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FAILURE ANALYSIS OF MECHANICAL SHAFT

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ABSTRACT
Cases of component failure from previous researchers encourage the analysis of the causes of shaft failure according to the operational conditions of production machining as an urgent matter to find the problem. The purpose of the analysis is to obtain the causes of shaft failure in production facility applications. The analysis method includes tracing the results of previous shaft research, studying the methods of testing mechanical properties, composition, hardness, simulation, stress concentration, torsional stress, bending stress, defects, cracks, scratches, inclusions, voids, corrosion, wear, heat treatment, comparison between the results, discussion and conclusions. Implementation of shaft analysis in machining, production facility construction, and research/test equipment. The conclusions obtained include: the crankshaft of the wheel loader diesel engine after operating for 4800 hours, broke at the fifth crankshaft with diameter 82 mm, the actual tensile strength of the shaft 832.3 MPa is still < DIN 1.7225 standard (42CrMo4) worth 900-1100 MPa; diameter 55.5 mm condensate pump shaft of SS 416 was broken, due to torsional fatigue from the sharp edge of the outer surface of the shaft where MnS inclusions were as a design error and MnS inclusions near the shaft surface as a metallurgical error; and so on.

Keywords : *composition, fatigue fracture, hardness,mechanical properties, shaft.*

1. Introduction

Failure of many machine shafts can be caused by various reasons and circumstances. Failure of a component can be detrimental and dangerous to a production system, therefore efforts to anticipate failures from occurring or being eliminated are very important steps for the success of a production. Analysis of a failure on a shaft is carried out on several previous research results to obtain information on the cause of failure, location of failure, and type of material that fails in the operation of a production.

Previous research results that focus on failure analysis on various shafts were obtained from the following groups: centrifugal pumps from 14 studies, diesel engines 3 studies, and one study each for cases on double screw pumps, shipping pumps, submersible pumps, condensate pumps, wheel drive cars, booster pumps, recycle pumps, drive shaft motors, vertical water pumps, gear pumps, ship-propellers, turbochargers, speed reducers, and stepped shafts.

Previous research results that focused on the causes of failure in various shafts were obtained due to the following reasons: material strength in 9 studies, stress concentration in 12 studies, corrosion in 2 studies, scratches and the alloy content in 2 studies, and one study each with reasons due to viscosity, friction, fatigue life, foreign objects, inclusion, and loosely tightened of nut.

2. Literature Review

The 107 JA centrifugal pump shaft for ammonia failed due to torsional stress exceeding the fatigue limit value of 17-4 PH stainless steel of 609.79 MPa in the middle near the keyhole, and 17-4 PH stainless steel is able to withstand stress of 655 MPa, so that the stress occurs at a value approaching the fatigue limit value of the material [1]. The stress that occurs on the 107JA shaft is close to (almost the same as) 6.9% with the stress of the SS17-4 PH material of 655 MPa. Initial cracks occurred in the centrifugal ore slurry pump shaft made of AISI 4140 material at the bottom of the thread through liquid penetrant testing and poor machining results caused stress concentrations (SC) that contributed to the formation of cores and fatigue crack growth that ended in shaft fracture [2]. Machine work should be carried out properly to minimize the chance of initial cracks within a certain radius. The shaft of the double screw pump that balances the axial force by sucking the medium at both ends and sending in the middle of the martensitic stainless steel ($05Cr_{17}Ni_4Cu_4Nb$) material is broken, due to the increase of the rotating and bending load caused by the low viscosity whose crack starts from the keyway and propagates to the center of the shaft, and finally breaks.

The specification of the crude oil pump manufacturer with a viscosity of 380 mm^2/s at a rotational speed of 1450 rpm, but the actual viscosity is very low at 60.4 mm^2/s, so that the positive pump pressure increases which causes the leakage to increase, and the pressure limit drops to below 0.8 MPa which should be 1.4 MPa, resulting in the pump flow decreasing from 190 m^3/h to 140 m^3/h [3]. It is recommended that the pumping be maintained for a constant crude oil viscosity of 380 mm^2/s, if there is a change, the motor speed should be adjusted first before operating or the pump design should be modified. High vibrations in the SS 410 multistage ammonia centrifugal pump cause friction between the ring and shaft to increase the temperature to around 1227-2427°C which exceeds the pump's working temperature of 40-60°C, causing expansion of the material of both components, so that the ring locks, until the shaft finally breaks at the point of friction with the ring [4]. Preventive advice by increasing the axle-ring clearance by 0.40 mm without changing the type of material, namely SS 304, if replaced with SS 410 the minimum clearance is 0.35 mm. The fracture of the AISI 1020 shaft in the 56GA4002A type centrifugal pump was due to having operated for 15,019,200 cycles which exceeded the fatigue threshold at 10,000,000 cycles, and the hardness at the non-fracture point was 67.06 HR_C, and the highest hardness value at the fracture point was 71.92 HR_C [5].

The excess operational fatigue life is 50.1% at 15,019,200 cycles and the highest hardness achieved is > 7.24% at the fracture point of 71.92 HR_C. The Hydraulic Power Unit (HPU) shaft made of AISI 4140 steel broke due to torsional stress at the base near the keyway, because an initial crack occurred that spread across the axis of the shaft with a beachmark as a fatigue fracture and from 9 hardness test points all the values were still within the normal range of 254-361 HV where the fracture surface was relatively flat and there was no plastic deformation [6]. If there is no plastic fracture, it means fatigue fracture due to brittleness. Failure of AISI 316 shaft started with the presence of initial crack at the keyway end, slow crack propagation and immediate fracture zone, because the working stress of 52.2 MPa was greater than the shaft torsional shear stress of 27.95 MPa [7]. The working stress is 86.76% higher at 52.2 MPa than the torsional shear stress of the shaft. The pump shaft made of Monel K-500 nickel-based alloy for electrical submersible pump (ESP) which broke at a depth of 1551.5 m from the ground surface.

The fracture occurred in an area where there was no plastic deformation on the surface which started from a micro crack/cavity in the area of origin of gouging (corrosion due to welding on a metal surface with holes/grooves). The overload due to the torque load due to the bearing seat was suddenly stopped until the impact load on the shaft occurred, because the motor condition was still working to rotate the shaft, so that torque force was generated [8]. In the weld area, the occurrence of grooved surface corrosion must be checked and avoided so that it does not become an initial opportunity for cracks. A 55.5 mm in diameter of condensate pump shaft made of martensitic SS 416 fractured due to torsional fatigue initiated from the sharp edge of the outer surface of the shaft where MnS inclusions were due to SC of sharp corner cracks (design error) tested with die-penetrant and MnS inclusions near the shaft surface (metallurgical error). The sharp corners are suggested to be smoothed and MnS removed from the alloy by pre-hardening at 1020°C for 1 hour to dissolve and redistribute MnS followed by oil quenching and tempering at 590-650°C to improve mechanical properties [9]. The cause of fracture of SS 416 shaft is due to MnS inclusion and sharp corner crack due to metallurgical design error. The fracture of the APP-4 stainless steel centrifugal pump shaft of the IP251-U153 centrifugal pump was due to scratches on the shaft surface and the Mo content was lower at 1.41% from the standard of 2.53% at a distance of 120 mm from the 60 mm in diameter shaft [10]. Installation of the impeller in the pump room must ensure that there are no foreign objects such as bolts, sand or other materials.

The SS 304 shaft of the submersible vertical centrifugal pump failed due to the presence of foreign objects in the form of sand and a bolt in the pump chamber which received a rotating load causing tensile stress that exceeded its tensile strength which occurred in the cross-sectional reduction area next to the rolling bearing seat [11]. The elongation of the speed reduction shaft material is 16.67% lower than the minimum value of 21%. The shaft of a 215 mm in diameter single-stage double-suction centrifugal pump with a discharge of 10,000 m^3/h, a head of 62 m, 2100 kW, 590 rpm, made of 06Cr17Ni12Mo2Ti austenitic forged steel with an elongation of 17.5%, less than the specification range of 21-29%, fractured due to initial cracking by MnS and Al_2O_3 inclusions that reduced the ductility of the alloy below the keyway [12]. The stress increase that occurs when the nut is not tightened is 4.75% higher at 319.2 MPa. The right front drive shaft of the car has an initial stress of 219 MPa when the shaft is not shifted, and when it is shifted 6 mm towards the non-hardening area, the stress increases to 319.2 MPa, due to the loosely tightened shaft nut retreating, so that the shaft breaks at the nut thread [13]. The stress increase that occurs when the nut is not tightened is 4.75% higher at 319.2 MPa. The feed water booster pump shaft was broken, because there were 3 initial crack points on the shaft surface which propagated towards the shaft center due to crevice corrosion, where the clearance at the back of the O-ring was between 0.007 mm and 0.0395 mm which met the conditions for crevice corrosion to occur in the range between 0.025 and 0.15 mm [14].

The gap opportunity for crevice corrosion to occur is still available at 0.0145 or 58%. The shaft of a multi-stage double-shell centrifugal pump feeding high-concentration boric acid water from a nuclear power plant broke after operating for 2913 hours with 25 starts and stops because the radius of the stress relief groove as measured in the actual results was 1.2 mm smaller than the design (2 mm) in the necking position of the balance thrust clasp of the pump shaft [15]. It is recommended that the groove radius be made 2 mm to reduce the stress concentration that occurs. The SS 316L boric acid recycling pump shaft experienced premature fracture after the pump had been operating for 1 year at a 1,000 MW nuclear power plant due to triple stress concentration by the transition arc radius, surface defects, and inclusion aggregation along 308.26 micrometer which initiated the occurrence of cracks, microcrack coalescence, propagation of crack, and finally fracture between the rotor and impeller shaft [16]. The cooling water pump shaft made of SS AISI 304 experienced fracture due to torsional fatigue whose crack started at the propeller side keyway. The keyway experienced loading at the bottom of the keyway which had a rough surface and appeared to experience local deformation with the shaft material having a Ni content of 7.83% which was lower than the standard of 8-11% [17].

Milling results with rough surfaces should be avoided so as not to act as initial cracks in the keyway and Ni content lower than the minimum standard of 8% by 2.1 percent should be increased to a maximum of 11% to make the material more ductile. The positive displacement motor drive shaft of 40CrNiMo steel is subjected to a drilling pressure of 100 kN and a torque of 20 kNm, the maximum stress on the outer thread surface near the shoulder reaches 658 MPa, which is greater than the maximum stress on the root of the first tooth of the inner thread reaches 571 MPa, so that it breaks between diameter 119 mm and diameter 105 mm [18]. The working stress of 658 MPa is 15.2% greater than the root stress of the first tooth of its internal thread. Centrifugal water pump shaft occurs initial crack in the keyway area whose surface is brittle, the crack propagates, and finally breaks. Impeller erosion due to cavitation causes a decrease in impeller mass of about 20% which according to the manufacturer, the impeller weight after fabrication is about 30.8 kg and after damage 24 kg after 3000 hours of operation.

The working tensile stress is 172 MPa, and the yield strength of the shaft material (chromium-nickel steel Cr17Ni2 H17N2) is 630 MPa (Polish Standard PN74/H-93.004) which means the reduction stress of the shaft material during normal pump operation is relatively low at about 27% with a safety factor of about 3.7 [19]. Brittle keyway surfaces are prone to crack initiation, especially at corners with small radii that need to be enlarged. Failure of the 7-stage centrifugal pump shaft for feed water in the HRSG (heat recovery steam generator) of a class 9F gas turbine generator unit from SS 17-4PH martensite was caused by lower torsional stress due to fewer alloying elements and alloy strengthening phases (ε-Cu, η-Ni) in the form of microsegregation and coarse $M_{23}C_6$ carbides at the grain boundaries, thereby accelerating the nucleation of microholes causing the formation of cracks at the thread roots [20]. The nucleation source is supported by the reduction of alloying elements and alloy strengthening phases (ε-Cu, η-Ni) in the form of microsegregation and brittle $M_{23}C_6$ carbides.

The vertical water pump shaft made of SS ASTM A276-316L material experienced initial cracking in the upper left arc and lower right arc opposite to its axis and high SC in the upper left arc on the fracture surface, and scratch zones were along the shaft arch at the bottom and lower right, cracks propagated oppositely and broke in the split ring groove with a depth of 5 mm found 4.4 m down from the top of the shaft 10 m [21]. Shaft imbalance creates eccentricity when the shaft rotates, creates bending deflection in a certain direction when the shaft rotates and misalignment of the bearing shaft axis must be avoided so as not to increase the bending stress of the shaft. The initial crack in the salt water pump shaft made of SS AU79TY316 material occurred on the inclusions with the roughest surface with a high hardness value, 2200 HV, and there was intergranular crack growth that hit the grain boundaries, which is known as dynamic crack propagation, after which the material could no longer withstand the load, which finally caused the shaft to rapidly fatigue, which is known as static crack propagation [22].

Brittle fracture initiates in areas with higher hardness (> 2200 HV) in the inclusion region. A hollow transmission gear shaft made of 20CrMnTi steel with a yield strength of 850 MPa and a tensile strength of 1100 MPa in a two-way gear pump fractured in a test bench after operating for 60 hours at a pressure of 21 MPa and a speed of 2000 rpm, which initially cracked due to multiple crack sources resulting from machining on the shaft arc which resulted in SC and worsened crack initiation, propagation, and finally fracture at the end of the gear [23]. The final machining stage should ensure that the surface is not rough (smooth enough) in order to avoid the occurrence of initial cracks. The 50.8 mm in diameter ship's propeller shaft made of SS AISI XM-19/UNS S20910 material with a power of 2x720 HP which had been in operation for 14 years, broke due to surface defects in the form of dents or indentations 100 μm deep, and scratches resulting in rotational flexural fatigue which initially cracked near the keyway area [24]. It is important to avoid defective or rough surfaces or wounds, especially where there is potential for initial cracking, such as dents as deep as 100 μm. A 158 mm in diameter of crankshaft made of AISI 4340 material for a marine diesel engine, 4 stroke, 12 cylinders, V-shaped arrangement, 3600 kW power at 1200 rpm, broke due to wear which became an initial crack 300 μm below the fillet surface and a non-metallic inclusion 180 μm long on the main shaft near the crank cheek, crack propagation, and finally fatigue failure at the main bearing journal number 5.

According to the Soderberg criteria requiring a minimum crankshaft of 214 mm in diameter, the stress that occurs is 457 MPa, lower than the yield strength of 822.75 MPa and the predicted fatigue life of 10750 hours under full load operating conditions at 1200 rpm, but failure at 1,948 hours [25]. The crankpin diameter was 26.2% smaller than the required 214 mm, greatly reducing the strength. The stress of 457 MPa reached 55.6% of the yield point, causing cracks. Failure occurred 81.9% earlier than the estimated 10750 hours. The turbocharger shaft made of Ck45 steel has a strength of 497.5 MPa, with a fillet radius of 0.4 mm to fracture in the area of diameter change from 12 mm to 8 mm, because the fillet radius is too small, which in the original design, the alternating von Mises stress of 649.17 MPa exceeds the endurance limit, and the enlargement of the fillet radius to 2 mm reduces the stress to 482.34 MPa which meets the requirements, so that the fatigue life increases from 21983 cycles to 39550 cycles at a combination of bending of 18.34 Nm and torque of 4.83 Nm [26]. The original shaft stress of 649.17 MPa exceeded the endurance limit by 81.2%, and the redesign decreased by 25.7%, approaching the safe range. The fatigue life increased by 79.9%, indicating that high SC was the cause of failure. The crankshaft with ASTM E41 standard material for wheel loader diesel engine operated for 4800 hours, broke at the 5th crankshaft with 82 mm, DIN 1.7225 (42CrMo4) material has a tensile strength of 900-1100 MPa and while the tensile strength of sample 1 and sample 2 is 823.92 and 832.3 MPa is still smaller than the standard [27].

The tensile strength of crankshaft material is 8.4-25.6% lower than 900-1100MPa, so it is not strong enough. The four-cylinder diesel engine crankshaft of EN-GJS-800-2 ductile cast iron has a yield strength of 608 MPa and a tensile strength of 958 MPa, meeting the minimum EN-GJS-800-2 standards of 480 MPa and 800 MPa respectively, but the nodularity is only 70%, below the typical standard of ≥ 80%, so it breaks at the 4th crankpin, due to its surface hardness 261-273 HB is lower than the recommended 335 HB. The von Mises stress in the critical crankpin-web fillet zone is 236 MPa, equal to 39% of its yield strength, and the working range of the crankshaft is 400,000-1,350,000 km [28]. The lack of nodularity of 30% and surface hardness of 22% is still below the target of 335 HB reducing the fatigue resistance. A 650 MW nuclear power plant speed reducer shaft, 38.04 mm in diameter having a shear stress of 740930 MPa and a working stress of 9 MPa, designed for 40 years fractured less than 12 years, because the initial crack occurred in the keyway shoulder area where the carburized martensite of ~1.2 mm contributed to increase the brittleness and hardness of the shaft to 584 HV, far exceeding the standard of 196 HV, and A-type sulfide inclusions (level > 3) were present near the crack origin [29]. Early brittle fracture occurred at low stress (9 MPa), due to poor heat treatment, and high surface hardness.

Keyway areas initiated cracks due to a combination of microstructural and geometrical defects, although far from overload conditions. The pump shaft of a water storage unit that has been in normal operation for 35 years made of 34MoCN15 steel (Romanian standard STAS 791-80) was broken, because there were several initial points of cracks distributed around the shaft, scratches and the appearance of wavy steps on the notch edge on the component surface as high local stress concentration, then merged through the tear peak when the crack grew larger and finally fatigue fracture occurred in the 2nd notch area with a radius of 0.5 mm and a depth of 2.28 mm near the end of the 3490 mm shaft [30]. The fracture started from the surface surrounding the 34MoCN15 steel shaft cylinder, propagated to the center and finally broke in the center area of the shaft. A stepped shaft of AISI 1065 with a yield strength of 443 MPa and a tensile strength of 466 MPa in a Qatar rubber recycling plant fractured, due to high stress concentration in the region of diameter change from 30 mm to 34 mm and shaft misalignment due to inadequate adjustment which caused excessive vibration [31]. The fillet radius on diameter changes should be made close to the radius of the rolling bearing.

3. Research Methods

The analysis method is carried out by tracing the results of previous shaft failure studies, studying mechanical property testing methods, composition testing, hardness testing, studying simulation results, stress concentration studies, torsional stress studies, bending stress studies, defect studies, crack studies, scratch studies, inclusion studies, void studies, corrosion studies, wear studies, heat treatment studies, comparisons between study results, discussions and drawing conclusions.

4. Results and Discussions

The results of the analysis of various shaft failures that have been used in various applications by previous researchers are summarized in for applications, causes of failure, crack initiation locations, and shaft materials are shown in Table 1.

Table 1 - Application, failure causes, crack initiation locations, and shaft materials

No.	Shaft application	Cause of failure	The initial location of the crack	Materials
1	Centrifugal ammonia pump	The material stress is lower than the allowable stress	Near the keyhole	Stainless steel 17-4 PH
2	Centrifugal ore slurry pump	Concentration of stresses	Bottom of the thread/di bagian bawah ulir	AISI 4140
3	Double screw pump	Low viscosity	Keyway	Martensitic stainless steel ($05Cr_{17}Ni_4Cu_4Nb$)
4	Multistage centrifugal ammonia pump	Friction between the ring and shaft	At the point of friction with the ring	SS 410
5	Centrifugal pump	Operating beyond fatigue life	Not mentioned	S35C
6	Hydraulic power unit of hydraulic axial pump	Due to torsional stress	At the base near the keyway	AISI 4140
7	Shipping pump	Working stress exceeds shear stress	At the keyway end	AISI 316
8	Submersible pump	Micro cracks/cavities due to corrosion	At a depth of 1551.5 m from the ground surface	Monel K-500 nickel-based alloy
9	Condensate pump	Stress concentration of sharp corner cracks	The sharp edge of the outer surface of the shaft where MnS inclusions	Martensitic SS 416
10	Centrifugal pump	Scratches on the shaft surface and the Mo content was < standard	At a distance of 120 mm from the 60 mm in diameter shaft	APP-4 stainless steel
11	Submersible vertical centrifugal pump	The presence of foreign objects in the form of sand and a bolt in the pump chamber	In the cross-sectional reduction area next to the rolling bearing seat	SS 304
12	Centrifugal pump	MnS and Al_2O_3 inclusions	Below the keyway	06Cr17Ni12Mo2Ti austenitic forged steel
13	Right front wheel drive car	Due to the loosely tightened shaft nut retreating	At the nut thread	S45C
14	Feed water booster pump	Due to crevice corrosion	At 3 initial crack points on the shaft surface	Martensitic stainless steel (equivalent to China X3CrNiMo13-4)
15	Multi-stage double-shell centrifugal pump	The radius of the stress relief groove (1.2 mm) is smaller than the design (2 mm)	In the necking position of the balance thrust clasp of the pump shaft	Z5CND13-04, low carbon & high strength martensitic SS (ZG0Cr13Ni4Mo Chinese brand, F6NM (S41500) ASTM-A182)

16	Boric acid recycle pump	Due to triple stress concentration by the transition arc radius, surface defects, &inclusion aggregation along 308,3μm	Between the rotor and impeller shaft	SS 316L
17	Centrifugal cooling water pump	Due to torsional fatigue	At the propeller side keyway	AISI 304 stainless steel
18	Drive shaft of positive displacement motor	The max. stress on the outer thread surface near the shoulder reaches > the max. stress on the root of the 1^{st} tooth of inner thread	Between dia. 119 mm and dia. 105 mm	40CrNiMo steel
19	Centrifugal sea water pump	Brittleness of the keyway surface area	In the keyway area	Silicon bronze BK331 (CuSi3Zn3Mn)
20	Seven-stage centrifugal feed water pump	Due to fewer alloying elements and alloy strengthening phases (ε-Cu, η-Ni) microsegregation and coarse $M_{23}C_6$ carbides at the grain boundaries	At the thread roots	SS 17-4PH martensite
21	Circulating vertical water pump	High stress concentration in the upper left arc on the fracture surface	At the 5mm-depth split ring groove exists 4.4m down from the top of 10m shaft	SS ASTM A276-316L
22	Centrifugal sea sewater pump	Porosity and excessive vibration caused by bearing damage	In the middle area of the shaft	Austenitic stainless Steel SS AU 79 TY 316
23	Two-way gear pump	Due to multiple crack sources resulting from machining on the shaft arc	At the end of the gear	20CrMnTi steel
24	Ship-propellers	Due to surface defects in the form of dents and scratches	Near the keyway area	SS AISI XM-19/UNS S20910
25	Marine diesel engine	Due to wear which became an initial crack 300 μm below the fillet surface and a non-metallic inclusion 180 μm long on the main shaft near the crank cheek	At the main bearing journal number 5	AISI 4340
26	Turbocharger	The fillet radius is too small	In the area of dia. change from 12 mm to 8 mm	The Ck45 steel
27	Wheel loader diesel engine	A tensile strength is smaller than the standard DIN 1.7225 (42CrMo4)	at the fifth crankshaft with 82 mm	ASTM E41
28	The four-cylinder diesel engine crankshaft	The nodularity is only 70%, below the typical standard of ≥80% and surface hardness is lower than the recommended	At the 4^{th} crankpin	EN-GJS-800-2 ductile cast iron

29	A 650 MW nuclear power plant speed reducer	Carburized martensite ~1.2 mm contributes to increased brittleness and hardness of 584HV shaft, far exceeding standard 196HV, & A-type sulfides	In the keyway shoulder area	
30	Centrifugal pump water storage unit	There are some initial cracks scattered around the shaft, scratches & wavy steps appearing on the notch edges of the component surface	In the 2nd notch area with a radius of 0.5 mm and a depth of 2.28 mm near the end of the 3490 mm shaft	34MoCN15 steel (Romanian standard STAS 791-80)
31	A stepped shaft of a rubber recycling plant	Due to high stress concentration in the region of dia. change and shaft misalignment caused excessive vibration	In the region of diameter change from 30 mm to 34 mm	AISI 1065

Cases of centrifugal pump shafts that eventually broke [1], [2], [4], [5], [6], [10], [11], [12], [15], [17], [19], [20], [22], [30], showed the result of fatigue fracture with various causes of the material working lower than those near the keyway [1], due to high stress concentrations which result in fractures in the thread section [2], friction between the ring and the shaft, so that it broke at the friction part because the temperature increased which reduced the strength of the shaft [4], the shaft broke because it exceeded its fatigue life limit [5], the shaft broke near the keyway because it exceeded its torsional stress [6], scratches occurred and the Mo content was lower than the standard which broke as far as 120 mm from the shaft diameter of 60 mm [10], there was a bolt and sand in the pump chamber, so that the shaft was unable to rotate and eventually broke at the cross section next to the rolling bearing seat [11], there were inclusions of MnS and Al_2O_3 which eventually broke below the keyway [12], the radius of the stress relief groove (1.2 mm) was smaller than the design (2 mm) which resulted in the breakage at the groove [15], the shaft broke at the propeller side keyway due to fatigue stress [17], the brittleness of the keyway surface caused the shaft to break in that area [19], because there were fewer alloying elements and alloy strengthening phases (ε-Cu, η-Ni) in the form of microsegregation and coarse $M_{23}C_6$ carbides at the grain boundaries caused the shaft to break at the root of the thread [20], there was excessive porosity and vibration caused by bearing damage causing the shaft to break in that area [22], and there were several initial points of cracks spread around the shaft, scratches and the appearance of wavy steps on the notch edge on the component surface caused the shaft to break in the 2nd notch area with radius of 0.5 mm and depth of 2.28 mm near the end of the 3490 mm long shaft [30].

Shaft failure due to stress concentration occurred in centrifugal ore slurry pump which broke at the bottom of the thread [2], in condensate pump, the shaft broke at the sharp edge of the outer surface of the shaft where MnS inclusions were [9], in multi-stage double-shell centrifugal pump, the shaft broke at the position of the pump shaft annular groove [15], in boric acid recycle pump, the shaft broke between the rotor shaft and impeller [16], in centrifugal sea water pump, the shaft broke in the keyway area [19], in circulating vertical water pump, the shaft broke in the 5 mm deep split ring groove 4.4 m down from the top of the shaft 10 m [21], in two-way gear pump, the shaft broke at the end of the gear [23], in marine diesel engine, the shaft broke in the main bearing of shaft number 5 [25], on the turbocharger, the shaft broke in the area of diameter change from 12 mm to 8 mm [26], in a 650 MW nuclear power plant speed reducer, the shaft broke in the shoulder area of the keyway [29], in centrifugal pump water storage unit, the shaft broke in the 2nd notch area with a radius of 0.5 mm and a depth of 2.28 mm near the end of a 3490 mm long shaft [30], and a stepped shaft of a rubber recycling plant broke in the area of diameter change from 30 mm to 34 mm [31].

The shafts used in diesel engines experienced fracture at the main bearing journal number 5, due to wear which became an initial crack 300 μm below the fillet surface and non-metallic inclusions 180 μm long on the main shaft near the crank cheek [25], the working tensile strength was smaller than the DIN 1.7225 (42CrMo4) standard of 900-1100 MPa [27], and the nodularity of the material was only 70% which was still below the typical standard which should be ≥ 80% and the surface hardness of the shaft was lower than the recommended one [28].

Most shafts that are ductile in nature when purely twisted show fractures that are perpendicular to the axis of the shaft, where most of the shafts analyzed fracture relatively perpendicular to the axis of the shaft, indicating a relatively ductile nature [32]. If the load is a combination of torsion and bending with the indication of the beachmark, then the conclusion is that the shaft experiences fatigue fracture.

Other shafts were broken respectively in double screw pump [3], shipping pump [7], submersible pump [8], condensate pump [9], wheel drive car [13], booster pump [14], recycle pump [16], drive shaft motor [18], vertical water pump [21], gear pump [23], ship-propellers [24], turbocharger [26], speed reducer [29], and stepped shaft [31].

5. Conclusion

The conclusions that can be drawn from the discussion include:

1) The crankshaft of the wheel loader diesel engine after operating for 4800 hours, broke at the fifth crankshaft with 82 mm, the actual tensile strength of the shaft of 832.3 MPa is still smaller than the DIN 1.7225 (42CrMo4) standard of 900-1100 MPa;
2) The 55.5 mm condensate pump shaft of SS 416 broke, due to torsional fatigue from the sharp edge of the outer surface of the shaft where the MnS inclusion was as a design error and the MnS inclusion near the shaft surface as a metallurgical error; and
3) Other mechanical shafts that broke during operation are shown in Table 1.

Follow-up suggestions for the conclusions include:

1) Stress concentration should be avoided as much as possible by smoothing the machining surface, fillet radius should be maximized to approach the inner ring radius of the rolling bearing, inclusions should be avoided in the shaft material, cavities in the shaft should be avoided as much as possible, hydrogen gas trapped from the welding process or residual stress from the welding process should be subjected to post-weld heat treatment, heat treatment should be carried out in accordance with manufacturer recommendations, gaps that allow crevice corrosion to occur should be avoided as much as possible, changes in cross-sectional dimensions should always be given a radius that is not too small, changes in load should be avoided as much as possible to cause shocks so as not to increase stress concentrations, especially in critical areas of the shaft; and
2) If there is a rolling bearing on a machine that is worn and causing significant vibration, it should be replaced immediately to reduce the potential for faster crack propagation.

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