Microstructure and fatigue fracture mechanism for a heavy-duty truck diesel engine crankshaft

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Abstract
The main goal of this research is the experimental and numerical study on the fatigue function and failure of the crankshaft of diesel engine of a heavy truck. To do this, a crankshaft of the diesel engine of a heavy truck that has gone under failure after traveling 95500 km, has been used. The crack has developed from the third crankpin leading to the final failure in the fillet of the 4th crankpin. To examine the sources of this failure, several experimental studies have been carried out. Besides, using three dimensional finite element method, the location of the maximum stress in the crankshaft was determined using the “complete crankshaft model” and “one crank model”. Using the results of stress analysis, was a basis for the crack growth model and fatigue life estimation to determine the stress intensity factor and fatigue life considering the related parameters and boundary conditions method. At the final stage, using the results gotten from the given model for the fatigue crack growth, comparing it with experimental results, and examining the whole process, it was concluded that the scratches in crankpin region, was the main reason for the fatigue failure got from bending-torsional load-combination.

Keywords: Fatigue crack growth; Fractography; Crankshaft; Finite element; Fracture mechanics.

1. Introduction
On the recent decades, by the emerge of numerical and non-destructive tests as well as powerful analytic software models, there have been numerous researches on the fatigue crack growth, as the main source of the failure of the mechanical parts [1-3]. The failure of crankshaft in the internal combustion engines is due to various factors such as insufficient oil between bearings of the main
journal and the crankpin, the emergence of crack between the crankpin-web fillets, the misalignment of the crankshaft, the failure of fatigue got from torsional and bending loads and the impulses of combustion. The initiation of crack and its propagation are most often in the regions that the stresses are their maximum level, unless there is an external factor such as impurities in the material or the scratch of the crankshaft. Therefore, it can be concluded that determining the location the stresses created in these regions is useful to examine the non-destructive tests and the methods of life estimation [4-5].

Fonte et al. [6] have focused the effect of a steady torsion on fatigue crack growth under rotating bending. Results compared for the same stress ratio and relevant differences on fatigue crack growth are commented based on crack closure concepts. Fonte et al. [7] have presented fatigue crack growth on rotating bending axles and shafts with or without an applied steady torsion. The results showed that crack growth rates decrease with increasing mode III for cyclic mode I (ΔK_I) + static mode III (K_{III}) loading. Fonte et al. [8] have compared short crack growth rates with long crack growth rates on cylindrical test specimens under rotary or alternating bending combined with steady torsion on power rotor shafts. Results showed a significant reduction of the crack growth rates when a steady torsion Mode III is superimposed to cyclic Mode I. Martins et al. [9] have presented, mode III and mode I fatigue loading tests on standard compact tension (CT) specimens. Fracture surfaces obtained under mode I loading using optical devices revealed that crack grew flat, coplanar and normal to the tensile axis, as expected. Also, SIFs present at tip of each branch crack loaded under mode III were calculated through FEA. Masoudi Nejad [10] has carried out extensive researches on the growth of fatigue crack due to the residual stresses in mechanical parts. Masoudi Nejad et al. [11-14] have examined the growth of fatigue crack and life estimation of bandage wheel of the railway got from contact stresses and heating residual stresses. The results showed that the growth of crack in the wheel of Iran railway system as the shear combined modes II and III without residual stress. Masoudi Nejad [15] has carried out another research on examining the stress field got from production procedure of the railway wheel. To do this, he constructed a three-dimensional elastic-plastic finite element model to estimate the given stress field. The comparison of the results got from this study with laboratory experimental results shows the expected standards. The existence of the notches or the stresses concentration in the crankshaft is unavoidable due to the variation of crankshaft diameter. To decrease the stress concentration, the fillets have been used.

The crack initiation in the crankshaft has been identified and the main origin of the crack is commonly close to the crankpin-web fillet or main journal fillets. Another source of the crack is using the improper fillets with improper radius or improper improvement in crankpin connection and fillets of the main journal. As the crankshaft is an important part in a vehicle, there have been lots of studies numerical analysis and the main source of crankshaft failure [16-18]. Moore et al.
[19] have studied and analyzed on the source of continuous failure of two cast steel crankshafts and drew the following conclusions: the woody fracture has initiated from the second crankpin perpendicular to the direction of piston load. Location, orientation and size of the woody fracture were alike. The fracture region was in the direction of applied stresses caused the secondary cracks. The woody fracture existed before final machining and heating operation. The segregated regions contained MnS impurities, higher carbon, and tampered martensite bands all over the crankshaft. The crack growth from the fillets of the woody fracture plan from the hardened region continued with low-cycle fatigue. The propagation of fatigue crack was affected by the combination of torsional and bending loads. Farrahi et al. [20] examined the fracture analysis of some crankshafts of four-cylinder diesel engine. The results of scanning electron microscopy (SEM) revealed the cleavage fracture proving that it is of brittle fracture type. The results of FEM showed that the most vulnerable point is related to the fillet of the last crankpin (the closest to the flywheel). Considering the experimental and numerical results suggested re-examining the design and make of the crankshaft. Fonte et al. [21] evaluated and analyzed the failure of two crankshafts of agricultural diesel engine. Each of the two crankshafts had undergone different crack initiation and the fatigue was the main source of both crankshafts failure. Crankpin of both crankshafts was under the reversed bending load and the mode I was the cause of failure. Crack growth in each crankshaft was from the region of the crankpin-web fillet and the form of symmetrical half-elliptical cracks indicated mode I. The results of FEM also showed the fillets of crankpin-web were the most vulnerable part. They suggested making the crankshaft based on the experimental and numerical results, re-examining the design and quality control.

On the matter of examining related parameters on fatigue crack growth such as initial crack angle and loading type have been studied in the Refs. [22-25]. On the matter of fatigue crack growth being affected by residual stresses in bandage wheels, Masoudi Nejad et al. [26-29] examined it concluding that dealt with fatigue crack growth and life estimation of railway bandage wheel being affected by contact stresses and heating residual stresses. The results of these researches showed that the growth of crack in the wheel of Iran railway system as the shear combined modes II and III without residual stress. Fonte et al. [30] studied the failure analysis of two crankshafts of diesel engine. The first crankshaft broke down on the third crankpin having functioned 1100 hours and the second crank breakdown after 105000 hours functioning in the first crankpin. Crack initiation is related to the crankpin-web fillets. The form of cracks was half-elliptical symmetrical type and failure was mode I. Alfares et al. [31] examined the bus that had traveled 300000 km and analysed the breakdown of six-cylinder diesel engine of this public bus. Crack initiation in the crankshaft is related to the fillets of both crankpin and the main journal. They concluded that removing the partial
nitride layer of both crankpin and main journal decreases the fatigue strength leading to crack initiation and propagation. The crack source and the form of symmetrical half-elliptical cracks indicated mode I. Silva [32] has studied on the two damaged crankshafts of diesel engine. Therefore, he carried out grinding operation on the crankshafts having traveled 300000 km. The results showed that wrong grinding operation has caused journals damage leading to small thermal fatigue crack initiation. Becerra et al. [33] analyzed the failure of the two crankshafts of diesel engine. The first crankshaft started failure having traveled 1100 hours in the third crankpin and the second crankshaft experienced failure having traveled 105000 km in the first crankpin. Crack growth in both crankshaft was related to the crankpin-web fillets. The form of the cracks was symmetrical half-elliptical under opening mode I. Aliakbari et al. [34] examined the failure of the crankshafts of wheel loader diesel engine. The crankshaft started failure having traveled 4800 hours in the fifth crankpin. Crack growth in crankshaft was related to the crankpin-web fillet root. The morphology of fracture surface showed that the fracture is of the smooth type and has occurred due to the fatigue. Aliakbari [35] has carried out another research on examining the light-duty truck diesel engine crankshaft failure. The failure occurred by fatigue crack growth which was initiated from a surface defect after about 95000 km on the second crankpin from the crankpin-web fillet where the stress concentration was at the highest level.

Most of the work on crack growth is related to crack initiation prediction. Some are also related to crack propagation. However, crack growth issue while considering stress field has never drawn attention. In this article, the maximum stress in the crankshaft was simulated while considering parameters leading to the failure of the crankshaft of diesel engine of a heavy truck. Then, the results were studied to be used as a basis for fatigue crack growth using boundary conditions method. In order to determine, the cause of failure, several experimental studies including determining chemical composition, the strength of the material and microstructure of crankshaft were carried out.

2. Material and Methods

The chemical composition of the crankshaft of diesel engine of a heavy truck was carried out using ASTM E415-14 & ASTM E1086 standard by spectrophotometer machine of SPECTROMAXx model made in Germany. Its geometrical dimension was determined using micrometer and Verner Caliper made by Mitutoyo company of Japan. Then some microscopic observations were done using 8MP AF Samsung camera the size of 1.22 µm. To examine hardness value and the microstructure of crankshaft cross section of second crankpin having the thickness of 15 mm of crankpin of the crankshaft was cut off using wire cut EDM machine, then it was sandpapered using
rough and soft sheets on the sample. To do the final polishing, diamond paste was used, the washing was completed by alcohol and it was dried using hot air.

To examine the mechanical properties of the crankshaft such as the ultimate stress, the yield stress, elongation the samples were cut off by wire cut EDM machine, and then it was undergone machinery using CNC turning. The samples were used following ASTM E8M-97a standard and the method of doing the experiment were on the basis of the Ref. [36-37]. The simple tension test was done by Zwick model 4100/425 servohydraulic machine having the capacity of 25 tons. Figure 1 shows the test machine, tension test sample and extensometer. The results of the test, with the sampling rate of 2 times in a second, in the form of force-displacement was stored. The displacement changes of the samples were done by extensometer with the length measurement of 50 mm. To do the test, the displacement rate of the grips to each other was 0.05 mm per minute. Figure 2 shows the standard samples of tension test before and after the experiment.

To observe the microstructure, immersion etching using Nital 2% and microscope images by optical microscope from different parts of cross-section of the crankpin was carried out. The hardness test was performed based on the ASTM: E384-11e1 standard by hardness measurement device applying 30 kg force (294.18 N) in 10 seconds. To examine the broken surface, first, the broken sample of the crankshaft was cut off by wire cut EDM machine and the sample was put in ultrasonic bath for an hour. Then the images were prepared using SEM, LEO model 1450VP made by Germany.

3. Experimental results and discussion

The sample case being studied in this research is to examine the crankshaft failure of six-cylinder heavy-duty truck diesel engine that its characteristics has been shown in Table 1. Figure 3 shows the failure location of the middle point of the 4th crankpin to the 3rd crankpin fillet having traveled 955000 km. Its measurement specifications have been listed in Table 2.

3.1. Chemical analysis
The chemical composition of the elements of the material of broken crankshaft compared with the applied standard specification based on the weight percentage has been listed on the Table 3. The quantity of determinant elements on the Table 3 is equivalent to steel 38MnVS6 / DIN1.1303 in Germany DIN standard.

Placement of Table 3

3.2. Tensile Behavior
Mechanical properties of the material of the crankshaft compared with the applied standard specification has been listed in Table 4 and the diagram of engineering stress-strain of them are shown in Figure 4. The quantities got from the tensile tests is within the frame work of applied standard specification equal to the DIN1.1303 / 38MnVS6 steel.

Placement of Table 4
Placement of Figure 4

3.3. Hardness
As Figure 5 shows, the hardness depth is 5 mm. In order to increase the fatigue life, the compression stresses are used in the surface of the part. Nitriding and carburizing is one of these methods. Nitriding of the surface is preferred to carburizing due to its low temperature and provides the good dimension tolerance for the final surface [38].

Placement of Figure 5

Figure 6 shows, as the results of hardness test were expected, the surface of the part was undergone surface hardness, and it has tinier microstructure than central part. Similar investigations on hardness (e.g. [3,17]) reported the existence and increasing trend from the surface towards the center in the crankpin journal by case-hardening treatment. Due to observing the microstructure of crankshaft in Fig. 6, the matrix is formed by ferrite-pearlite that is pointed in Refs. [30].

Placement of Figure 6

3.4. Fracture Surface
Locating the source of fracture is the first goal of fractography science. It is needed for the proper analysis of the cause of fracture. The signs of the formed fracture are similar to the map of a road that is made to evaluate the fracture. The initiation and propagation of crack, make specific signs of the fractured surface such as river signs, radial lines, chevrons, beach signs, showing crack direction. The signs are used to ideate the source of crack. The appearance of such signs on the
fracture surface is a factor following the type of tension, shear, bending, fatigue or torsion loading. It also follows stress mode, the magnitude of stress, the existence of stress concentrators, environmental and material factor. One of the macroscopic features of observing fatigue fracture is related to appearance of the signs of ratchet especially in the shafts. These signs appear in places that several fatigue crack origins grow and then they are linked to each other. The signs of ratchet are steplike junctions between adjacent fatigue cracks [39]. Figure 7 shows the fracture location of sample crankshaft, the scratch location, crack initiation and crack propagation is observed. The scratch can be produced due to factors such as over loading, probable misalignment of the crankshaft on journal bearings, severe wear, the entrance of impurities to lubricating and the increase in the temperature of lubricating.

Placement of Figure 7

The sample of the fracture surface of the crankpin, the crack initiation location, the crack propagation and also the final fractured location has been shown in Figure 8. The fracture surface morphology shows a smooth and at crack initiation with the beach marks and ratchet marks and second crack propagation zone with beach marks and final fast fracture zone near the end. According to the beach marks orientation, it can be concluded that fracture starts from the middle part of the crankpin of crankshaft by arrow. To locate, the source of fracture of Figure 8, SEM images with different magnified sizes were prepared. The image of fracture surface by SEM, shows the direction of intergranular crack growth (Refer to Fig. 9). Besides, it should be mentioned that the impurities observed in the images were negligible.

Placement of Figure 8
Placement of Figure 9

4. Evaluation of fatigue crack growth

4.1. Stress Analysis

Stress analysis of the crankshaft for the purpose of determining the level of stress concentration in fracture region is done using finite element method (FEM). The analysis of stress of the complete crankshaft and camshaft model have been studied in the Refs. [40-42]. The stress analysis of the one crank model was studied in the Refs. [43-44]. Both models are studied in this research. Due to the crankcase oil temperature in diesel engines is in the range of 90-110 °C, thus according to Refs. [45-46], in this temperature range, about 10% of the yield and tensile strength are reduced. On the other hand, according to Refs. [20, 41], the maximum stresses of the torsional and bending forces are about 15% and 30% of yield strength occurring in the crankpin fillet regions. It seems that
achieving mechanical properties of engineering stress-strain diagram at ambient temperature is similar to true stress-strain diagram considering the temperature ranges and the maximum stresses. First, the geometry of the crankshaft is simplified and then, meshing, element modeling, applying boundary constraints, the kind of applying load, and stress analysis were done. In this research, bending (F_x), torsional (F_y), and longitudinal (F_z) force directions are mentioned in crankshaft coordinate geometry. The simplification on the crankshaft is as follows: (1) the neglecting oil ways in the direction of main journal and crankpin bearings; (2) the drilled holes in the counterweight in order to balance the crankshaft; (3) the neglecting of the crank web slope, the thickness of the crank web in the current model has been considered on the same level. The presence of these cases complicates the geometry of the crankshaft. It does not have any effect on the stress at critical locations [41]. The quadratic tetrahedral and triangular elements to mesh the geometry of the crankshaft was done according to Figure 10. These elements are usually used for meshing of complicated geometries [47]. Therefore, due to complicates the geometry of the crankshaft, the results of linear elements have less accuracy. Using quadratic elements increase the accuracy and decrease the rigidity of the geometry. Mostly triangular elements are used in the areas with high stress gradients and mostly large tetrahedral element are used in the zones that are farther away and are not within applying forced zone. Considering altogether, 482920 elements have been used for the complete crankshaft model and 263387 elements were used for the one crank model. The number of the complete crankshaft elements was 1.83 times the one crank model and this causes the lengthening of the problem solving time.

Placement of Figure 10

According to Ref. [44] the front side of the crankshaft has been fixed in all degrees of freedom directions; however, the end side of the crankshaft has been fixed in two directions. The distribution of the load on the connecting rod bearing on the uniform pressure of 120 degrees on the contact area was done. The load distribution was done by Webster et al. (1983) based on experimental test [48]. Considering the combustion pressure and cylinder diameter in Table 1, the applied force on the bearing is 104 KN in the F_x direction. While the crankshaft is rotating, the crankshaft on the crank is affected by different forces. To analyse the conditions of different loadings, to determine dangerous locations, three analysis of the stress of pure bending loads, pure torsional, bending-torsional loads in different angles, were done. Having examined the case, only the pure bending loads and perpendicular to the crankpin has the maximum applied stress and the other loads are negligible. The stress distribution of von Mises of the complete crankshaft model and the one crank model has been shown in Figure 11 during the combustion. The both have the same quantity and the difference
is negligible. The stress distribution of von Mises of in both crankpins has been shown in Figure 12 during the combustion. In Figures of 11-12 the stress distribution of von Mises in the crankpin fracture zones is 344.4, 261.06 MPa respectively. This is equal to 55 % and 22% of the yield stress of crankshaft material. The most stress occurs in the crankpin-web fillet. However, crack initiation is in the middle part of the crankpin as shown in Figure 4. Considering the results of FEM and images of fracture surface, scratches occurred in the middle part of the crankpin is the only factor responsible for fatigue failure of bending-torsional type. This caused the crack growth initiation.

Placement of Figure 11
Placement of Figure 12

4.2. Fatigue crack growth

To simulate fatigue crack growth, Frank 3D software has been used by boundary elements method done by the researchers in Cornell University [49-50]. The modeling and mechanical properties of the crankshaft is the same as pervious section. In Figure 13, boundary element model and its crack has been shown.

Placement of Figure 13

In this study, the geometry of crack in quarter-elliptical form in the plan got in stress analysis has been shown. The reason for selecting this form of crack is the comprehensiveness of the crack and the samples being observed in the fracture of the crankshaft. The initial crack length examined for different cracks is got with the dimension of 0.5, 1.5 and 2 mm. Having being loaded, the sample is being meshed. Then the stress analysis is carried out through using Boundary Element Solver (BES) software. Then the values of stress intensity factor are calculated for mode I. Having had the values of stress intensity factor to measure crack growth is determined. There are two possible crack growth in 3-Frank software crack growth: in manual and automated form. In manual crack growth the value of crack growth is optional. It should be mentioned that crack growth should not exceed 30% of the all previous cracks. The direction of crack growth is calculated based on the maximum tangential stress criterion. After determining the direction of crack growth it is possible to fix tip curve of the crack and let the crack grow to another stage. After the growth, the sample is meshed. Now it is ready to solve. This process should be repeated for each crack growth. Crack growth can be repeated up to 14 stages in order to get to the critical length of 38 mm. As the process continues, after getting the information and calculating the value of stress intensity factor, the fatigue life of crankshaft is calculated. The crack growth rate and fatigue life can be obtained by using the
equations in [51-53]. In this research the improved equation of Paris has been used and the effect of fatigue crack closure has also been taken into account. The growth rate based on this equation is defined as follows [54].

\[
\frac{da}{dN} = C \left( \Delta K_{eff} \right)^n = C \left( K_{max} - K_{op} \right)^n
\]  

(1)

In the above equation, \( \Delta K_{eff} \) is the effective stress intensity factor and is equal to difference between the maximum mode I stress intensity factor (\( K_{max} \)) and the stress level needed for opening the crack tip is \( K_{op} \). The factors of \( C \) and \( n \) in this equations are material constants. In this study are \( C = 3.89 \times 10^{-9} \) m/cycle and \( n = 3.12 \). The function of crack opening, \( f \) for plasticity induced closure of the crack by Newman is defined as follows [55].

\[
f = \frac{K_{op}}{K_{max}} = \begin{cases} 
\max(R_{ratio}, A_0 + A_1 R_{ratio} + A_2 R_{ratio}^2 + A_3 R_{ratio}^3) & R_{ratio} \geq 0 \\
A_0 + A_1 R_{ratio} & -2 \leq R_{ratio} < 0
\end{cases}
\]  

(2)

The coefficients of the above equations are obtained by the following equations.

\[
A_0 = (0.825 - 0.34\alpha + 0.05\alpha^2) \left[ \cos \left( \frac{\pi S_{max}}{2 \sigma_0} \right) \right]^{1/\alpha}
\]  

(3)

\[
A_1 = (0.415 - 0.071\alpha) \frac{S_{max}}{\sigma_0}
\]  

(4)

\[
A_2 = 1 - A_0 - A_1 - A_3
\]  

(5)

\[
A_3 = 2A_0 + A_1 - 1
\]  

(6)

In this equations, \( \alpha \) is a constraint factor for plane strain/ plane stress, \( \frac{S_{max}}{\sigma_0} \) is the ratio of maximum stress to its flow stress (the required stress to create plastic flow) and \( R \) is the ratio of stress to each load cycle. The number of cycles needed to crack growth from its initial length is \( a_0 \) up to the final length is \( a_f \). It can be calculated by the following integral equations as follows:
The results got for the stress intensity factor in crack tip has been shown in Figure 14. The results show that the history of stress intensity factor is for longer crack length. The longer the crack length; the difference of stress intensity factor of the crack and its length is uniformly increased. Based on the Paris equation, as the crack length is increased; the crack growth is increased. Figure 14 shows maximum stress intensity factor of the crack with longer length. As it is observed, the more the length of the crack; the more the difference in the stress intensity factor. In other words, the difference the stress intensity factor for the crack is 2 mm more than its difference in the crack with its initial length of 0.5 mm. Figure 15 shows the fatigue life got per the length of the crack to the different values of the initial crack length. It shows that as the length of the crack increases, the fatigue life decreases.

5. Conclusions

In this study, a 3D model has been presented to demonstrate fatigue crack growth and crankshaft life estimation of six-cylinder heavy-duty truck diesel engine. The results were obtained using numerical method and specialized software of boundary elements. The parametric analysis of the fracture was obtained by using linear fracture mechanics showing the influential factors such as crack length and loading for surface crack on the fracture done. The failure occurred in the 4th crankpin of the middle part of the crankpin to the fillet of the third crankpin-web having traveled 955000 km. Having examined the cases by optical microscope, it can be said that steel microstructure of ferrite-perlite type. The micrograph of the crankshaft after being etched and the results of hardness tests show that the surface of the part was undergone surface hardness, it has
tinier microstructure than central part. The numerical results and the fracture images show that creation of scratches in crankpin surface is the main reason for the bending- torsional fatigue failure. It in turn leads to crack initiation and its propagation. The numerical results show that the maximum stress is in the fillet in the third main journal of the crankshaft. The value of the stress in the third fillet of crankpin-web is more than the center of the third crankpin which the crack has been originated from.

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Table 1.

|                          |       |
|--------------------------|-------|
| Max. Power (kw/rpm)      | 290/ 1800 |
| Max. Torque (Nm/rpm)     | 1850 / 1080 |
| Bore (mm)                | 100   |
| Stroke (mm)              | 105   |
| Cylinder pressure (bar)  | 78.5  |
Table 2.

| Crankshaft Parameter         | Specification (Unit) |
|------------------------------|----------------------|
| Crankshaft mass              | 54.43 Kg             |
| Crankshaft length            | 590 mm               |
| Main journal diameters       | 108 mm               |
| Crankpin diameters           | 94 mm                |
| Operation                    | 955000 Km            |
Table 3.

| Symbol | DIN1.1303 | Standard alloy |
|--------|-----------|----------------|
| Fe     | 97.2      | Base           |
| C      | 0.353     | 0.34-0.41      |
| Si     | 0.59      | 0.15-0.8       |
| Mn     | 1.36      | 1.2-1.6        |
| P      | 0.013     | ≤0.025         |
| S      | 0.02      | 0.02-0.06      |
| Cr     | 0.174     | ≤0.3           |
| Mo     | 0.005     | ≤0.08          |
| V      | 0.096     | 0.08-0.2       |
Table 4.

| Symbol          | $\sigma_Y$ (MPa) | $\sigma_U$ (MPa) | Elongation (%) | Young’s modulus (GPa) |
|-----------------|------------------|------------------|----------------|----------------------|
| Standard alloy  | Min. 520         | 800-950          | Min. 12        | -                    |
| Current work    | 626.5            | 927.1            | 13.8           | 210                  |
Figure 1.
Figure 2.
Figure 4.
Figure 5.
Figure 6.
Figure 7.
Figure 9.
Figure 10.
Figure 11.
Figure 13.
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