Journal of Multidisciplinary Engineering Science and Technology (JMEST)
ISSN: 3159-0040
Vol. 2 Issue 2, February - 2015

Analyzing and Identifying Various Approaches for Crankshaft Failures

A state-of-the-art Review

Pratik Kakade
P.G. Student,
Department of Mechanical Engineering
SRES's College of Engineering
Kopargaon, Maharashtra, INDIA
kakadepratik@gmail.com

Manish D. Pasarkar
Assistant Professor,
Department of Mechanical Engineering
SRES's College of Engineering
Kopargaon, Maharashtra, INDIA
mdpasarkar@gmail.com

***Abstract*— Purpose: The main objective of this paper is to critically review various papers related to analysis of crankshaft failure and identify different approaches in solving the problem. Design/methodology/approach: The paper critically examines 10 different papers related to analysis of crankshaft. The intention of the review is to find the different models used by various authors in solving the problem and study which method is suitable for various models. Findings: The review of various papers revealed which are the major factor for the fatigue failure and the means to check the cause of failure. Research limitations/implications: This research is just the comparison of different model used for analysis. It can specify which model is suitable for particular problem. Practical implications: The problem of crankshaft failure is very severe and cannot be eradicated. The review will help in choosing the method for solution**

***Keywords*— FEM, crankshaft, crankcase**

I. INTRODUCTION

The failure of crankshaft is a very severe problem faced by engineers recently. Most of the fracture is due to fatigue failure. Fatigue phenomenon is very difficult to understand and analyse and now a days treated extremely important as severe problems are encountered. There isn't any specific formula to find the fatigue life and hence obtaining exact solution is a myth. Lot of parameters are involved and needs to be considered for fatigue life calculation and thus making formulation of problem very complicated and tedious. As Fatigue phenomenon comes in the category of dynamic analysis measuring the variation of loads with available measuring devices is difficult and costly. Thus, a necessity to develop various simplified models is a prime concern in many industries. This paper intends to cover methods used in fatigue failure analysis of crankshaft and thereby critically suggest the improvised method to reduce its effect.

II. CRITICAL AND SYSTEMATIC REVIEW OF METHODS

The methodologies adopted to solve different cases of crankshaft failure are described here separately. This paper gives critical analysis including drawbacks depicted in Table 1. Various methods and testing in each paper are tabulated in Table 2. The ultimate aim of this paper is to throw some light on used methodologies and suggested improved methodologies on fatigue failure analysis in crankshaft failures.

A. Ktari et al (2011), carries out the investigation of the failure of three different crankshafts of 12 cylinders V-12 design engine used in trains. The engine run at nominal speeds of 1050 rpm and the periodic maintenance is carried after 40000 km. The crankshaft is dismantled and NDT testing is carried out on it to check any traces of cracks. Any mistake in determining the cracks can lead to catastrophic failure. All the three shafts were working under same working conditions. Chemical analyses of the fractured surface of all the specimens were carried out and the material was found satisfactory to the working condition. Tensile test also was carried out which provided required material properties. Hardness test unveiled that the hardness reduced steeply as measured towards the center from the surface and became steady after 1 mm. thus stating surface hardening due to thermal treatment of plastic deformation. Fracture toughness was also measured using standard Charpy V-notch (CVN) specimen. The average CVN shows that the fracture was brittle.

The first crankshaft was visually inspected at the fracture section to see the beach marks on the surface which justified that crack progressed slowly till the 60-70% of section until overload fracture. Fracture was near fillet region and the fracture plane was inclined at 450 with shaft axis. As fillet was very small, it became susceptible to high local stress concentration. Ratchet marks were found on surface which indicated multiple fracture origins [1]. Fracture was examined under Scanning Electron Microscope (SEM) (Fig. 1, d) to detect fatigue striation which was not detected. As the bearings were in good condition, problem of faulty lubrication is out of concern. Thus fracture was due to mechanical loads like bending and torsion. For the second crankshaft the crack was analysed which took place on the rough journal surface of the shaft. As the crack was far from fillet region, the causes should be different than the previous.

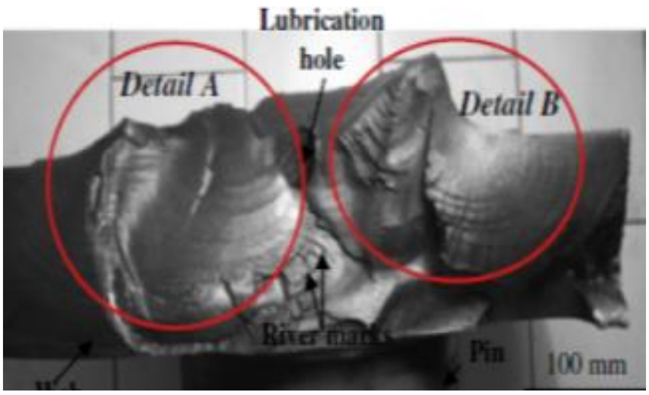
Rough surface predicts defective contact between

ISSN: 3159-0040
Vol. 2 Issue 2, February - 2015

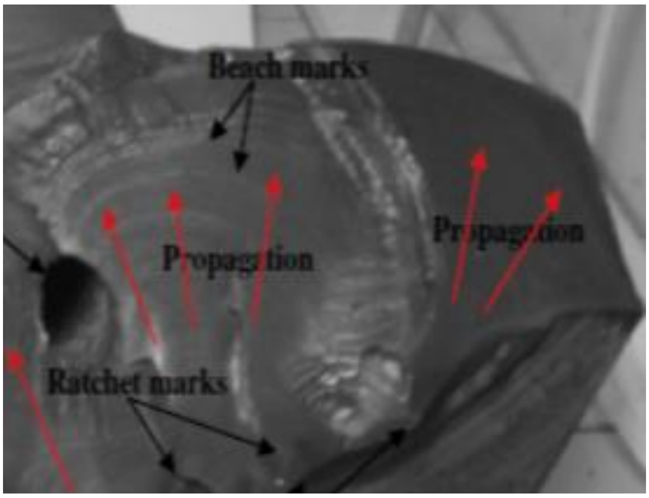
journal and bearing to be the cause of failure. Defective contact can be either due to the defective lubrication system, high oil pressures, temperatures and misalignment of the shaft or by thermal loads, and frictional forces resulting in bad contact. The heat that is generated due to friction causes thermal gradients in journal surface and creates thermal stress in heated areas. When the journal comes in contact with bearing, compression stress occurs and it cools for the remaining rotation where tension acts. This cyclic thermal tension and compression causes thermal fatigue on surface in the form of parallel cracks perpendicular to thermal gradient which was observed on the original shaft and thus might be the reason for failure. The third crankshaft was tested during the regular maintenance period with the magnetic particle testing and networks of cracks were observed on the journal no. 2. The feature of the crack stated it was initiated due to thermal fatigue.

After polishing the surface with abrasive paper, the crack disappeared. It shows that these thermal fatigue cracks were stabilized which was the result of shielding effect explained by the mechanical unloading generated by the presence of crack in the structure. Such phenomenon was found in number of application. Thus it was concluded that the failure of first shaft was due to mechanical loading. Second shaft failure was a consequence of defective lubrication or thermal fatigue or combination of both. A network of cracks was found on the third shaft which was the result of thermal fatigue. These cracks didn't pose any threat as they were of superficial nature. All the shafts were satisfactorily analysed for the cause of failure just by visual inspection.

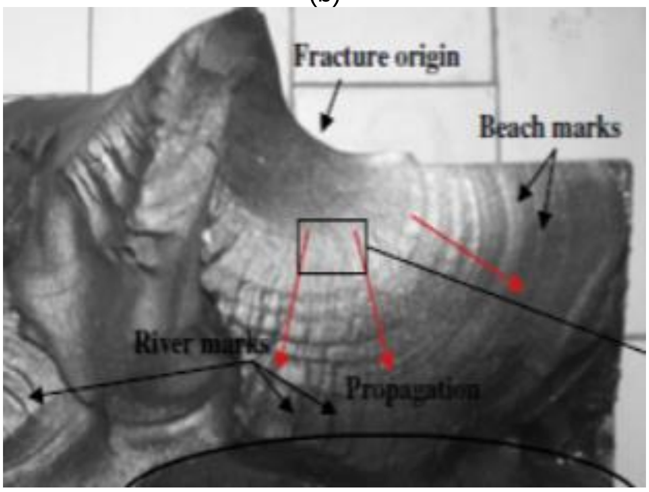
Even though the problem specified to the authors was specifically mentioned, validation of the stresses at various locations would have given more precise and accurate theory to solve such type of problems. Also, FEA model would have helped to calculate critical stresses at certain critical locations where cracks are observed. Visual inspection of crack propagation certainly gives exact solution but it could have been solved using



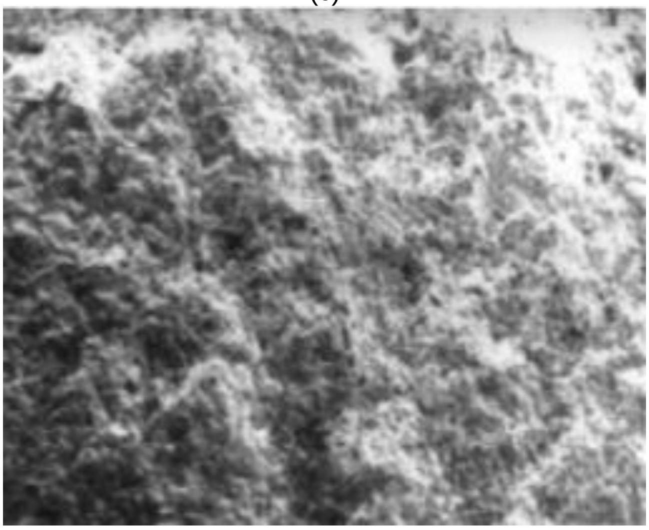
(a)



(b)



(c)



(d)

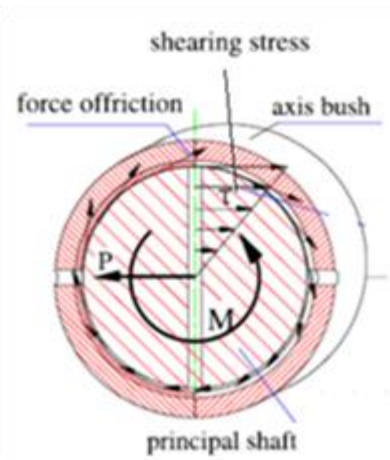
Figure 1: (a) Fatigue fracture crankshaft views; (b) details observed with a naked eye; c) details observed with a naked eye; (d) Benchmarks observed with SEM

Changli Wang et al (2005), carries out the analysis of the crankshaft which failed in a strange manner when it was under testing only for 20 min. Four cracks were found on the edge of oil hole. The reason for this failure is analysed in this paper using various methods. Friction was caused due to improper repairs which lead to failure. How did the crack initiate and expand is

Journal of Multidisciplinary Engineering Science and Technology (JMEST)
ISSN: 3159-0040
Vol. 2 Issue 2, February - 2015

discussed in this paper. The failed shaft was new and after repairing the crankcase it was attached to the engine and tested with no loads. Due to unusual noise the test was stopped after 20 min. Four cracks were found on the fourth main journal. Main bush was seriously damaged. Copper on inside of the bush was torn off completely. Some copper was stuck on the surface of the shaft. A sample from the specimen was chemically analysed and found the content of Mo and Mn lower than standard value but they didn't pose any appreciable threat. Hardness test was carried out throughout cross section and was found that the hardness of the surface was reduced due to heating. Under electron microscope it could be seen that the rains were coarse at the surface and finer as the distance increased from the surface of the shaft. Fractographic analysis revealed that the crack had originated from the edge of the oil hole and its orientation was parallel to the longitudinal section of the shaft. The cracks were formed in three zones. To understand the failure process, focus was given on friction between shaft and the bush (Fig 2). It was found that bearing hole were bored with boring machine which disturbed the alignment and so assembly was not in good condition which was the cause of friction. Improper lubrication was not the cause of friction by observing the other components which were found in good condition. Thus it was deduced that the cause of friction was improper repairs. Also the molten copper on the shaft also suggest that the temperature was above 6000C due to friction. The molten metal would have led to embrittlement in the crack-shaft journal [2].

As the process implemented by the author to find the initiation and propagation of crack is sufficient according to the scenario, still other probable causes of the crack initiation could have been discussed. Stress concentration factor could be calculated by FEM which might have been the reason of crack generation.



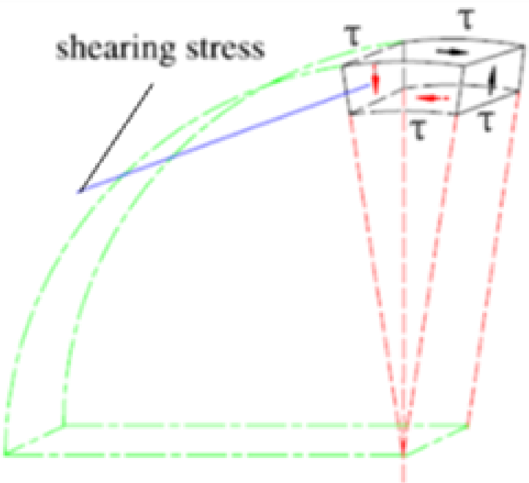
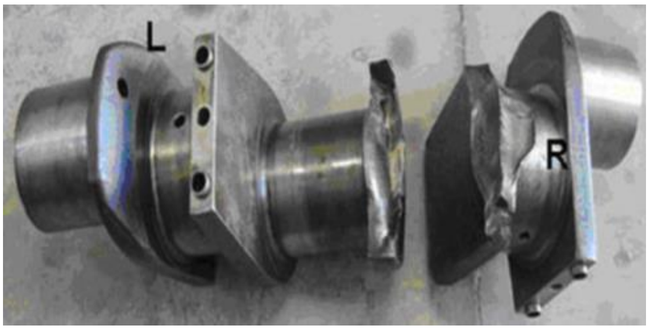


Figure 2: Schematic illustration of friction force between shaft and bush and stress distribution in shaft.

F. Jiménez Espadafor et al (2009), presents the study of the catastrophic failure of the crankshaft of the V16 diesel engine of a power plant for electrical generation which was running at 1500 rpm and failed after 20000 h of its continuous service operation on its maximum loading. Different study methods were carried out to study the reason of failure and each of them concluded fatigue as a cause of failure. This paper focuses on the methodology to identify the point of crack initiation on the surface of the crank. The crankshaft was made of low alloy steel forged as one, then machined and, finally, tempered. Chemical analysis was performed on the specimen to calculate the composition of the material. It was found out to be satisfactory for the functioning of the crankshaft. Standard cylindrical tensile specimens were machined to plot the tensile properties of the specimen which proved to be in the range of application. The failure took place after 20000 h at the alternator end between the 15-16 and the 14-15 pistons. It was found that crack initiated on the surface and propagated at 450 to the axis and covered 80% of the cross-section. Benchmarks on the crack surface suggested that the fatigue was of high cycle low stress type. Initiation point can be determined by the elliptical curves of the benchmarks as shown in Fig. 3(a) and (b). Hardness was measured for both the specimen. In case of left specimen (L) hardness was measured from the surface to core and it was found that for nearly 4mm thickness the hardness value was constant and decrease gradually at the core. Microstructure of the part showed matrix formed by ferrite and pearlite which is appreciable. In case of right component (R) the hardness value was very large than the L on the surface and was not constant and abruptly decreased. Also the microstructure displayed the matrix of martensite which is not appreciated and thus giving it a fragile behaviour. It suggested oxidation taken place in this area. In order to analyse crankshaft failure more profoundly, a torsional lumped dynamic model of the system with 12 degrees of freedom (DOF) linked to a finite element (FE) model was developed. The inertia of each DOF was evaluated directly from crankshaft dimensions, and material characteristics and torsional stiffness of each crank were simulated with FE. Later the torsional lumped system model was formulated and solving it produced the instantaneous angular oscillation of each DOF from which torsional load in whole crankshaft were estimated. Finite element model of the crankshaft was developed using Nastran and was meshed with four-node tetraedic solid element. Two different types of load were applied. First is the torsional load derived from the lumped model and the angular displacements calculated from the dynamic lumped model were imposed. Second is the radial and tangential load which produces bending and engine torque respectively. It was also found that maximum of radial and tangential loads never coincide at the same crank angle. After applying all the above loads in FEM for dynamic analysis, the results obtained were close to the stress behaviour under operation. The maximum obtained stress were 40% of the yielding point of the material [3], the most loaded crank being at the alternator end. Thus it was concluded that the fracture was due to fatigue. Crack initiation was depicted from the beach marks. Also microstructure suggested presence of martensite at the region of failure. FEM model depicted number of sights with high stress region and one of them was the sight of fracture. Thus there is probability of failure in other points and hence FEM can be assumed reliable.

As specified in this paper, a number of stress concentration sites were found by FEM which included the one of actual fracture. Thus a detailed analysis of all the stress concentration sites could have helped to justify the reason for crack initiation at only one site. These reasons might have been local friction, scaling, pitting, etc. Ambiguity remains whether the hardness variation of R was manufacturing defect or an operational defect and if the operational defect then the reason for such abrupt variation of hardness is missing.



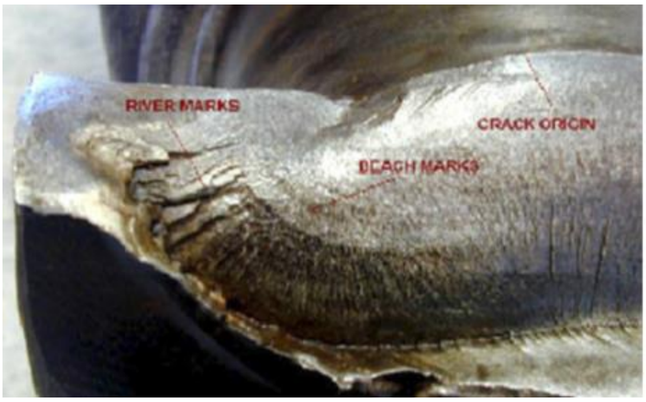


Figure 3: (a) Fracture Surface in the Crankshaft; (b) Fracture Surface in the Crankshaft

Gul Cevik & Rıza Gurbuz (2013), investigated the effect of fillet rolling on the fatigue behaviour of the ductile cast iron crankshaft used in diesel engines. Fillet rolling is a widely used process in automotive industry for improvement of the fatigue life of crankshaft by inducing compressive residual stresses. The induced compressive residual stresses and increased hardness at the deformation zone retards fatigue crack initiation and propagation under cyclic loading. Hence the objective of this paper is to analyse the effect of the local strain hardening and the compressive residual stresses on the fatigue life by constructing the stress versus number of cycles curve and evaluate the endurance limit of the fillet rolled and un-rolled crankshaft. Material of crankshaft used for test is ductile cast iron of grade EN GJS 800-2 which is not heat treated later. Tensile test specimens were prepared from crankshafts and tested by a hydraulic type tensile test machine. Test results were evaluated according to ASTM-E8. True stress strain curve of the crankshaft was also constructed from tensile test data of the crankshaft. Brinell hardness tests were carried out with 2.5 mm diameter Brinell indenter with a force of 187.5 Kgf according to ASTM-E10. Microstructural analysis of the crankshaft was carried out by optical microscopy study on the samples taken from the pin journal of the crankshaft. Since pin fillet regions of the crankshafts are subjected to mainly cyclic bending loads in addition to normal and torsional load, most of the failures occur by bending fatigue mechanism at the fillet regions [4]. By the application of fillet rolling, localized strain hardening occurs and residual stresses develop at the deformed region. Two specimens were taken for testing one with fillet rolled and other without fillet rolled. The undercut radius used is 1.55mm as in Fig. 4. After machining fillet rolling is applied to the undercut region in two stages, shown in Fig. 5. Hardness values were calculated at fillet areas for rolled and unrolled crankshaft resulting in increased hardness due to local plastic deformation and strain hardening.

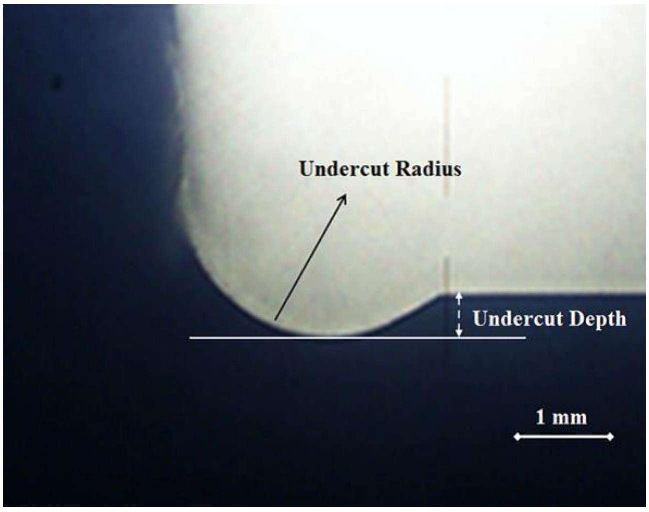
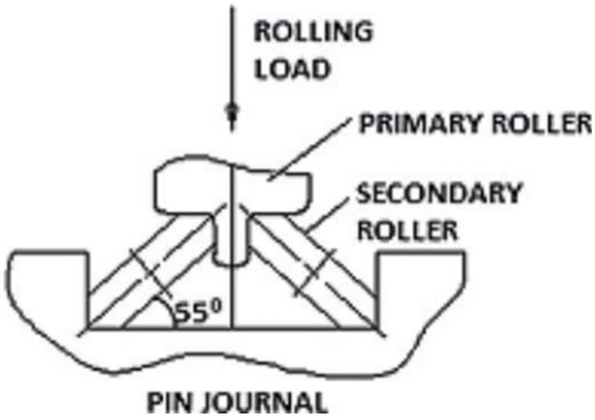


Figure 4: 2D view of the undercut region.

Specimens were cut for fatigue rig testing under cyclic bending on resonant type fatigue test machine. Tests were carried near the resonance frequency (34 Hz) and number of cycles was recorded for the respective bending moments. Tests were conducted according to staircase test methodology to calculate the fatigue endurance limits by Dixon–Mood method. FEM was used to calculate the stresses at the fillet region and as expected the stress values are found to be maximum at the top centre of the circular cross-section of the fillet rolled region. The maximum stress values were used as the corresponding stress levels to the applied test moments. With these values, the S–N curves were constructed. To perform the tests to calculate fatigue life by staircase method, 24 un-rolled specimen and 26 rolled specimens were used. Using these test data along with Dixon-Mood equation it was found that fatigue endurance limit for unrolled and rolled crankshaft are 201 MPa and 811 MPa respectively. Thus it can be derived that fatigue strength of rolled is greater than unrolled by the factor of 4.04. This increase in strength is attributed to compressive residual stresses and local plastic deformation. Also crack is arrested at the compressive residual stress region and there propagation is lowered drastically. The increase in local hardness also makes it difficult for crank initiation on the surface. By liquid penetration test it was found that crack initiated on top centre of fillet region and propagated approximately the same angle through the cross-section of the sample. Thus fillet rolling has appreciable effect on fatigue life of the crankshaft. Test plan according to staircase test methodology has been effectively used and accelerated the fatigue testing of the crankshafts to construct S–N curves.

As the study of this paper mentioned by the author was strictly about the fillet rolling process, still other strain hardening process could be studied. The main effect of this method is strain hardening of the surface at the fillets and other stress concentration areas. Similar strain hardening effect with other processes and its comparison with fatigue life of component should be done to find out the best method.



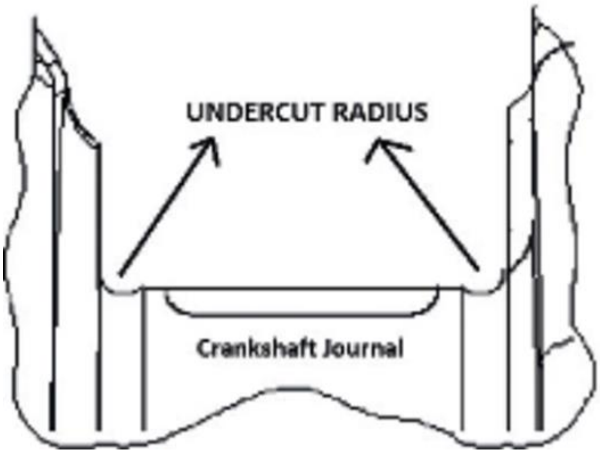


Figure 5: Schematic representations of the crankshaft and rolling operation.

J.A. Becerra et al (2011), carries out the analysis of premature failure of the crankshaft of a reciprocation compressor used in bus climate control system. Various methods were used to analyse as discussed one by one below. The compressor consisted of 4 cylinders in V shape coupled to engine of the bus through V belt and operated under the variable speed of 1000 to 2000 rpm. The shaft was running under harmonic torsion combined with cyclic bending. The work involves determination of von Mises stresses in the shaft through dynamic analysis. This methodology is based on the results of a dynamic lumped model developed jointly with a finite element model (Fig. 6). The crankshaft was made of 34CrMo4 low alloy steel forged as a single piece prior to quenching and tempering. The chemical test carried on three specimens matched the actual composition with little lower percentage of carbon. Brinell hardness test was also satisfactory except for one specimen. The tensile test carried was also odd for the same component out of three. Static and dynamic results were obtained which supports the probability of fatigue failure due to overloading. To calculate gas forces on the shaft thermodynamic model of refrigeration was developed. The friction force was estimated from a model developed by Rezeka and adapted to this compressor [5]. Values were plotted of the pressure, mass and temperature of the fluid in cylinder and static analysis was carried for various speeds of compressor. All scenarios were studied and it was observed that there was overpressure in exhaust process which had significant effect in performance. Insufficient cross section area of the exhaust is the prime cause and its redesign is needed. For compressor torque model the gas pressure and its associated torque should be calculated to get the total torque due to 4 cylinders and this value with friction and inertia torques will give the overall torque. Through the phase diagrams of the cylinders it is deduced that in the 4th harmonic all the cylinders are in phase so it will produce highest torque on crankshaft. For dynamic torsional analysis a torsional lumped dynamic model of the system with 5 DOF linked to finite element was developed. The schematic is shown in Fig. 7. This step gives the relative angular displacement between adjacent DOFs. The results (angular displacements) that involve the crankshaft (DOFs 2, 3 and 4) were subsequently applied in the FE model and in this way a von misses stress contour in the crankshaft could be obtained. The FE model was developed using Nastran. By considering the free vibration system the critical frequencies of the system were evaluated. The first two critical frequencies were found out to intersect the main harmonic speeds and its multiple which were far too less than the lower limit of operating speed. Thus the problem would affect only during starting and stopping of machine. The torsional loads calculated above were used in the FE model along with the radial load and the dynamic analysis executed thus showed close results to the stress behavior during operation. It is found that most loaded region is near the keyway. There are certain possibilities that the maximum stress was overestimated because of neglecting damping effect. Also boundary condition used for static resolution could lead to underestimation of maximum stress value. Thus it is assumed that the main controlling factor for higher stress is torsional loads due to transient torque. This load along with stress concentration factor near the keyway is the reason of failure. As a solution for above study cold working process of shot peening was performed which the manufacturer later adopted and the cracks didn't appear later on.

Estimation of the cause of the problem has been systematically encountered but the data used to evaluate the solution is incomplete. Damping factor is not used in any of the models above which could have significant effect in the values. The life estimation of the product was not evaluated along with the time for crack propagation using methods like LEFM. Solution provided for above problem was the cold working process of shot peening, which is one of the several effective processes. Hence other effective and economic processes could be adopted.

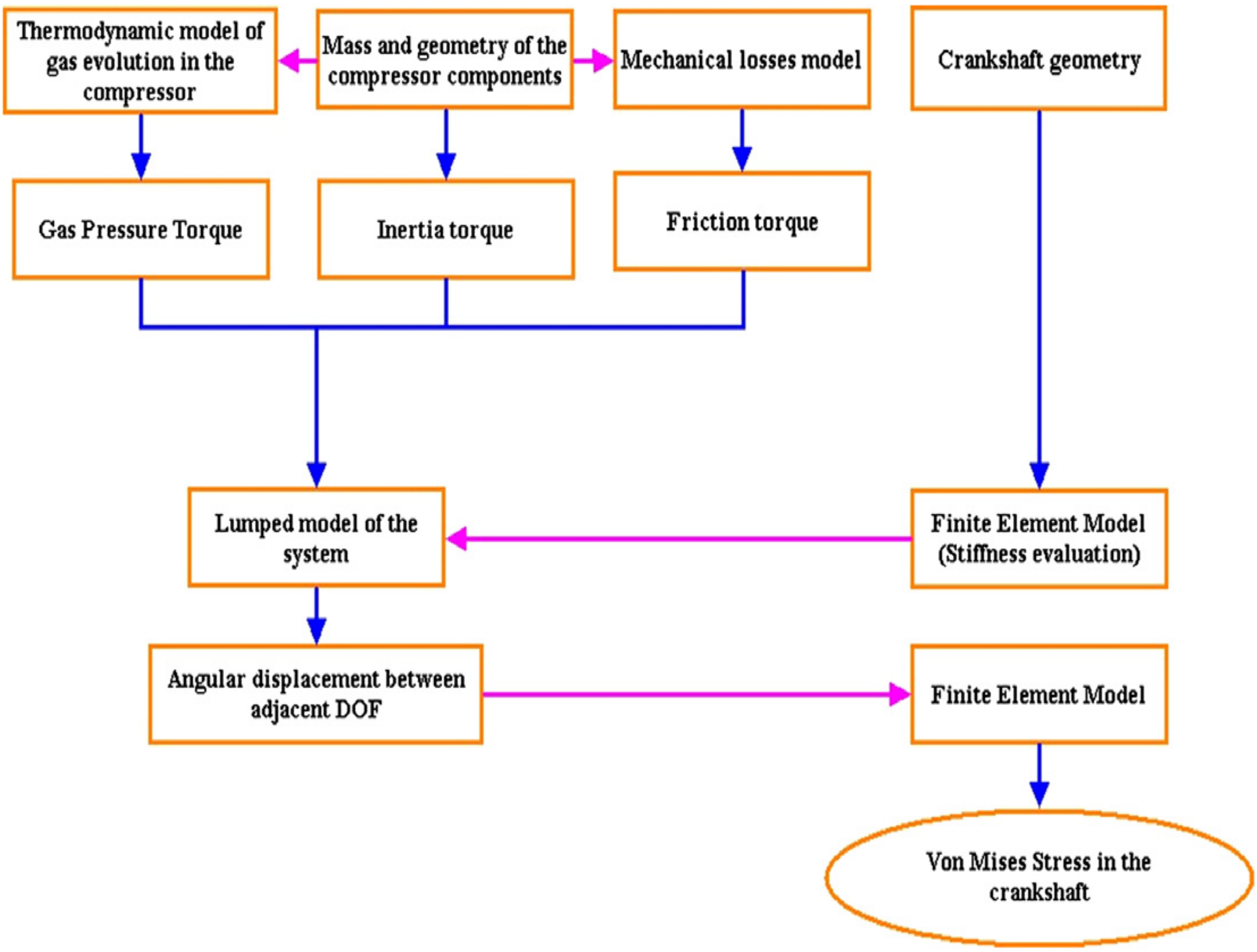


Figure 6: Methodology to evaluate the stress level in the crankshaft due to the dynamic behavior of the system.

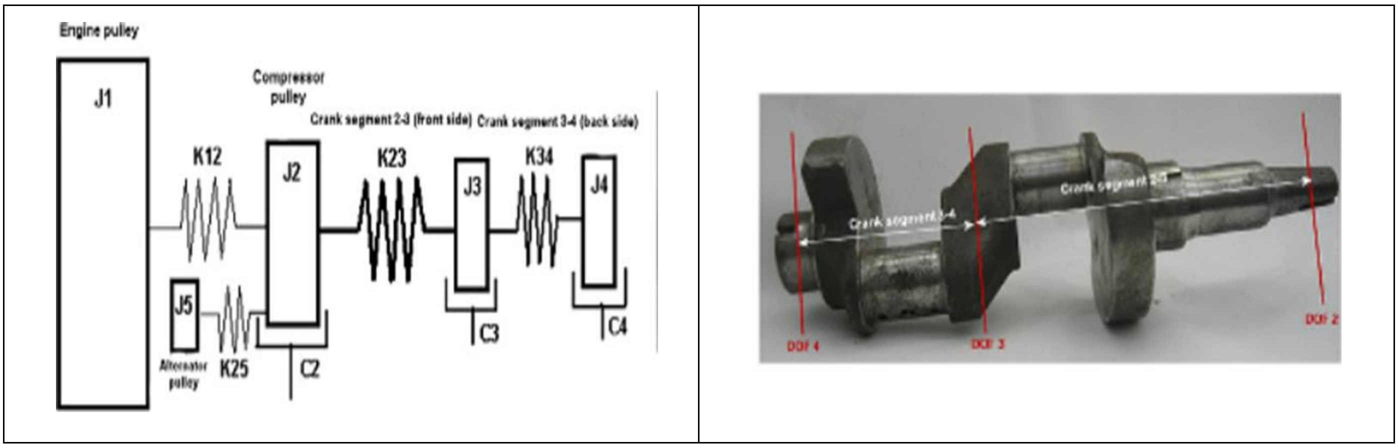


Figure 7: Lumped model of the system.

M. Fonte et al (2013), presents the case study of the crankshaft catastrophic failure of a motor vehicle and its failure analysis. Two different types of loads act on the crankshaft which causes bending and torsion. First is inertial forces caused by the weight of the rotating and sliding components which increases with the speed of the engine. Second is the force due to connecting rod which is caused by the combustion of fuel in the combustion chambers. These two forces are plotted along with the crank angle in Fig. 8. In majority cases the torsional load is neglected as it is 10% of the bending load and neglecting the torsional loads has no significant effect on the von Mises stress at critically stressed locations [6]. Crack starts on the surface due to rotating bending and when there is some misalignment of main journals or unbalance due to counterweight effect. Sometimes there is inclined crack path at 450 due to the change of course of crack by the lubricating holes. This paper aims to study the root cause of the catastrophic failure of the crankshaft. The crankshaft was of diesel motor vehicle of 90HP with a torque of 180 Nm at 2500 rpm. Material of the crankshaft is typical alloy obtained from SAE 1045 cold drawn, forged, normalized. Its nominal tensile strength is about 625 MPa and yield strength 530 MPa. It has a mass of 21 kg and length 560 mm. 3 years after the use the crankshaft failed and seizure was found at crankpin no.2. Repairs and modification were done by non-authorised mechanical shop and the failure took place again after 30000 km. Fig. 9 shows the fractured crankshaft on the crankpin no. 2. Sample of the crankpin no. 2 was cut to obtain the macro and micrographs, and the Vickers hardness from the inner to outside of crankpin surface. This crankpin was rectified to 48.5 mm diameter and then rebuilt by adding a metal alloy layer to the nominal size of 52.5 mm. The heat thermal zone (HTZ) is also observed between the metal base and the added metal alloy. It was found that the microstructure of filled metal alloy was different from the base metal. The micrograph obtained revealed that it was rectified and filled with a metal alloy layer of 2 mm thickness. It also specifies that the crack propagated radially inside from two opposite direction on the circumference and failed at the centre. The fracture morphology shows a clear fatigue failure process under a relative low fatigue crack growth rate, where the typical benchmarks are also visible. The fatigue failure on the crankpin occurred by reversed bending under high stress concentration on the circumferential crankpin web-fillet corresponding to the first 1.5 mm and low bending load which is consistent with the fatigue fracture morphology. The upper and bottom main cracks have different depths as a result of different bending stress levels due to the opening and closing of the webs during the translation motion of the crankpin no. 2. Fig. 10 shows the added metal and its interface with the base metal (HTZ) and some welding defects. The initial circumferential crack path reveals the effect of a small bending load under a high stress concentration on the crankpin-web fillet at the interface of the base metal with the added metal alloy layer. The initial 450 slant path of the crack is due to the filler material. The study finally concludes that the catastrophic failure is due to fatigue and its reason can be anyone of the following: i) inadequate added metal alloy, ii) absence of heat treatment of the repaired crankpin surface, iii) probable misalignment of the crankshaft on journal bearings, iv) imbalance of the crankshaft in consequence of the deep rectification, v) the crankshaft probably was not submitted to a dynamic testing before assembly.

The author has analysed the component only physically. Hence it was not possible to find the exact solution. Chemical test if performed could have given clear idea of microstructure and the chemical composition which could have been the reason of failure. No clear reason of failure is found which its mere disadvantage is. Negligence of torsional load may vary results to a large extent. Analysing analytically and validating it with FEM could have effectively done.

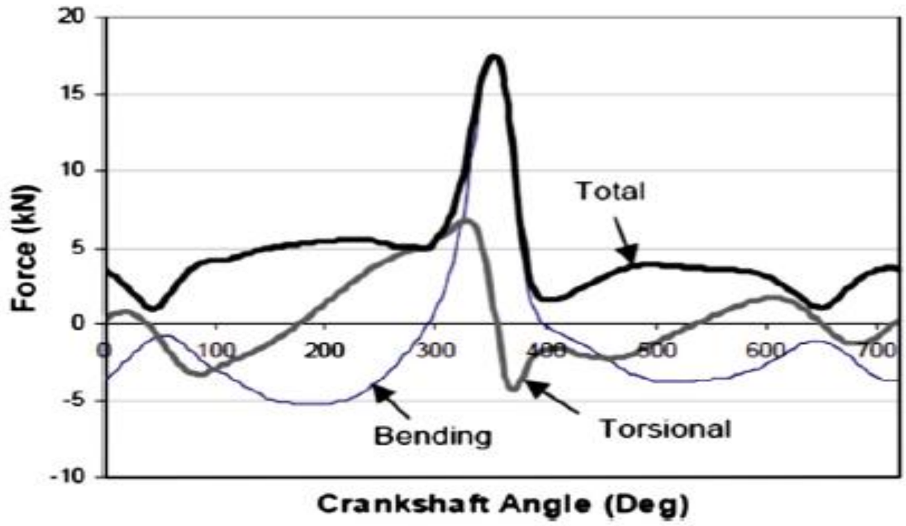


Figure 8: Bending, torsion, and the resultant force at the connecting rod.

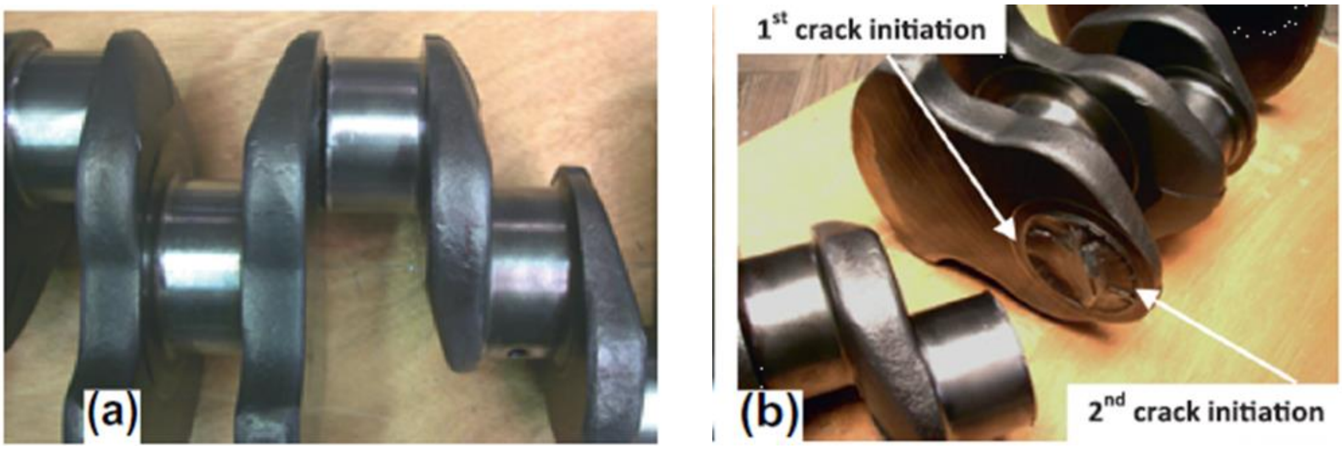


Figure 9: Close-up of the fractured crankpin, (a and b) showing the 1st and 2nd crack initiation.

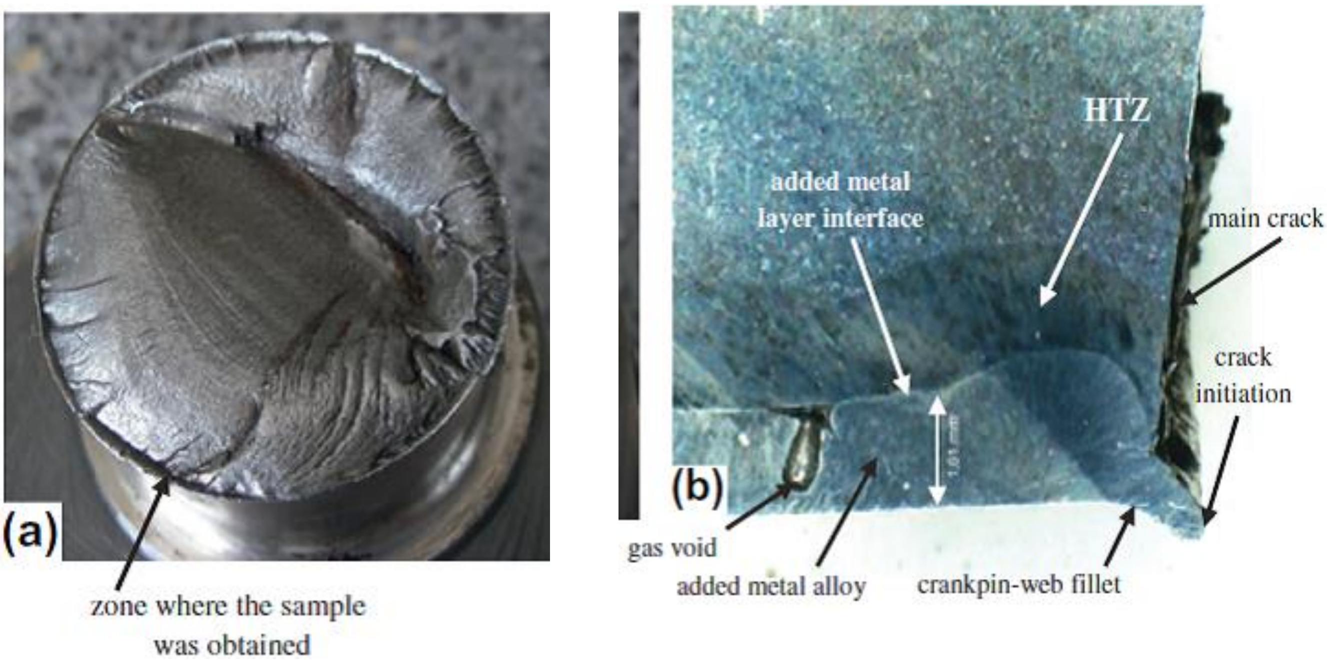


Figure 10: (a) Pin zone where the radial sample was cut, (b) the added metal alloy macrograph.

M. Fonte & M. de Freitas (2009), carries out the case study of a catastrophic failure of a marine crankshaft and the failure analysis to determine the cause of failure. The author tried his best but the reason for the failure was not determined. Two main approaches have been considered, viz. crack nucleation and crack propagation approach. The crankshaft under study is of material 42CrMo4 + Ni + V of an 8 cylinder 4 stroke engine of power 3250 kW at 600 rpm. The broken crankshaft was analyzed and certain facts were found. The journal bearing shells were changed after 29952 h as per plan and the deflection measured before and after the changes were satisfactory. Measurements were also carried out after 31687 h i.e. 1147 h before damage and no problem was found. All the shell bearings were in normal wear condition except of crank pin no.4 which was the sight of catastrophic failure. The beach marks on the fatigue crack surface were studied with the voyage book records [7]. In vibration damper considerable number of springs were damaged which could have been reason of crack generation. Fatigue occurred after 32000 h on the crank web no.4, which was near to the mid of the shaft. By first approach the fatigue seemed to be due to high bending moments on crank web no.4. The fracture section was examined neatly to study the beach marks and it was proposed that crack initiation was due to rotating bending and the propagation was due to combined effect of rotating bending and torsion. The morphology of failure surface suggest that the fracture presents two different surface, one perpendicular to the crankpin section and another horizontal to the crankshaft axis (Fig, 11). Detailed study of the surface suggested that crack initiation (Fig. 12) occurred due to the growth of three parallel cracks which linked together at certain depth. Hardness test was carried out near the fracture area and they satisfied the manufacturers criterion with the material found out to be Bainite. Micro-fractography revealed no pre-cracks, inclusions or other abnormal stress raisers. Also by experience it is genuine that any object running for 32000 h at 600 rpm cannot fail by material abnormalities. The fatigue surface isn't flat surface but an oblique surface. The reason is because, in the beginning, the crack initiation starts from three short parallel cracks nucleated by rotating bending, that were linked, after some millimeters of depth, by the effect of torsion. After this joining, the crack propagation goes on with a typical helical surface due to the effect of torsion. Later fatigue crack growth was studied by assuming an initial crack till the final fracture using the Paris law. This life using LEFM method was constantly compared with their voyage records which proved fruitful as they were compared with the beachmarks and the lines corresponding to each engine stops were analyzed. Thus a value of cycles after crack initiation was calculated to be approximately 2400 h which is less than the last change in journal bearing. Consulting the main engine book on board, it is possible to identify the beachmark zones of the fatigue fracture with the last voyages of the vessel, starting from the fast fracture (catastrophic failure) towards the crack initiation point. It was estimated that 80-90% life was spent for crack nucleation. Thus the study hitherto concludes that the fatigue was caused by rotating bending with steady torsion. The damaged vibration dampers would have caused torsional failure but the surface study emphasized majorly on high rotational bending.

The author was not able to determine the actual reason for the fatigue failure, which the biggest shortcoming of this paper is. The life up-to crack nucleation should have been compared with FEM analysis for validation. The failure of vibration dampers would have helped to find cause of fatigue. The analysis of engine speed variation along with natural frequency could have helped for checking of resonance and the damage to vibration dampers.

a


b


Figure 11: Failure surface. (a) Parallel to the axis (final fracture) and (b) perpendicular to the axis (fatigue crack growth).

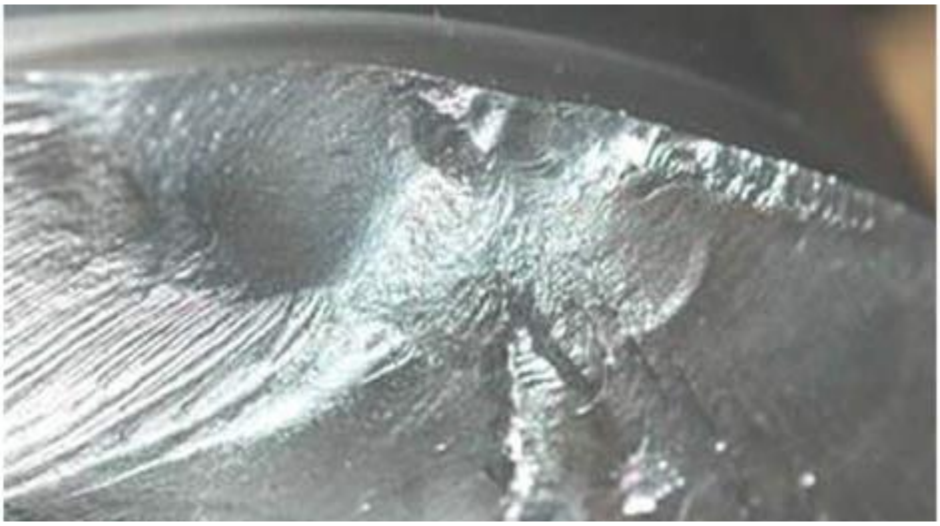
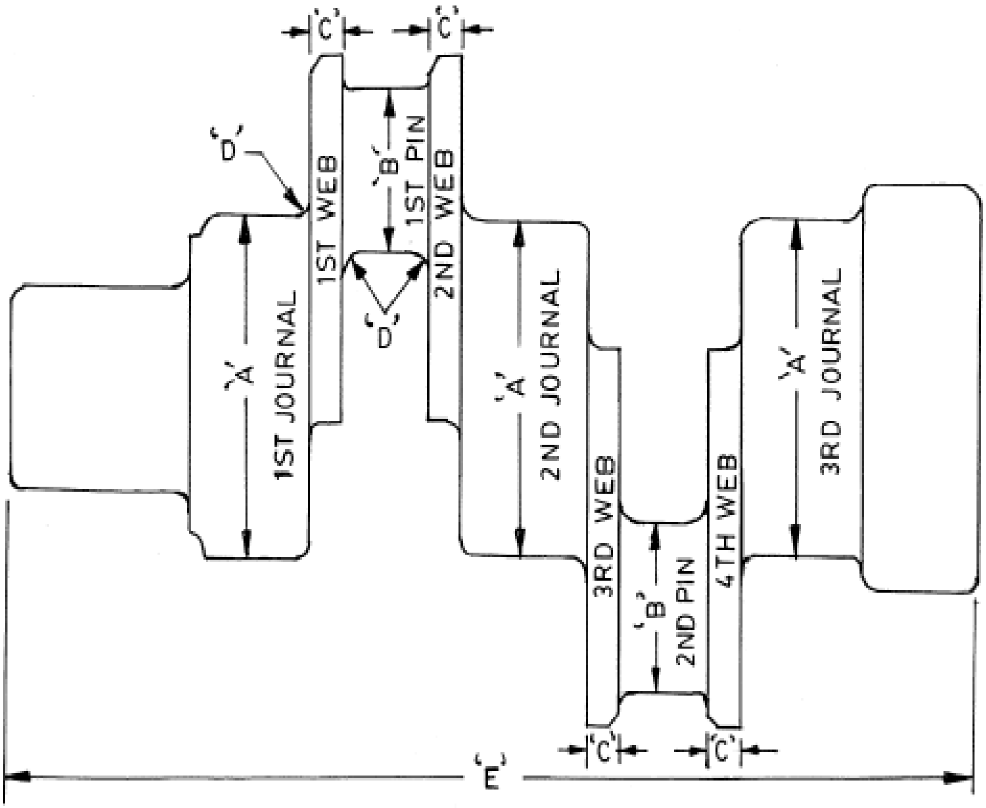


Figure 12: Crack initiation zone. Magnification of the crack initiation site

R.K. Pandey (2003), investigates the failure of crackshafts of a diesel engine used in tractors made from 0.45% carbon steel. Failure occurred at the web section. Attempts were made to estimate the stress level required for fatigue initiation from the crankpin-web fillet region. The failure time was estimated and this was correlated with the observed failure times in different crankshafts. Recurrence of the premature failue of the crankshafts of 2 cylinders 35 HP were reported. The failure time were between 30 h to 700 h. the material was C45 steel and was normalized to give hardness before machining. Induction hardening was done at pins and journals and were ground before assembling. Fabrication process is described in Fig. 13 below. In each crankshaft (Fig 14), failure took place in web region. Chemical analysis done on the failed section of the shaft stated that the composition was in expected range. The microstructure as per composition and heat treatment appears to be normal. Specimens were created from the fractured material for tensile and Charpy test. Tensile test proved the values to be in expected range. Hardness also was in specified range. Maximum charpy energy was calculated which was later used to calculate the fracture toughness and the minimum fracture toughness was evaluated. Visual inspections depicts the fracture due to fatigue. From the beachmarks it was found that the crack initiated from the filet region between pin and web and it progressed throughout the cross-section. The overall fatigue surface has about a 450 inclination with respect to the shaft axis. Attempt was made to estimate the fatigue initiation stress at the pin web fillet. The fractographic studies were used to estimate the stress level present during fatigue crack propagation. Fatigue striations were examined under Scanning Electron Microscope (SEM) and the distance between the striations were measured to determine the approximate stress level during the fatigue propagation stage. Fracture crack growth rate (FCGR) was calculated. Once a fatigue crack has nucleated at the pin fillet, the stress required for its further propagation was estimated through a calculation employing fatigue crack propagation threshold (Kth) data for the steel. Assuming number of cycles required for fatigue initiation to be in the range of 107 and considering a loading frequency of 90, 000 cycles/h, time for fatigue initiation is found to be about 110 h. The fracture surfaces of crankshafts Nos. 1, 2, 3 and 5 were examined under the SEM at appropriate magnifications to study the mechanism of failure initiation and propagation. In first crankshaft multiple nucleation was found near the pin web fillet. Failure of this shaft could have been caused due to relatively high stress during the fatigue cycle at the initiation stage. In second crankshaft a number of long and deep inclusions and cracks parallel to the interface were noticed in the pin fillet where fatigue had apparently nucleated. Presence of Al, Si, Pb, P and S within the inclusions was revealed by EDAX test [8]. The fatigue appears to have initiated through the above surface defects which were likely to be the source of fatigue cracking. In the third crankshaft, multiple nucleation of fatigue at the pin-web fillet was evident from the nature of beach patterns in the initiation zone. In another location, a surface discontinuity was also noticed. In the fatigue initiation zone of fifth crankshaft, large scales inter granular cracking and a network of grain boundary cracks were noticed. In addition, some cracks were noticed at the surface. EDAX examination revealed the presence of Al, Si, Ca and Mg in addition to P and S along the grain boundaries and in the inclusions. Thus the failure of fifth shaft is different than the others.

The author here has systematically analysed the crankshaft of diesel engine but the solution obtained were based on various assumptions in the calculation of crack initiation and propagation rather than evaluating the actual values. Validation of these results through FEM would have proved better. Other approaches like LEFM and CDA could have backed up the results deduced by FCGR.



Dimensions after machining (mm)	'A' Journal dia	'B' Pin dia	'C' Web thickness	'D' Radius	'E' Total length
	80	75	22	4	435

Figure 13: Typical diagram of the crankshaft.

Bloom
↓
Forging to shape
↓
Trimming
↓
Normalizing
↓
Machining
↓
Induction hardening
↓
Stress relieving annealing (180 °C, 2 h)
↓
Grinding

Figure 14: Steps in fabrication of crankshaft

Rajesh M. Metkar et al (2013), carries out the comparison of two different methods of fatigue life estimation of a single cylinder diesel engine crankshaft by fracture mechanism. The two methods are linear elastic fracture mechanism (LEFM) and recently developed critical distance approach (CDA). LEFM is an analytical method. It uses the stress intensity factor which characterizes the stress distribution in the vicinity of crack tip. In CDA a group of methods predicts failure using stress distance plot. Finite element analysis based ABAQUS is used to calculate maximum stress value for both methods. The test material is a forged steel crankshaft of weight 4.8 kg designed for 480cc single cylinder diesel engine having power 9 kW. Fig.15 shows the model of crankshaft with critical locations. Crack length of various sizes has been assumed from 0.5 to 8mm with an interval of 0.5 mm. In case of LEFM fatigue life is predicted assuming crack already pre-exist in the component and life is directly dependent in stress intensity factor [9]. The fatigue load and crack length is defined in terms of stress intensity factor. The linear relation between crack growth and threshold stress intensity is represented by Paris law. Thus fatigue life of a component can be calculated and hence we get the number of cycles before failure. In CDA parameters taken in front of notch or stress concentration area is used. Here PM (Point Method) is used for its simplicity and effective prediction of failure. The PM predicts that failure will occur if the stress at a distance L/2 (L is critical distance) from the notch root is equal to the plain strength of the material. The same principle will be applied for fatigue, replacing the stress with a stress range and the plain strength with the plain fatigue strength. If the critical distance and accurate estimation of stress is known, point method is best to determine failure. The equation used in CDA allows the critical distance to be expressed as function of fracture toughness and also. The PM calculates the stress values and equates to the characteristic strength of material to consider the propagation of crack of finite size. For analysis, boundary condition is used as shown in Fig. 16. The distribution of load over the connecting rod bearing is uniform pressure on 1200 of contact area. The dynamic boundary condition and load is determined by the pressure and crank angle diagram. Stress analysis is performed in FEM commercial software ABAQUS. Global size of 5mm was considered for meshing of components. Results were obtained from ABAQUS indicating maximum stress to appear at 4b. Hence crack was created at 4b ranging from 0.5 to 8mm (Fig. 17) for LEFM and CDA method calculations. Stress intensity factor for each length were calculated and LEFM values were tabulated which indicated component failure when crack propagates from 1mm to 1.5mm. The CDA formulated values were also tabulated and it suggested that the component fails when transition of crack occurs from 5mm to 6mm. This concludes that critical locations are at fillets. Analytical and FEA do not show close agreements due to the complex geometry of component which is approximated in analytical which basically suggests the need to use FEA softwares. It also states that LEFM under predicts the life of the component compared to CDA method. This paper also gives insight of the two methods and the benefit to the design engineer to correctly asses the life of crankshaft in the early stage.

As the scope of this paper is limited to only two methods CDA and LEFM by the author, still consideration of other method like FCGR could have led to better comparison. The main drawback of this paper is that the two methods used for analysis vary in a large scale. There was negligence of the torsional load. The torsional lumped model could be used to for accurate solutions. The analysis does not specify which method is suitable to adopt for a particular problem statements. None of the two methods specifies the cause of fracture.

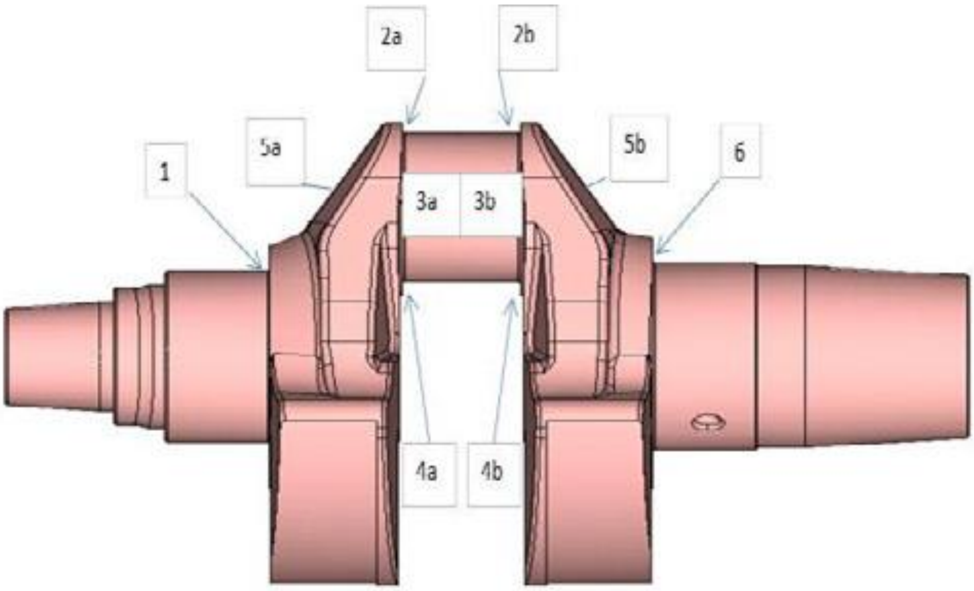


Figure 15: Crankshaft model with critical locations

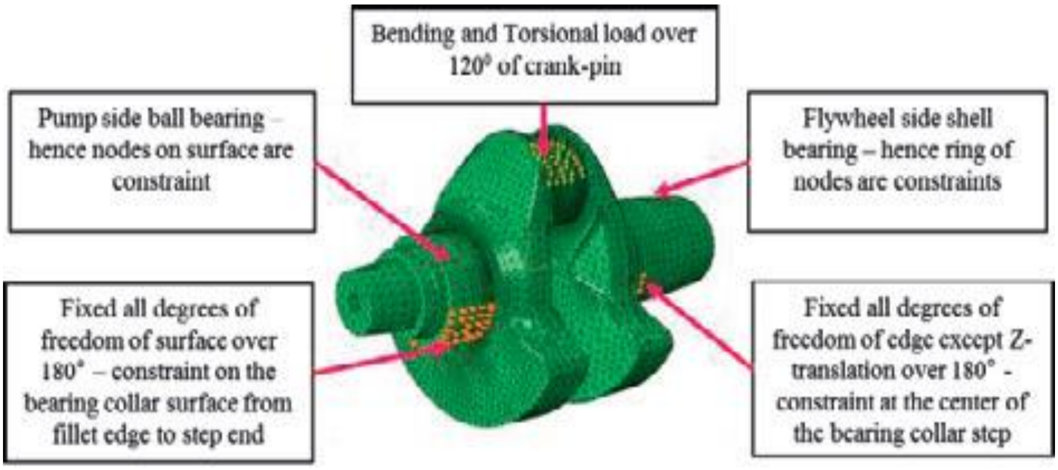


Figure 16: Crankshaft boundary condition and loading

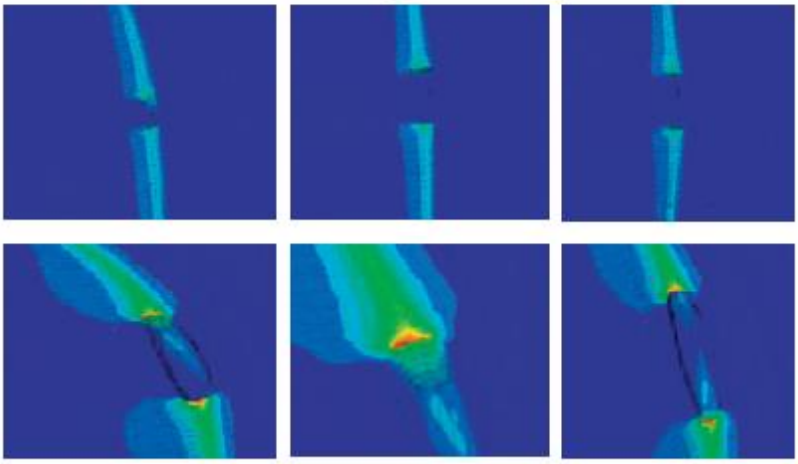


Figure 17: Stress around the crack for 0.5 to 8mm

S.K. Bhaumik et al (2002), studied the crankshaft of a piston engine of a transport aircraft that failed during climb. Cracks were found to initiate at the web radius region of journals 2 and 3. Crack was initiated due to surface contact fatigue caused by constant rubbing of the bearings against the journal web. Movement of bearing was due to axial load. This paper has undertaken the task to analyse failure. The aircraft was withdrawn from the service due to the noise from the engine. It ran 1460 h in service and 262 h since last overhaul. No defects were found during overhaul. It was a complex shaped crankshaft machined from an alloy steel forging of SAE 4340 grade and case hardened. Visual and stereo-binocular observations found cracks at journal 2 and 3. Careful investigation revealed that failure occurred transverse to the axis of the shaft. The crack in journal no. 30 had propagated 80% with distinctive beachmarks whereas in case of no. 2 the beachmarks were not distinctive. Both the cracks were initiated at the web radius. Careful observations found pitting and spalling at the web radius close to crack origin. This suggested frictional fatigue. So the damaged crankshaft was removed to examine and found that the split ring had gone through extensive wear. Later it was found that tenon of this particular bearing had considerably suffered some metal loss. The component was then examined under scanning electron microscope. The beachmarks clearly suggested that the failure was due to fatigue. On higher magnification they found some traces of spalling near the crack initiation region. In addition to pitting, scaling and surface cracking, scoring marks along the circumference of the web can also be seen. A suitable sample piece was cut from the crankshaft close to the crack origin region of journal 3, metallographically prepared and observed in an optical microscope in both unetched and etched conditions. In the unetched condition, isolated globular type inclusions were seen. The inclusion rating was estimated to be ASTM designation E 45, Plate 1, D, Globular Type oxides, 1, Thin series [10]. This was found to be within the specification of the crankshaft material. After etching with alcoholic ferric chloride solution the material showed a tempered martensitic structure. No abnormality was observed in the microstructure. A case hardened surface layer was also observed macroscopically. Semi quantitative analysis carried out by EDX attached to the SEM showed that the crankshaft was made of SAE 4340 steel. The surface of the crankshaft was hardened by nitriding. The split ring bearing was found to be made of steel with 0.4% Mn. The core and surface hardness were measured to be HV 310 and HV 550, respectively. In the web radius region, the hardened surface layer was lost due to pitting and spalling, and the hardness of the exposed surface was found to be about HV 440. The hardness of the split ring bearing material was measured to be HV 175. Finally all the analysis concluded that the failure occurred was merely due to friction fatigue. The crack initiated at web radius region. There were no defects in the material of component. Fracture at journal no.3 was of High cycle low stress type depicted from its beachmarks where as in no.2 it was due to overloading. Pitting, scaling was found at web radius due to friction with the split ring due to axial loading. But no misalignment of shaft was found. Analysis found out the tenon was degraded and worn out. Also there was drastic difference in the hardness of the split ring and the web surface leading to the subsurface crack caused by constant friction.

Author here has limited his scope to physical and chemical analysis which could have analyzed effectively with the help of numerical analysis. This numerical analysis could have proved with the FEM analysis. Using FEM analysis a number of high stress concentration areas could be determined which could be one of the site of crack initiation. It could also be proved that this stress concentration was the cause of crack initiation.

III. CONCLUSION

All the cases studied in above paper were all related to fatigue failure. It is found that the crack is initiated at high stress concentration area which is mainly fillet between crankpin and web in many cases. Various methods from static analysis to dynamic analysis, crack initiation evaluation, crack propagation evaluation were studied. Failure of crankshaft is inevitable and hence maintenance and periodic checking of the components should be done. Designing of the crankshaft is necessary in the view that less concentration of stresses lies at the fillet section. Thus it proves that there is no single method to solve a problem and there is not a single method that can solve all the failure problems. Table 1 show summary of the Crankshaft Analysis Papers and Table 2 shows evaluation of models used in referred papers.

TABLE I. SUMMARY OF THE CRANKSHAFT ANALYSIS PAPERS

Paper No./ Type	Key Approaches and Deductions	Select Weaknesses/ Limitations
P1. Fatigue fracture expertise of train engine crankshafts	Three crankshafts were studied. Failure occurred was due to fatigue in each case. Crack initiation occurred due to mechanical as well as thermal fatigue loads whereas crack propagation was merely by mechanical loads of cyclic bending and torsion.	Shafts were analysed by visual inspection. No validation was done analytically. FE model was not used for critical stresses calculation. The life was not estimated for crack nucleation and crack propagation.
P2. Analysis of an unusual crankshaft failure	Four cracks were found on the edge of the oil hole. Using mechanical analysis, microstructure and metallurgy the reason of this event has been revealed. Force of friction caused by improper crankshaft repair and assembling is main factor of the failure. Why friction occurs, how the crack initiates and expands and what the process of failure is were studied.	FEM model was not used generated to find the critical stress areas on the crankshaft. Crack nucleation and crack propagation period was not studied.
P3. Analysis of a diesel generator crankshaft failure	The fracture occurred in the web between the 2nd journal and the 2nd crankpin. Fractographic studies show that fatigue is the dominant mechanism of crankshaft failure, where the beach marks can be clearly identified. A thin and very hard zone was discovered in the template surface close to the fracture initiation point, which suggests that this was the origin of the fatigue fracture.	Only single reason of presence of martensite is specified for fatigue failure. Causes of the crack initiation like oxidation or wear is not mentioned. Solution to avoid fatigue failure is also missing.
P4. Evaluation of fatigue performance of a fillet rolled diesel engine crankshaft	Based on the staircase test methodology, fatigue tests were conducted to evaluate the fatigue endurance limits of ductile iron crankshaft under fillet rolled and un-rolled conditions. The corresponding stress levels were calculated by the finite element modeling. Test data was analysed by Dixon–Mood method to calculate the endurance limits. Fatigue endurance limits measured for un-rolled and rolled crankshafts are 201 MPa and 811 MPa respectively, emphasizing a significant fatigue endurance limit improvement by fillet rolling process.	Analysis involves study of one method of cold rolling. Study of other methods along with its comparison is lacking.
P5. Failure analysis of reciprocating compressor crankshafts	The simulation included several sub-models, viz. Thermodynamic model, Compressor torque dynamical model, Finite element model (FEM), Dynamic lumped system model. Results from the lumped model were incorporated into the FEM in order to evaluate the stresses due to the torsional dynamic in the crankshaft.	Damping was not involved in the FE model. The life estimation of the product was not evaluated along with the time for crack propagation using methods like LEFM.
P6. Crankshaft failure analysis of a motor vehicle	A transversal macrograph of the crankpin revealed that the crankpin was rectified and filled with a metal alloy for the same nominal diameter. Two fatigue cracks growing to the center of the crankpin where the final fracture occurred. The	Chemical test if performed could have given clear idea of microstructure and the chemical composition if might be the reason of failure. No clear reason of failure

	symmetric semi-elliptical crack front profile confirms the effect of a pure mode I under alternating bending. The catastrophic failure was a consequence of i) inadequate added metal alloy, ii) absence of heat treatment of the repaired crankpin surface, iii) probable misalignment of the crankshaft on journal bearings, iv) imbalance of the crankshaft in consequence of the deep rectification, v) the crankshaft probably was not submitted to a dynamic testing before assembly	is found which its mere disadvantage is. Negligence of torsional load may vary results to a large extent. No analytical solution is provided with FEM model.
P7. Marine main engine crankshaft failure analysis: A case study	The cycles calculated by the linear elastic fracture mechanics approaches showed that the propagation was fast which means that the level of bending stress was relatively high when compared with total cycles of main engine in service. Microstructure defects or inclusion were not observed which can conclude that the failure was probably originated by an external cause and not due to an intrinsic latent defect.	Some causes are analysed and reported here but the origin of the fatigue fracture is not clearly determined. Failure of vibration dampers was not analysed.
P8. Failure of diesel-engine crankshafts	The investigation included determination of chemical composition, microstructural examination, evaluation of tensile properties and charpy toughness as well as hardness determination. The fracture toughness was estimated from the charpy energy data. The failure zones in various crankshafts were examined using the SEM. Fractographic methods were used to estimate the stress required for fatigue propagation. The failure time was estimated and this was correlated with the observed failure times in different crankshafts. The studies indicated that fatigue initiation from the crankpin-web fillet region necessitated a stress level of about 175 MPa	Various asumptions were made in crack initiation and propagation calculations. Validation of these results through FEA is not done. Numerical calculations should have been backed up with FE analysis. Other approaches Like LEFM not used.
P9. Evaluation of FEM based fracture mechanics technique to estimate life of an automotive forged steel crankshaft of a single cylinder diesel engine	Comparative studies of two methods of fatigue life assessment of a single cylinder diesel engine crankshaft by using fracture mechanics approach viz. linear elastic fracture mechanics (LEFM) and recently developed critical distance approach (CDA) is done. Detailed overview of failure analysis process including theoretical methods and result integration for predicting life of components as compared to life estimation by means of software is also given	Two methods used for analysis used vary in a large scale. Torsional loads are neglected in calculations. The torsional lumped model not used. Cause of failure is not determined.
P10. Fatigue fracture of crankshaft of an aircraft engine	Cracks were found to initiate at the web radius region of journals 2 and 3 and progressed in the transverse direction of the shaft axis causing it to fracture. The initiation of cracks was by surface contact fatigue due to constant rubbing of bearings against the journal webs. The movement of the bearing was caused by axial load on the crankshaft	Validation is not done using FEM or any analytical methods.

TABLE II. EVALUATION OF MODELS USED IN PAPERS

Sr. No.	Methods and Models	Papers									
		P1	P2	P3	P4	P5	P6	P7	P8	P9	P10
1.	Visual Inspection	•	•	•	•	•	•	•	•	•	•
2.	Microscopic Inspection	•	•	•	•		•	•	•		•
3.	Fractography		•					•	•		
4.	Chemical Test	•	•	•		•			•		
5.	Hardness Test	•	•	•	•	•	•	•			•
6.	Tensile Test	•		•	•	•			•		
7.	Charpy Test	•							•		
8.	NDT	•			•						
9.	LEFM							•		•	
10.	CDA									•	
11.	FEM			•	•	•				•	
12.	Thermodynamic model					•					
13.	Dynamic lumped system			•		•					
14.	EDAX								•		•
15.	Dixom Mood Method				•						
16.	Staircase Test method				•						

FUNDING STATEMENT OR DECLARATION OF CONFLICTING INTERESTS

This research received no specific grant from any funding agency in the public, commercial, or not-for-profit sectors.

REFERENCES

[1] A. Ktari, N. Haddar, H.F. Ayedi. Fatigue fracture expertise of train engine crankshafts. Engineering Failure Analysis 18 (2011) 1085–1093.

[2] Changli Wang, Chengjie Zhao, Deping Wang. Analysis of an unusual crankshaft failure. Engineering Failure Analysis 12 (2005) 465–473.

[3] F. Jiménez Espadafor, J. Becerra Villanueva, M. Torres García. Analysis of a diesel generator crankshaft failure. Engineering Failure Analysis 16 (2009) 2333–2341.

[4] Gul Cevik, Rıza Gurbuz. Evaluation of fatigue performance of a fillet rolled diesel engine crankshaft. Engineering Failure Analysis 27 (2013) 250–261.

[5] J. A. Becerra. F. J. Jemenez, M. Torres, D. T. Sanchez, E. Carvajal. Failure analysis of receprocating compressor crankshafts. Engineering Failure Analysis 18 (2011) 735-746.

[6] M. Fonte, Bin Li, L. Reis, M. Freitas. Crankshaf failure analysis of a motor vehicle. Engineering Failure Analysis 35 (20130 147-152.

[7] M. Fonte, M. de Freitas. Marine main engine crankshaft failure analysis: A case study. Engineering Failure Analysis 16 (2009) 1940–1947.

[8] R.K. Pandey, Failure of diesel-engine crankshafts. Engineering Failure Analysism 10

(2003) 165–175.

[9] Rajesh M.Metkar, Vivek K. Sunnapwar, Subhash Deo Hiwase, Vidya Sagar Anki, Mahendra Dumpa. Evaluation of FEM based fracture mechanics technique to estimate life of an automotive forged steel crankshaft of a single cylinder diesel engine. Procedia Engineering 51 (2013) 567 – 572.

[10] S.K. Bhaumik, R. Rangaraju, M.A. Venkataswamy, T.A. Bhaskaran, M.A. Parameswara, Fatigue fracture of crankshaft of an aircraft engine. Engineering Failure Analysis 9 (2002) 255–263.