

Failure Analysis of a Four Cylinder Diesel Engine Crankshaft Made From Nodular Cast Iron

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Abstract

The premature breakage in some four cylinder diesel engine crankshafts was reported. All crankshafts were failed from the same region. Failures had occurred in the first crankpin, the nearest crankpin to the flywheel. Dynamic analysis and finite element modelling were carried out to determine the state of stress in the crankshaft. FEM results revealed that the first crankpin fillet is the most vulnerable point to fracture. Soderburg diagram of the studied crankshaft showed that the service operation point, which stands for mean and alternating stresses of the critical region (first crankpin fillet) was located in the safe region. Therefore, it can be concluded that fatigue fracture has not occurred in the crankshaft. SEM images of the fractured surface also showed cleavage fracture and put in evidence that the failure was brittle fracture. No sign of fatigue failure was observed. The fracture may have been caused by an overload. However, the results suggest re-evaluation of the design and manufacturing. The fillet rolling may play an important role in this matter. Optimization of the fillet rolling process by changing process parameters has been recommended to the manufacturer. This recommendation has been adopted by the manufacturer and no further fracture has been reported since.

Keywords: Crankshaft, Failure Analysis, Fracture, Fatigue

1- Introduction

Engine and its components are the origin of most of the failures taking place in automotive [1]. Among them, failure of crankshaft is frequently reported in different engines and therefore, a fundamental understanding of its operation and failure mechanisms can be of a great value. Either an improper manufacturing and operation or mechanical fatigue may lead to a catastrophic failure in crankshafts. More detailed crankshaft failures can be attributed to high stress concentration produced at critical location, misalignments, defective lubrication, overheating during operation, overloading, metallurgical defect and bearing vibration.

Generally, improper operation results the fatigue crack to be initiated in the weakened area. The propagation of this crack until the remaining cross section could not tolerate the load, result in final fracture. A failure investigation of a nitrided crankshaft revealed that the partial absence of the nitride layer in the fillet region decreased the fatigue strength and eventuated in fatigue initiation and propagation in weaker region [2]. In some cases, failure is related to a material production problem. For example casting defect could lead to a sudden fracture after a very short period of usage [3]. In the case of a casted crankshaft that a break into two pieces had been occurred in the crankpin region, examination of the failure zone proved the absence of hardened case in the fillet region and the presence of free graphite in the microstructure, led to initiation and propagation of fatigue cracks [4].

Even if neither manufacturing defect presents nor improper operation takes place, there are high stressed regions where under a cyclic situation will be appropriate sites for fatigue crack initiation. Fatigue limit of the crankshaft material is exceeded in these regions due to high stress concentration [5]. An investigation of a catastrophic failure of a web marine crankshaft showed that the crack initiation started on the fillet of crankpin by bending stress while propagation was due to the combination of cyclic bending and steady torsion [6]. Fracture of a crankshaft due to cracks on the edge of the oil hole was also investigated. Friction between surface of the shaft and the main bush due to improper repairing and assembling and therefore, the introduced shear stress was the main cause of the fracture [7].

A short survey on previous studies showed that fatigue is the most common cause of crankshaft failure. Employing fatigue life improvement techniques such as fillet roll-

ing and shot peening can effectively increase the fatigue strength of crankshaft and prevent the catastrophic failures [8, 9]. Exploring the reliable criteria for assessment of fatigue life would also be very important. A comparison between different failure criteria in crankshaft showed that using surface crack initiation criterion compared to resonant shift or two pieces failure criterion decreased the apparent fatigue limit [10].

For assessment of a crankshaft fatigue life, one has to determine the stress state first. This could be carried out either experimentally by inserting the strain gauges into areas [11] or numerically by employing the finite element simulation [12]. Nowadays, with a sharp progress in the computer powers, finite element is effectively used for stress state determination in crankshafts. However applying simple loading condition which is far from the reality has oversimplified the result. A dynamic lumped model linked to a finite element model was used to investigate the stress state of a fractured crankshaft [13]. Combination of finite element and dynamic analysis provides more accurate simulation of what a crankshaft experiences during operation. According to the results, different zones where the maximum stress is reached were reported. However, microstructure investigations revealed that, among these areas with the same failure probability, fatigue fractured occurred where a martensite structure with very high hardness, distributed in a thin layer, was formed [13]. Fracture surface of a number of broken compressor crankshaft showed no sign of beachmarks but shows the brittle appearance [14].

The present work investigates possible reasons for the failure of four cylinder diesel engine crankshaft. The premature breakage in some diesel engine crankshafts due to high speed engine and sudden stop was reported. Some crankshafts also failed the durability test. All crankshafts were failed from the same region. Failures had occurred in the first crankpin, the nearest crankpin to the flywheel.

2- Experimental procedure

2.1 Material Properties

The crankshaft was made from GGG 70 ductile cast iron. Chemical analysis of a sample taken from fractured crankshaft was carried out using a spectrometer. Table 1 gives the chemical composition of the crankshaft material. Spectrometry analysis revealed that the material is GGG

70 ductile cast iron.

Table 1. Chemical analysis of crankshaft

C	Si	Mn	Cu
3.4-3.6	2.2-2.4	0.2-0.5	0.5-0.8

Tensile test of a standard specimen taken from fractured crankshaft was also conducted in order to obtain yield and ultimate tensile stress of the material. The specimen gauge length and cross section were 50 mm and 123.7 mm² respectively. The mechanical properties are shown in table 2. It was observed that the tensile properties are within the expected range.

Table 2. Mechanical properties of the material

Microstructure	Surface hardness HRC	Elongation %	Yield Stress MPa	Ultimate tensile strength
Perlitic	57	2	402	675

2.2. Fractured surface examination

A broken crankshaft is shown in Fig. 1. The Fractured surface shows a brittle appearance and no Beachmark was observed on the failed region. Fig. 2 shows the microstructure of the material. spherical nodules of graphite surrounded by ferrite in a perlitic matrix is observed.



Fig 1. The fractured surface of the crankshaft

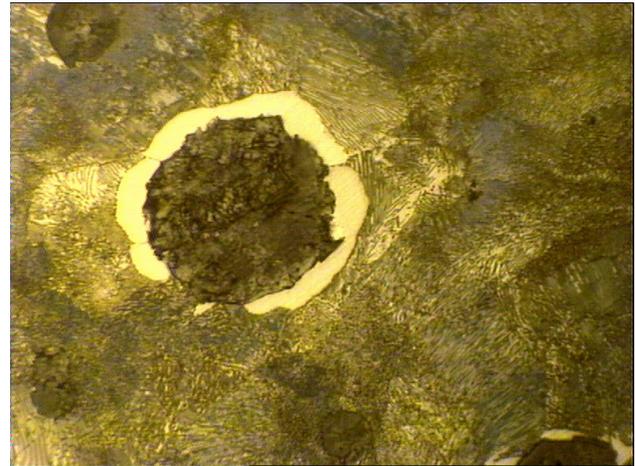


Fig 2. Optical micrograph of the material

Figs 3 and 4 show SEM images of the fractured surface. The figures show cleavage fracture and put in evidence that the failure was brittle fracture. These two pictures are representative of the entire fracture surface. We could not observe any sign of fatigue failure. Therefore, fatigue is eliminated from the list of possible causes of the failure.

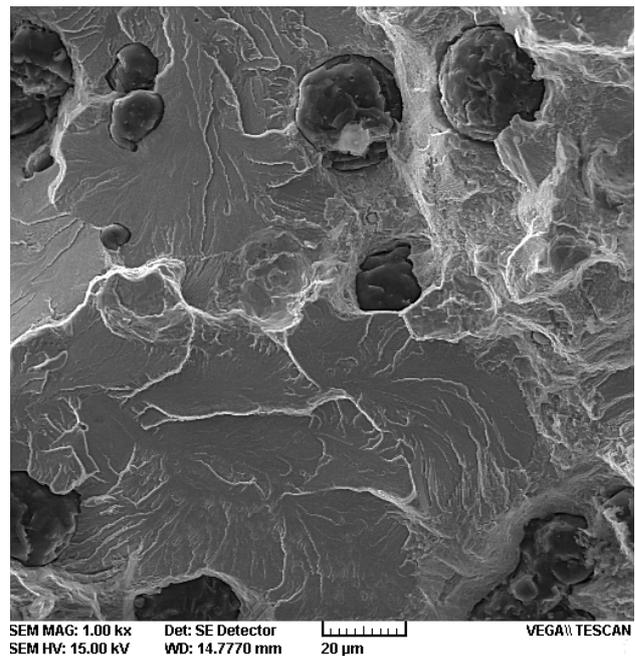


Fig 3. SEM image of the fractured surface

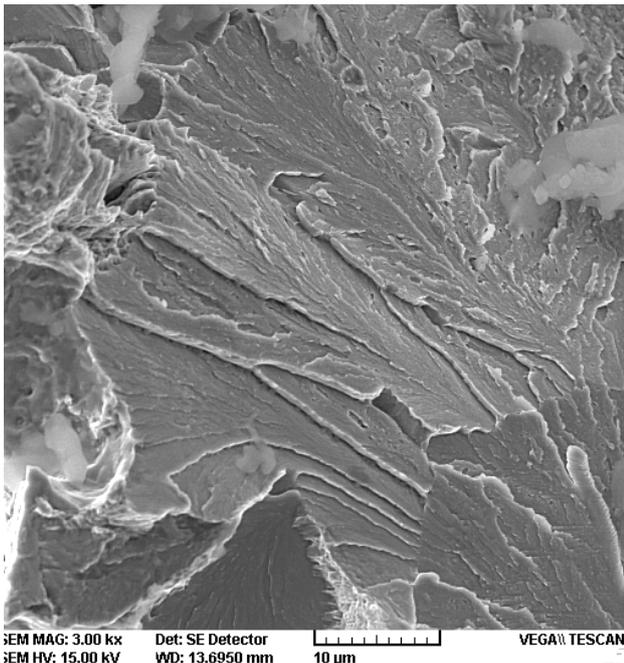


Fig 4. SEM image of the fractured surface

3- Life assessments

3.1 Stress State Determination

Stress state of crankshaft during service condition was determined to find out whether the fracture has originated from an unexpected high stress point under fatigue condition or not. Two main sources are responsible for the present forces on a moving crankshaft. The pressure developed on the top of the piston during combustion is the first one which is transmitted to bearings through connecting rod. Inertia forces due to rotating movement of shaft, reciprocating movement of piston and combined movement of connecting rod are the other one. In order to determine the crankshafts stress state more profoundly both types of forces must be taken into account. To this end a dynamic analysis was used to obtain the equivalent pressure in the journal bearings. The result was then recalled in a finite element method to determine the stress state in the crankshaft.

3.1.1 Dynamic Analysis

Dynamic analysis of a crank and slider can successfully consider the combine effects of combustion pressure and inertia forces. This three member mechanism is shown in Fig.5. The maximum pressure of 46.2 bar was expe-

rienced in the cylinder during combustion. This pressure was exerted on the top of the piston in the dynamic analysis. Since the amount of pressure in suction, exhaustion and compression is very low in comparison with combustion pressure, it can be reasonably neglected in analyses of these three stages. Piston diameter and mass were 88.3 mm and 505 gr, respectively. A 181 mm, length connecting rod has been connected into both piston and crankshaft. Connecting rod's smaller and bigger hole diameters were 20.948 mm and 53 mm respectively. Its mass was considered 740 gr in the dynamic analysis. Engine revolution of 30000 degree per second has been considered as angular velocity of the crank in the dynamic model.

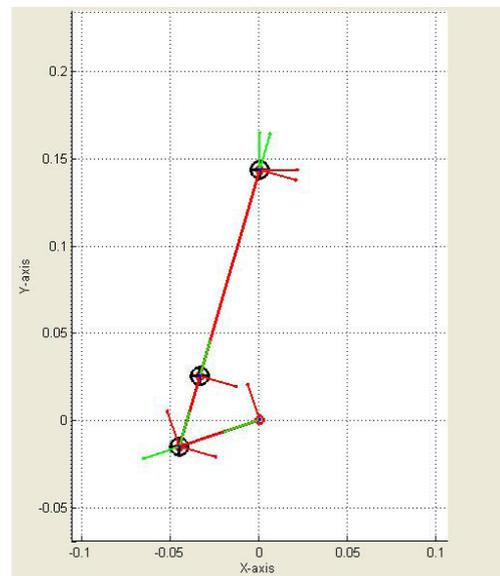


Fig 5. Mechanism used in dynamic analysis

Dynamic analysis of this configuration was carried out in each angular position of the crank. Calculation of the time history of resultant reaction in the journal bearing in which connecting rod and crankshaft are connected together was the main concern of the dynamic analysis. Having the amount of the reaction forces in journal bearing in hands, one can easily calculate the equivalent pressure which is exposed on bearings in each stage of the engine service by equation 1.

$$p = \frac{F}{2rW} \quad (1)$$

Where $F_{\text{resultant}}$ is the reaction force in journal bearings, r and w are the bearings radius and thickness respectively and p is the equivalent pressure. Radius and thickness of

the studied bearings were 30 mm and 22.16 mm respectively. Table 3 summarizes the results of dynamic analysis for each engine cycle.

Table 3. Result of dynamic analysis

Cylinder Number	Pressure (MPa), during 1 st cylinder combustion	Pressure (MPa), during 2 nd cylinder combustion	Pressure (MPa), during 3 rd cylinder combustion	Pressure (MPa), during 4 th cylinder combustion
1	-6.23	11.3	11.3	15
2	11.3	-6.23	15	11.3
3	11.3	15	-6.23	11.3
4	15	11	11.3	-6.23

3.1.2 Finite Element

A three dimensional finite element model was constructed to assess the stress state in the fractured crankshaft. The model is illustrated in Fig. 6. Twenty noded finite elements were used to discretize the crankshaft. Finer mesh was used in the regions of fillet where severe stress gradient is expected to be present. Linear elastic material behavior was considered by introducing Young's modulus of 178 GPa and Poisson's ratio of 0.3. There are five fixed journals in the studied crankshaft which only can only rotate around their own axes. These journals were constrained in the radial and axial directions in the finite element model. The first journal was also fixed in the axial direction to prevent the crankshaft to move axially. The calculated equivalent pressures from the dynamic analysis were recalled here and were distributed on the surface of remaining journals for the analysis of each engine cycle. Torsional torque that is exerted to crankshaft during its operation was calculated by equation 2.

$$T = \frac{P}{w} \quad (2)$$

In this equation P is the power and w represent the engine revolution. Accordingly, the amount of 128 N.m torque was applied to the extreme surfaces of the crankshaft.

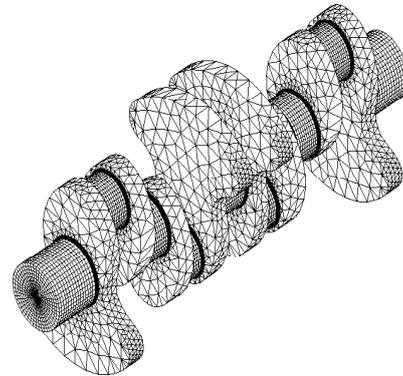


Fig 6. Finite Element Mesh.

Four cases, i.e. combustion in each cylinder, were examined by alteration of the boundary condition in the finite element model. Results revealed that the most dangerous case has occurred in the first crankpin when the third cylinder was combusted. The equivalent von Mises stress distribution of the whole crankshaft during combustion of third cylinder is shown in Fig. 7. The maximum equivalent stress is 62.7 MPa which is about 15% of the yield stress of the material.

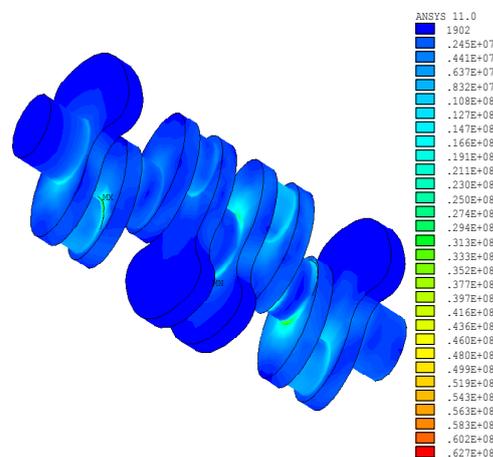


Fig 7. Equivalent von Mises stress distribution

3.2 Prediction of the fatigue life

Finite element results revealed that the stress state of the crankshaft during operation falls into the elastic region. Therefore, stress based approach was utilized for assessment of the fatigue life. The calculated fatigue limit for the standard specimen of crankshaft material was 293 MPa,

(equations 3 and 4) [15].

$$S'_e = (0.61 - 0.00026 S_{ut}) S_{ut} \tag{3}$$

$$S'_e = 293 \text{ MPa} \tag{4}$$

This amount however, has undergone some modifications according to equation 5 in order to be applicable for the crankshaft's geometry, loading and surface finish.

$$S_e = (k_a \times k_b \times k_c \times k_d \times k_e) \times S'_e \tag{5}$$

In this relation S'_e is the fatigue limit of the standard specimen while S_e is the fatigue limit of the crankshaft. K_a , K_b , K_c , K_d , and K_e are the correcting factors of surface finish, size, loading mode, operation temperature and reliability. Relations for obtaining these factors could be found in [16]. Table 4 shows the amount of correcting factors for the case of the present study. According to these corrections the fatigue limit of the crankshaft was obtained 180 MPa.

Table 4. Fatigue limit correcting factors

K_a	K_b	K_c	K_d	K_e
0.802	0.794	1	1.02	0.814

Knowing the fatigue limit and the yield stress of the material one can easily draw the Soderburg diagram. In the Soderburg diagram, the abscissa represents the amount of mean stress and the ordinate represents the amount of alternating stress. The region which is enclosed by abscissa, ordinate and the straight line between yield stress and fatigue limit is the safe region. In another words, if the point of service operation falls into this region, fatigue fracture will not occur. Fig. 8 shows the Soderburg diagram of the studied crankshaft along with the service operation point. This point stands for mean and alternating stresses of the first crankpin fillet that had been recognized as the most vulnerable point to fracture according to finite element results. Since the point was located in the safe region, it can be concluded that fatigue fracture has not occurred in the crankshaft. Calculations showed that the safety factor against the fatigue fracture was 4.77.

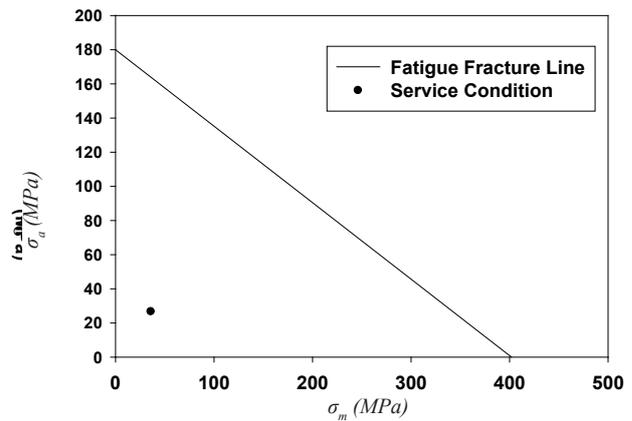


Fig 8. Soderburg diagram of the studied crankshaft

4- Discussion

Dynamic analysis was carried out in each angular position of the crank. The equivalent pressure which is exposed on bearings in each stage of the engine service was calculated.

The equivalent pressures were recalled in finite element model and were distributed on the surface of remaining journals for the analysis of each engine cycle. FEM results revealed that the most dangerous case has occurred in the first crankpin when the third cylinder was combusted. The maximum equivalent von Mises stress is 62.7 MPa which is about 15% of the yield stress of the material. Therefore, stress based approach was utilized for assessment of the fatigue life. The fatigue limit of the crankshaft, considering all correction factors, was obtained 180 MPa.

Soderburg diagram of the studied crankshaft showed that the service operation point, which stands for mean and alternating stresses of the critical region (first crankpin fillet) was located in the safe region. Therefore, it can be concluded that fatigue fracture has not occurred in the crankshaft.

SEM images of the fractured surface also showed cleavage fracture and put in evidence that the failure was brittle fracture. No sign of fatigue failure was observed. The fracture may be caused by an overload. The above explained analysis showed that an overload could not be probable. Therefore, the problem lays in design procedure. The fracture which occurred from the fillet, suggests re-evaluation of the design. The fillet rolling may play an important role in this matter. Optimization of the fillet rolling process by changing process parameters has been recommended to

the manufacturer. This recommendation has been adopted by the manufacturer and no further fracture has been reported since.

Conclusion

Failure analysis of broken four cylinder diesel engine crankshafts was carried out and the following conclusions are drawn from this work.

- i. FEM results revealed that the first crankpin fillet is the most vulnerable point to fracture.
- ii. Soderburg diagram of the studied crankshaft showed that the service operation point, which stands for mean and alternating stresses of the critical region (first crankpin fillet) was located in the safe region.
- iii. SEM images of the fractured surface also showed cleavage fracture and put in evidence that the failure was brittle fracture
- iv. The results suggest re-evaluation of the design and manufacturing.
- v. Optimization of the fillet rolling process by changing process parameters has been recommended to the manufacturer. This recommendation has been adopted by the manufacturer and no further fracture has been reported since.

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