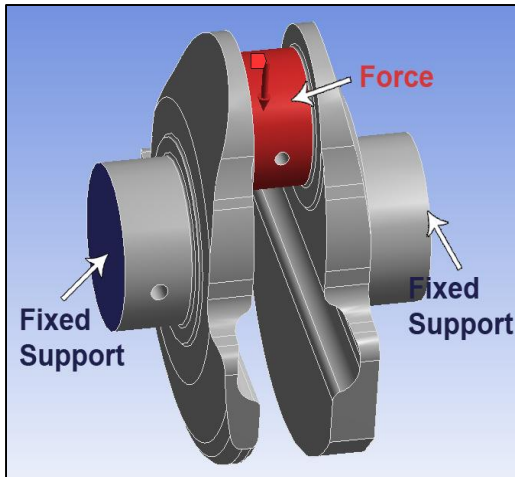


Fig. 11. Boundary conditions

$$2\sigma_v^2 = (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2 \quad (1)$$

Where σ_v is the equivalent Von Mises stress and σ_1 , σ_2 and σ_3 are the principal stresses.

On the other hand, the maximum shear stress criterion which is also known as Tresca’s or Guest’s criterion could be used also to predict the failure and yielding of ductile materials [21]. The maximum shear stress equation is as follow:

$$\tau_{max} = \frac{\sigma_{yield}}{2} = \frac{|\sigma_1 - \sigma_2|}{2} \text{ or } \frac{|\sigma_2 - \sigma_3|}{2} \text{ or } \frac{|\sigma_3 - \sigma_1|}{2} \quad (2)$$

Von Mises criterion was chosen to be used in this crankshaft failure study to determine the

stress distribution and fatigue performance. This is because the actual torsion tests used to develop pure shear have shown that, the criterion results of von Mises stress are more accurate than the maximum shear stress theory [19].

6.5 Stress Distribution and Critical Locations

Fig.12 and Fig.13 shows the 3-D stress distribution result from ANSYS. From the Figures, the colour represents the intensity of the stress from blue (low stress) to red (high stress), the overall stress distribution could be observed clearly from the model, the arrow points to the critical areas with the highest stress. The highest Von Mises stress value recorded is 4.672×10^7 Pascal at one of the fillet areas. Moreover, the analysis shows that the crankshaft fillet areas were encountered the highest stress when the bending and torsional loading exerted on the crankshaft. Wherefore, the critical areas are located on the fillets of the crankshaft.

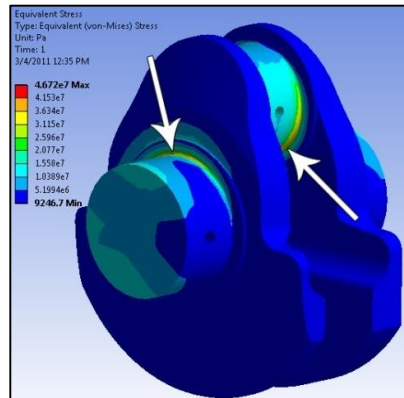


Fig. 12. The crankshaft stress distribution

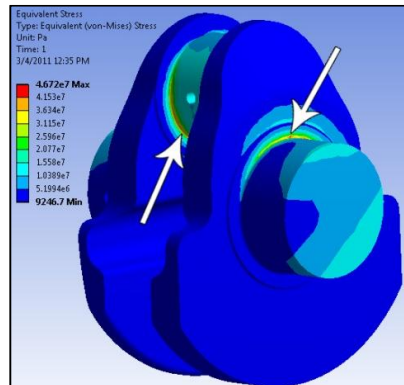


Fig. 13. The crankshaft Stress distribution

7. FINDINGS OF ANSYS FOR FATIGUE LIFE ANALYSIS

The performance analysis of the crankshaft on fatigue life was evaluated using finite element method. The fatigue material properties are chosen from a relevant published study [22]. The fatigue life equation for notched crankshaft was determined by:

Jonathan R. et al. 2007; Budynas R & Nisbett J. 2003:

$$\sigma_a = \frac{557.1 \text{ MPa}}{0.339 + (2N_f)^{0.087}} \quad (3)$$

Fig. 14 shows, the fatigue life diagram generated by ANSYS, it shows that the location at where the fatigue failure starts (the areas in red colour means lower fatigue life for that areas). In other word, the initiation of microscopic crack will first occur at this location. The fatigue location is located at the crankshaft critical fillet areas because these areas experienced overall highest stress concentration. From the figure, the actual location where the fatigue failure starts is located at one of the fillets of the crankpin journal and this was proven by researchers and Various studies analysis as mentioned in the literature background.

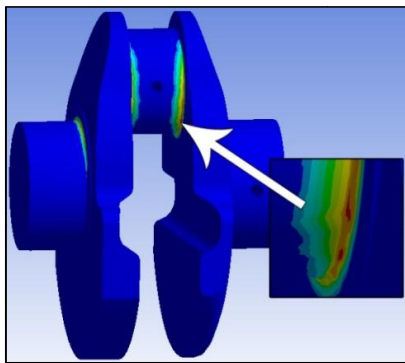


Fig. 14. The location areas of Fatigue

Normally, fatigue material properties are based on uniaxial stresses, but real-world stress states are usually multiaxial. The biaxiality indication result in ANSYS shows the overall stress state over the model. Fig. 15 shows the contour plot of biaxiality indication of the crankshaft. Further, it seems, the majority of this model is under a pure uniaxial stress, with parts exhibiting both pure shear and nearly pure biaxiality. From further investigation, it can be seen that the most crankshaft critical fillet areas occur at a point of mostly uniaxial stress. Therefore, the usual

fatigue material properties obtained from uniaxial stress test shall be used confidentially, to estimate and predict the fatigue life performance for the crankshaft.

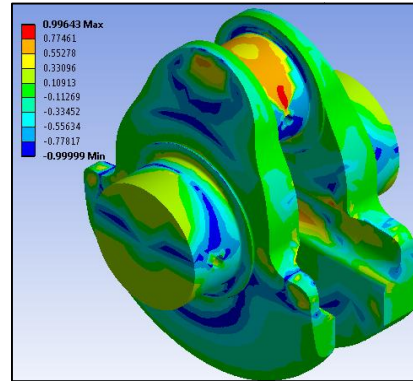


Fig. 15. Contour plot of biaxiality indication

Fig. 16 shows, the S-N curves comparison for the three different fatigue life prediction approaches. It can be seen that the strain-life approach is overall most conservative for fatigue life prediction.

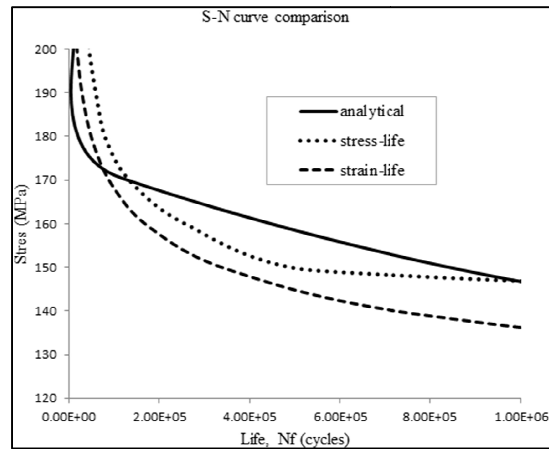


Fig. 16. S-N curves comparison

8. CONCLUSIONS

Crankshaft is one of the most critically loaded components of internal combustion engine and experiences cyclic bending and torsion loads during its service life. For this crankshaft, the maximum load occurred at the orientation of 365 degrees of crank angle. In addition, only bending loading is employed at this angle to the crankshaft [2]. Fillets in the crankshaft act as stress raisers and endure the highest level of stress under service loading. These are critical locations which is due to high stress gradients in

these locations, where consequently it resulted in high stress concentration factors. The maximum stress occurred on the crankshaft fillet at the angle of 365 degrees of the crank. The torsion loading in the overall dynamic loading conditions considered has no effect on Von Mises stress at the location of critical stresses. The torsional load influence on the stress range is also relatively small at other locations undergoing torsional load conditions. Accordingly, the analysis of crankshaft can be clarified that it is more suitable only for bending load condition. The location where the starting of the crack growth and fatigue failure are located at one of the crankpin journals fillets areas. However, it is observed from the contour plot of biaxiality indication in ANSYS, uniaxial is mostly affect the crankshaft critical areas stress. Regarding, fatigue life prediction, the strain-life approach gives the overall most conservative fatigue life results for mid to high cycle fatigue compared to other methods. For low cycle fatigue, stress-life approach shows more conservative fatigue results compared to strain-life approach.

COMPETING INTERESTS

Author has declared that no competing interests exist.

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