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Effect of Auxiliary Engine No. 1 Crankshaft Reconditioning on Lubricating Oil Temperature Rise in MT. SENIPAH

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ABSTRACT

This study investigates the effect of crankshaft reconditioning (regrinding/undersizing) on the lubricating oil temperature rise (ΔT) of Auxiliary Engine No. 1 aboard MT. Senipah. A quantitative descriptive field case study approach was employed, comparing thermal performance data before and after the reconditioning process under three load scenarios: 50%, 75%, and 85–90%. Primary data were collected through onboard measurements of lube oil inlet/outlet temperatures, bearing clearances, crankshaft deflection, and journal surface roughness. Secondary data were obtained from the Engine Log Book, workshop reconditioning reports, and manufacturer specifications. Post-reconditioning running tests revealed that the lubricating oil temperature differential (ΔT) remained within the manufacturer's acceptable limit of 10–12°C across all load conditions, with outlet temperatures consistently below the 75°C normal threshold. The frictional heat generation analysis confirmed that proper clearance restoration and adequate surface finish reduced hydrodynamic friction to within design parameters. A Fishbone (Ishikawa) root cause analysis identified four contributing factor categories—Man, Machine, Material, and Method—to potential post-reconditioning overheating. The study concludes that properly executed crankshaft reconditioning effectively restores thermal stability in the lube oil system, provided that bearing clearances conform to the maker's undersize chart and a controlled running-in procedure is followed.

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Keywords:

Crankshaft Reconditioning; Lubricating Oil Temperature; Auxiliary Engine; Bearing Clearance; Frictional Heat Generation; MT. Senipah

1. INTRODUCTION

The reliability and operational continuity of auxiliary engines aboard merchant vessels are fundamental to safe and efficient maritime operations. Auxiliary engines, commonly serving as diesel generator sets, supply the primary electrical power for navigation equipment, cargo handling machinery, and

essential hotel services. Among the critical rotating components in a diesel engine, the crankshaft occupies a pivotal role as the main power transmission element, converting reciprocating piston motion into continuous rotational output [1]. Due to the demanding operational environment in marine engine rooms—characterized by continuous high loads, elevated ambient temperatures, and vibration—crankshaft journals are susceptible to progressive wear, scoring, and surface degradation over extended service periods [2].

When crankshaft journal wear exceeds the manufacturer's allowable tolerance, standard maintenance practice in the maritime industry involves reconditioning through precision regrinding and undersizing rather than outright replacement, given the substantial cost and lead time associated with procuring a new crankshaft [3]. The reconditioning process entails machining the main journals and crankpin journals to a specified undersize dimension (typically 0.25 mm or 0.50 mm below the original standard diameter) and fitting correspondingly oversized bearing shells [4]. While this practice is well established, the post-reconditioning geometric changes—particularly alterations in bearing clearance, journal surface roughness, and shaft deflection characteristics—directly influence the tribological behavior of the bearing-journal interface and, consequently, the thermal load imposed on the lubricating oil system [5].

The lubricating oil system in a marine diesel engine serves the dual function of reducing friction between moving parts and absorbing the heat generated at bearing surfaces, cylinder walls, and piston cooling passages [6]. Under normal operating conditions, the temperature differential (ΔT) between the lube oil outlet and inlet remains within a predictable range established by the engine manufacturer. An abnormal rise in this temperature differential signals excessive frictional heating, inadequate oil film formation, or compromised cooling capacity—conditions that, if left unaddressed, can lead to bearing seizure, crankshaft damage, and catastrophic engine failure [7].

MT. Senipah, a tanker vessel operating in Indonesian coastal waters, experienced a documented case of elevated lubricating oil temperatures on its Auxiliary Engine No. 1 following a scheduled crankshaft reconditioning during a dry-dock period. The engine crew recorded lube oil outlet temperatures approaching the alarm threshold of 80°C during initial operation at moderate loads, prompting an investigation into the relationship between the reconditioning parameters and the observed thermal anomaly [8]. This operational incident provided a practical case study for examining how post-reconditioning clearance settings, surface finish quality, and installation practices affect lube oil temperature behavior under varying load conditions.

This research aims to: (1) compare the lubricating oil temperature rise on Auxiliary Engine No. 1 before and after crankshaft reconditioning under multiple load scenarios; (2) analyze the frictional heat generation resulting from post-reconditioning bearing clearance and surface roughness conditions; (3) identify the dominant root causes contributing to abnormal lube oil temperature rise through Ishikawa (Fishbone) analysis; and (4) provide operational recommendations for engine room crew regarding post-reconditioning running-in procedures and monitoring protocols.

2. LITERATURE REVIEW

Crankshaft Reconditioning in Marine Diesel Engines

Crankshaft reconditioning, also referred to as crankshaft regrinding or journal re-machining, is a standard repair practice employed when journal surfaces exhibit wear, scoring, or ovality beyond the manufacturer's permissible limits [3]. The process involves precision grinding of the main journal and crankpin journal diameters to a predetermined undersize, followed by the installation of bearing shells manufactured to match the new journal dimensions. According to classification society guidelines, the undersize increment typically follows a 0.25 mm step pattern (e.g., −0.25 mm, −0.50 mm, −0.75 mm), with the maximum permissible undersize determined by the minimum journal diameter that maintains adequate bending strength and fatigue resistance [4].

The critical post-reconditioning parameters include the final journal diameter, surface roughness (Ra), cylindricity, and hardness. The surface roughness of a reground journal directly influences the formation and stability of the hydrodynamic oil film in the bearing—a rougher surface promotes asperity contact and increased frictional heating, while an excessively smooth surface may impair oil retention [9]. Industry standards generally specify a target Ra value of 0.2–0.4 μm for reground crankshaft journals [10].

Lubrication Theory and Hydrodynamic Bearing

Performance

The operating principle of crankshaft bearings in marine diesel engines is based on hydrodynamic lubrication theory, first formalized by Osborne Reynolds in 1886 [11]. In a properly functioning hydrodynamic bearing, the rotating journal draws lubricating oil into a converging wedge-shaped gap between the journal and bearing surfaces, generating sufficient pressure to support the applied radial load without metal-to-metal contact. The minimum oil film thickness (h_0) is a critical parameter determined by the bearing clearance (c), eccentricity ratio (ε), journal speed (N), oil viscosity (μ), and specific bearing load (P) [12].

The Sommerfeld Number (S) is a dimensionless parameter that characterizes the operating condition of a hydrodynamic bearing. It is defined as $S = (r/c)^2 \times (\mu N/P)$, where r is the journal radius, c is the radial clearance, μ is the dynamic viscosity, N is the rotational speed, and P is the specific bearing load [11]. A higher Sommerfeld Number indicates a thicker oil film and lower friction coefficient, while a very low value suggests boundary or mixed lubrication with elevated friction and heat generation. Post-reconditioning changes in clearance directly alter the (r/c) ratio and therefore shift the bearing's operating point on the Sommerfeld curve [13].

Heat Generation and Transfer in Lubricating Oil Systems

In a marine diesel engine, lubricating oil absorbs heat from multiple sources: frictional heating at bearing surfaces, piston cooling, cylinder liner lubrication, and gear train friction [6]. The total heat absorbed by the lube oil is quantified through a heat balance approach, where the rate of heat absorption ($\dot{N}$_lube) equals the product of the mass flow rate ($\dot{m}$), specific heat capacity (C_p), and temperature differential (ΔT) across the engine: $\dot{N}$_lube = $\dot{m} \times C_p \times \Delta T$ [14].

Under normal conditions, the lube oil cooler dissipates the absorbed heat to the jacket cooling water circuit, maintaining the oil outlet temperature within the manufacturer's specified range. When frictional heat generation at the bearings increases—due to tight clearances, rough journal surfaces, or elevated loading—while the cooling capacity remains constant, the equilibrium outlet temperature rises. Persistent operation above the alarm temperature accelerates oil oxidation, reduces viscosity, and may precipitate bearing failure [15].

Root Cause Analysis: Ishikawa Diagram

The Ishikawa (Fishbone) diagram, introduced by Kaoru Ishikawa in 1968, is a systematic quality management tool used to identify and categorize potential causes of a specific problem or effect [16]. In marine engineering applications, the Fishbone approach is commonly structured around the 4M framework: Man (human factors), Machine (equipment conditions), Material (consumables and component quality), and Method/Environment (procedures and external conditions). This framework enables engineers to trace the root cause of operational anomalies—such as abnormal temperature rise—beyond the immediate symptom to its underlying contributing factors [17].

3. RESEARCH METHOD

Research Design

This study employed a quantitative descriptive approach with a field case study method. The research design focused on evaluating and comparatively analyzing the thermal performance of the lubricating oil system on Auxiliary Engine No. 1 aboard MT. Senipah before and after the execution of crankshaft reconditioning (regrinding/undersizing). Through the quantitative approach, parametric changes including bearing clearance, crankshaft deflection, surface roughness, and engine load variation were mathematically and thermofluidically linked to the dynamics of lubricating oil temperature rise (ΔT_oil). The study was structured in seven sequential phases: problem identification and literature review, historical data collection (pre-reconditioning), post-reconditioning dimensional measurement and installation verification, running tests and thermal variable monitoring, thermofluid and tribological calculations, root cause analysis using the Ishikawa (Fishbone) diagram, and formulation of conclusions and operational recommendations.

The research was conducted in the engine room of MT. Senipah during both underway (sailing) and anchored conditions. The pre-reconditioning baseline data were extracted from the Engine Log Book records covering the six-month period preceding the scheduled dry-dock maintenance. Post-reconditioning measurements and running tests were performed during the vessel's initial operating period following the completion of crankshaft regrinding at the repair workshop. The total research observation period spanned approximately eight months,

encompassing both the historical data review and the post-reconditioning monitoring phases.

Population and Sample

The population for this study comprised the complete set of operational thermal and mechanical data associated with Auxiliary Engine No. 1 on MT. Senipah. The study sample was delimited as follows: (1) pre-reconditioning baseline data, consisting of lube oil temperature readings, bearing clearance measurements, and crankshaft deflection records from the Engine Log Book for the six months prior to reconditioning; and (2) post-reconditioning test data, consisting of lube oil temperature readings, bearing clearance measurements, crankshaft deflection records, and journal surface roughness measurements obtained during the controlled running test program conducted at three load levels (50%, 75%, and 85–90%) after reassembly and commissioning.

Variables and Instruments

The research variables were classified into three categories. The independent variable was the post-reconditioning crankshaft condition, measured through main bearing and crankpin bearing clearance (mm), journal surface roughness Ra (μm), and crankshaft deflection values (mm). The dependent variable was the lubricating oil temperature rise, measured through the lube oil inlet temperature T_inlet (°C), lube oil outlet temperature T_outlet (°C), and the temperature differential $\Delta T_oil = T_outlet - T_inlet$ (°C). The control variables included the generator electrical load (kW/% load), engine speed (RPM), lube oil viscosity and grade (SAE), cooling water inlet/outlet temperatures at the lube oil cooler (°C), and lube oil pressure at the main gallery (bar). Table 1 summarizes the research instruments deployed.

Table 1. Research Instruments and Measurement Parameters

No.	Instrument	Parameter Measured	Unit	Accuracy
1	Engine Panel Thermocouple	Lube oil inlet/outlet temperature	°C	±0.5°C
2	Infrared Thermometer (calibrated)	Temperature verification	°C	±1.0°C
3	Outside Micrometer (0–100 mm)	Main journal & crankpin diameter	mm	±0.01 mm
4	Inside Micrometer / Plastic Gauge	Bearing clearance	mm	±0.01 mm
5	Dial Gauge Indicator	Crankshaft deflection (TDC, BDC, P, S)	mm	±0.01 mm
6	Surface Roughness Tester	Journal surface roughness (Ra)	μm	±0.01 μm

Data Collection

Data collection was divided into primary and secondary sources. Primary data were acquired through direct onboard measurements and included: (a) thermal measurements of the lube oil system, recording inlet and outlet temperatures at the Auxiliary Engine No. 1 using the engine panel thermocouple and verified with a calibrated infrared thermometer, along with cooling water temperatures at the lube oil cooler; and (b) geometric and mechanical component measurements, comprising journal diameter measurements before and after regrinding using an outside micrometer, bearing clearance measurements using plastic gauge and inside micrometer, crankshaft deflection readings at TDC, BDC, Port, and Starboard positions using a dial gauge indicator, and journal surface roughness assessment using a surface roughness tester.

Secondary data were obtained from official vessel and workshop documentation: (a) the Engine Log Book of MT. Senipah, providing historical temperature and pressure records from the pre-reconditioning period; (b) the Workshop and Reconditioning Report, containing the certified results of the crankshaft regrinding (work order report); (c) the Instruction Manual Book for Auxiliary Engine No. 1, providing the maker's technical specifications, allowable clearance ranges, and critical

temperature limits; and (d) the Lube Oil Analysis Report, documenting laboratory test results for viscosity, Total Base Number (TBN), and wear metal contamination levels.

Data Analysis

Quantitative data were analyzed comparatively using thermofluid and basic tribological equations. The temperature differential was calculated as $\Delta T_oil = T_outlet - T_inlet$. The frictional heat generation rate (Q_f) at the bearings was computed using $Q_f = f \times W \times v$, where f is the hydrodynamic friction coefficient (a function of the Sommerfeld Number $S = (r/c)^2 \times \mu N/P$), W is the radial bearing load (N), and v is the journal surface linear velocity (m/s). The lube oil heat balance was analyzed through the equation $\dot{N}_lube = \dot{m} \times C_p \times \Delta T_oil = \rho \times \dot{S} \times C_p \times (T_outlet - T_inlet)$, enabling assessment of whether the cooling system capacity remained adequate for the post-reconditioning heat load. Additionally, an Ishikawa (Fishbone) root cause analysis was conducted using the 4M framework (Man, Machine, Material, Method/Environment) to identify the dominant factors contributing to elevated lube oil temperatures.

3. RESULTS AND DISCUSSIONS

3.1. Results

Auxiliary Engine No. 1 Technical Specifications

Auxiliary Engine No. 1 on MT. Senipah is a four-stroke, inline marine diesel engine configured as a diesel generator set providing the vessel's primary electrical power supply. The principal technical specifications relevant to this study are presented in Table 2.

Table 2. Technical Specifications of Auxiliary Engine No. 1

Parameter	Specification
Engine Type	4-Stroke, Inline Marine Diesel
Function	Diesel Generator Set (DG Set)
Number of Cylinders	6
Rated Output	450 kW
Rated Speed	720 RPM
Number of Main Bearings	7
Standard Main Journal Diameter	120.000 mm
Standard Crankpin Journal Diameter	100.000 mm
Lube Oil Type	SAE 40 (Marine Diesel Engine Oil)
L.O. Cooler Type	Shell-and-Tube (FW Cooled)
T_{outlet} Normal Limit	≤ 75°C
T_{outlet} Alarm	≥ 80°C
T_{outlet} Trip / Shutdown	≥ 85°C

Pre-Reconditioning Condition and Baseline Data

Prior to the scheduled dry-dock maintenance, routine measurements revealed that the main journal and crankpin journal diameters on Auxiliary Engine No. 1 had worn beyond the manufacturer's permissible limit. The recorded bearing clearances exceeded the upper tolerance, and crankshaft deflection readings at several crank throws showed values approaching the alarm threshold. The pre-reconditioning condition measurements are presented in Table 3.

Table 3. Pre-Reconditioning Crankshaft Condition Measurements

Parameter	Standard Value	Max. Allowable	Measured Value	Status
Main Journal Ø (mm)	120.000	119.900	119.840	Exceeded Limit
Crankpin Journal Ø (mm)	100.000	99.900	99.830	Exceeded Limit

Main Bearing Clearance (mm)	0.07–0.13	0.20	0.22–0.26	Exceeded Limit
Crankpin Bearing Clearance (mm)	0.05–0.11	0.18	0.19–0.23	Exceeded Limit
Max. Deflection (mm)	±0.03	±0.05	+0.04 to −0.06	Near Alarm
Surface Roughness Ra (μm)	0.2–0.4	0.8	1.2–1.6	Exceeded Limit

The baseline lube oil temperature data from the Engine Log Book for the six months prior to reconditioning showed progressively deteriorating thermal conditions. At 75% load, the average lube oil outlet temperature had risen from 68°C in the initial months to 76°C in the final month before dry-docking, with the temperature differential (ΔT) increasing from 9°C to 14°C over the same period. These readings confirmed a direct correlation between journal surface degradation and rising thermal stress in the lube oil system.

Post-Reconditioning Dimensional Verification

The crankshaft was reground to the first undersize (−0.25 mm) at a certified marine workshop. Upon completion, the reconditioning report and onboard verification measurements confirmed the post-reconditioning dimensions as presented in Table 4.

Table 4. Post-Reconditioning Crankshaft Dimensions and Clearances

Parameter	Undersize Target	Measured Value	Status
Main Journal Ø (mm)	119.750 ± 0.02	119.745	Within Tolerance
Crankpin Journal Ø (mm)	99.750 ± 0.02	99.748	Within Tolerance
Main Bearing Clearance (mm)	0.07–0.13	0.09–0.11	Within Tolerance
Crankpin Bearing Clearance (mm)	0.05–0.11	0.07–0.09	Within Tolerance
Max. Deflection (mm)	±0.03	+0.01 to −0.02	Within Tolerance
Surface Roughness Ra (μm)	0.2–0.4	0.28–0.35	Within Tolerance

All post-reconditioning dimensional measurements fell within the manufacturer's specified tolerance for the −0.25 mm undersize. The journal surface roughness values of 0.28–0.35 μm indicated a high-quality finish that would promote stable hydrodynamic oil film formation. The crankshaft deflection readings were significantly improved compared to the pre-reconditioning condition, confirming proper alignment after installation.

Post-Reconditioning Running Test: Lube Oil Temperature Data

Following reassembly and a controlled running-in procedure, Auxiliary Engine No. 1 was subjected to a staged load test program at 50%, 75%, and 85–90% of rated capacity. Lube oil inlet and outlet temperatures were recorded at 30-minute intervals until steady-state conditions were achieved at each load level. The results are presented in Table 5.

Table 5. Post-Reconditioning Lube Oil Temperature at Steady State

Load Level	T_{inlet} (°C)	T_{outlet} (°C)	ΔT (°C)	L.O. Press. (bar)	$T_{CW\ in}$ (°C)	Status
50%	48	55	7	4.2	32	Normal
75%	52	62	10	3.8	33	Normal
85–90%	54	66	12	3.5	34	Normal

The running test results demonstrated that all post-reconditioning lube oil outlet temperatures remained below the 75°C normal threshold. The temperature differential (ΔT) ranged from 7°C at 50% load to 12°C at 85–90% peak load—both values falling within the manufacturer's acceptable range of 10–12°C for normal operation. Lube oil pressure readings at the main gallery were stable and within the expected operating range, confirming adequate oil flow through the restored bearing clearances.

Comparative Analysis: Pre- vs. Post-Reconditioning Temperature

To assess the effectiveness of the crankshaft reconditioning, a direct comparison was made between the pre-reconditioning baseline temperature data (from the final month of operation before dry-docking) and the post-reconditioning running test results. The comparison is summarized in Table 6.

Table 6. Comparative Lube Oil Temperature: Pre- vs. Post-Reconditioning

Load Level	Pre-Recon. T_{outlet} (°C)	Pre-Recon. ΔT (°C)	Post-Recon. T_{outlet} (°C)	Post-Recon. ΔT (°C)	ΔT Reduction (°C)
50%	68	11	55	7	−4
75%	76	14	62	10	−4
85–90%	82	18	66	12	−6

The comparative analysis reveals a substantial improvement in thermal performance following the crankshaft reconditioning. At the 75% normal operating load, the lube oil outlet temperature decreased from 76°C (exceeding the 75°C normal limit) to 62°C, representing a 14°C reduction. The temperature differential dropped from 14°C to 10°C, bringing it back within the acceptable 10–12°C range. At peak load (85–90%), the most significant improvement was observed: the outlet temperature fell from 82°C (above the 80°C alarm threshold) to 66°C, and the ΔT was reduced by 6°C from 18°C to 12°C.

These improvements are directly attributable to the restoration of proper bearing clearances and journal surface finish. The pre-reconditioning condition—characterized by excessive clearances (0.22–0.26 mm versus the 0.07–0.13 mm standard) and degraded surface roughness (Ra 1.2–1.6 μm versus the 0.2–0.4 μm specification)—had resulted in mixed-mode lubrication with elevated asperity contact and increased friction. Paradoxically, while excessively large clearances might seem to reduce friction by allowing more oil to pass, they actually destabilize the hydrodynamic oil film, leading to journal whirl and periodic metal-to-metal contact that generates significantly more heat [13].

Frictional Heat Generation Analysis

To quantify the thermal impact of the reconditioning, the frictional heat generation rate (Q_f) at the main bearings was estimated for both the pre-reconditioning and post-reconditioning conditions at 75% load using the formula $Q_f = f \times W \times v$. The calculation parameters and results are presented in Table 7.

Table 7. Estimated Frictional Heat Generation at Main Bearings (75% Load)

Parameter	Pre-Reconditioning	Post-Reconditioning
Journal Diameter, d (mm)	119.840 (worn)	119.745 (U/S −0.25)
Radial Clearance, c (mm)	0.12 (avg. measured)	0.050 (avg. measured)
Surface Roughness, Ra (μm)	1.40	0.32
Rotational Speed, N (RPM)	720	720
Journal Linear Velocity, v (m/s)	4.52	4.52
Estimated Friction Coefficient, f	0.012 (mixed regime)	0.004 (hydrodynamic)
Bearing Load, W (kN) – estimated	18.5	18.5

Frictional Heat, Q_f (kW)	**1.003**	**0.334**
Reduction (%)	**–**	**66.7%**

The analysis reveals that the estimated frictional heat generation at the main bearings was reduced by approximately 66.7% following the reconditioning—from 1.003 kW in the pre-reconditioning (mixed lubrication) regime to 0.334 kW in the post-reconditioning (full hydrodynamic) regime. This reduction is primarily attributable to two factors: the restoration of an appropriate radial clearance (from 0.12 mm average to 0.050 mm) that re-established stable hydrodynamic film conditions, and the significant improvement in journal surface roughness (from Ra 1.40 μm to 0.32 μm) that eliminated asperity-level contact points.

The transition from a friction coefficient of 0.012 (characteristic of mixed/boundary lubrication where surface asperities interact) to 0.004 (characteristic of full hydrodynamic lubrication with complete film separation) represents a shift to the left on the Stribeck curve toward the optimal operating region. This transition confirms that the reconditioning process successfully restored the bearing system to its design operating condition [11].

Lube Oil Heat Balance Assessment

The lube oil heat balance was evaluated using the equation $\dot{N}_lube = \dot{m} \times C_p \times \Delta T$ to determine whether the lube oil cooler capacity remained adequate for the post-reconditioning thermal load. Using the manufacturer's specified lube oil flow rate of 1.8 m³/h (0.0005 m³/s), SAE 40 oil density of 880 kg/m³, and specific heat capacity of 1.95 kJ/kg·°C, the heat absorption rates were calculated for the 75% load condition:

Pre-reconditioning: $\dot{N}lube = 880 \times 0.0005 \times 1.95 \times 14 = 12.01$ kW

Post-reconditioning: $\dot{N}lube = 880 \times 0.0005 \times 1.95 \times 10 = 8.58$ kW

The reduction of 3.43 kW in the heat load absorbed by the lube oil represents the combined effect of lower frictional heating at the restored bearing surfaces and improved oil film stability. The post-reconditioning heat absorption rate of 8.58 kW at 75% load falls well within the lube oil cooler's rated dissipation capacity, confirming that the cooling system operates with adequate thermal margin after the reconditioning.

Root Cause Analysis: Ishikawa Diagram

To provide a comprehensive diagnostic framework for the elevated lube oil temperatures observed prior to reconditioning and to identify potential risk factors for post-reconditioning thermal issues, an Ishikawa (Fishbone) root cause analysis was conducted. The analysis identified contributing factors across four categories, as summarized in Table 8.

Table 8. Ishikawa Root Cause Analysis – Lube Oil Temperature Rise

Category (4M)	Contributing Factor	Corrective / Preventive Action
Man	Incorrect bearing installation torque; inadequate running-in monitoring	Torque verification per maker's manual; continuous temperature monitoring during first 72 hours
Machine	Tight clearances post-reconditioning; fouled lube oil cooler tubes; deflection misalignment	Clearance verification against undersize chart; L.O. cooler chemical cleaning; deflection re-check after chocking
Material	Oil viscosity mismatch for undersize condition; grinding particle contamination in oil	Lube oil analysis (viscosity, TBN, wear metals) before start-up; full system flushing post-reconditioning
Method / Environment	Improper break-in / running-in procedure (immediate high-load operation); elevated engine room temperature	Graduated load increase: 25% → 50% → 75% over 24 hours; engine room ventilation check

3.2. Discussion

The findings of this study demonstrate that properly executed crankshaft reconditioning effectively restores the thermal equilibrium of the lubricating oil system on Auxiliary Engine No. 1 aboard MT. Senipah. The successful outcome is contingent upon three interdependent factors: dimensional accuracy of the regrinding process, surface finish quality, and correct post-installation procedures.

The pre-reconditioning condition presented a case of progressive thermal degradation driven by compound wear mechanisms. As the journal surfaces wore beyond tolerance, the resulting excessive clearances destabilized the hydrodynamic oil film, causing the bearing system to transition from the hydrodynamic regime into mixed and boundary lubrication zones. Simultaneously, the increased surface roughness (Ra 1.2–1.6 μm compared to the 0.2–0.4 μm standard) multiplied the number of asperity contact points, each acting as a localized heat source. The combined effect produced a frictional heat generation rate approximately three times higher than the design value, overwhelming the lube oil cooling system's capacity to maintain the outlet temperature below the normal limit.

The post-reconditioning results confirm the restoration of full hydrodynamic lubrication, as evidenced by the friction coefficient returning to the 0.004 range characteristic of a well-functioning plain bearing. The temperature differential reduction of 4–6°C across load conditions, combined with the 66.7% decrease in estimated frictional heat generation, provides quantitative evidence that the bearing-journal interface is operating within its designed tribological regime.

It is noteworthy that the initial post-reconditioning thermal anomaly reported by the crew—outlet temperatures approaching the 80°C alarm threshold during early operation—was not attributable to a deficiency in the reconditioning itself but rather to an abbreviated running-in procedure. The crew initially applied 75% load within the first hours of operation, bypassing the manufacturer's recommended graduated load increase protocol. Once the running-in procedure was adjusted to follow the 25% → 50% → 75% graduated sequence over 24 hours, the temperatures stabilized within normal parameters. This finding underscores the critical importance of operational procedures as a complement to precision mechanical repair [4].

The Ishikawa analysis provides a practical diagnostic and preventive framework applicable to any post-reconditioning thermal monitoring scenario in marine diesel engines. Of the four 4M categories, the Machine and Method factors were identified as the dominant contributors to the MT. Senipah case, while the Man and Material factors represented secondary risk areas that were mitigated through proper crew training and pre-start oil quality verification.

4. CONCLUSION

Based on the quantitative analysis and field case study conducted on Auxiliary Engine No. 1 aboard MT. Senipah, the following conclusions are drawn. First, the crankshaft reconditioning (regrinding to −0.25 mm undersize) successfully restored the lubricating oil temperature to within normal operating parameters, reducing the outlet temperature from 76°C to 62°C at 75% load and from 82°C to 66°C at peak load. The temperature differential (ΔT) decreased from 14°C to 10°C at normal load and from 18°C to 12°C at peak load, both within the manufacturer's 10–12°C acceptable range.

Second, the frictional heat generation at the main bearings was reduced by approximately 66.7% following reconditioning, from 1.003 kW (mixed lubrication regime) to 0.334 kW (full hydrodynamic regime). This reduction was achieved through the restoration of proper radial clearances (0.050 mm versus the pre-reconditioning 0.12 mm) and journal surface roughness (Ra 0.32 μm versus the pre-reconditioning 1.40 μm).

Third, the Ishikawa root cause analysis identified the Machine category (bearing clearance and surface condition) and Method category (running-in procedure) as the primary contributors to post-reconditioning thermal anomalies. Adherence to a graduated load increase protocol (25% → 50% → 75% over 24 hours) is essential for allowing bearing surfaces to achieve proper conformity before full-load operation.

Fourth, it is recommended that marine engine room crews implement the following practices after any crankshaft reconditioning: (a) verify all post-reconditioning clearances against the maker's undersize chart with a tolerance of ±0.02 mm; (b) conduct a full lube oil system flushing and oil analysis prior to initial start-up; (c) follow a minimum 24-hour graduated running-in procedure with continuous temperature monitoring at 30-minute intervals; and (d) perform crankshaft deflection checks at 500 and 1,000 running hours post-reconditioning to verify

long-term alignment stability.

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