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Contents

Experimental Investigation to the Effect of Heavy Load on the Performance of Different Grades of Lubricating Oil for Slow Speed Marine Diesel Engine

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المستلخص

من المعروف أن الحمولات الثقيلة هي من بين أكثر العوامل التشغيلية الحرجة المؤثرة علي آداء طبقة زيت التزليق داخل المحامل الانزلاقية الخاصة بمحركات الديزل البحرية بطيئة السرعات. وفي ضوء تلك الحقيقة تم تنفيذ بحث شامل لفحص آداء المحامل الانزلاقية عند الأحمال الزائدة والناتجة عن أقصي ضغط داخل غرفة الاحتراق لمحرك الديزل. علاوة علي ذلك قد امتد نطاق إجراءات التجارب المعملية ليشمل أكثر من مدي سرعة وهي ٤٠ – ٨٥ – ١٠٥ لفة في الدقيقة بما يتوافق مع أحمال المحرك (٤٠% - ٨٥% - ١٠٠%) علي التوالي وبالجمع أيضا مع اختبار أنواع مختلفة من زيوت التزليق تشمل 0W30 – 5W40 – 20W50. واستخدمت منصة تجارب المحامل الانزلاقية متعددة الأغراض لإجراء التجارب المعملية بسبب قدرتها علي استيعاب مختلف التغييرات في العوامل التشغيلية الرئيسية. ولقد أوضحت التجارب المعملية المنبنية أساسا على تتبع التغيرات الطارئة على ملف توزيع ضغط طبقة زيت التزليق وكذا قيم أقصى ضغط لطبقة زيت التزليق - أوضحت تأثير الأحمال الزائدة علي المحامل الانزلاقية لمحرك الديزل البحري. وبناء على كل ما تم اجراؤه من تجارب معملية في ظروف حمولات مثالية وزائدة اتضح قدرة زيت التزليق من درجة 5W40 على خلق أفضل ظروف تشغيلية عند مختلف السرعات المطبقة. وأيضا وفي ضوء مخططات التحميل التي تم الحصول عليها من خلال التجارب بات من الواضح أنه عند السرعة ١٠٥ لفة في الدقيقة والتي تمثل ١٠٠% من الحمل في حالة الحمولات الزائدة - اتضح أنه كلما زادت درجة لزوجة زيت التزليق كلما قلل ذلك من قدرة زيت التزليق على تحمل حمولات قسوى. وبالتالي لضمان الحصول على تشغيل آمن في كلا الحالتين من الحمولات المثالية والزائدة لابد من الانتقاء الصحيح لزيت التزليق الذي يوفر اللزوجة الكافية المثالية المطلوبة.

Abstract

Heavy loads are known to be among the most critical operational factors affecting the behavior of the lubricating oil film within journal bearing related to marine slow speed diesel engines. In light of the fact, a comprehensive investigation was carried out to examine the performance of journal bearing under heavy loads resulting from increased maximum pressure within the combustion chamber. Further, the research test trial procedures have been extended to cover the different speed ranges of 40 rpm, 85 rpm and 105 rpm, as corresponding to the engine loads (40 %, 85% and 100 % respectively), together with the different oil grades of 20W50, 5W40 and 0W30. Universal Journal Bearing Test Rig (UJBTR) was utilized for conducting the experimental test trials due to its capability to contain versatile variations regarding the key operational factors. Experimental test trials basically based on tracing the changes in the oil film pressure distribution and the maximum oil film pressure values, have given insight into the behavior of heavy-loaded journal bearing of diesel engine related to ships. Based all the experimental test trials under optimal and heavy load conditions, oil grade 5W40 was concluded to provide the most optimal operating conditions for all tested speed ranges.

Keywords: Hydrodynamic Lubrication, Pressure Distribution, heavy loads, Slow Speeds, Oil Grades.

1. Introduction

Hydrodynamic lubrication within main journal bearing of marine diesel engines is of foremost importance, as it prevents metal-to-metal contact between journal shaft and journal bearing. In this way, the failure of the journal bearing due to applied loads could be prevented. The oil film lubrication within main journal bearing is liable to complete failure in case heavy loads which result from early injection. Early injection (VIT) leads to the increase of the maximum pressure of the combustion gases. This, in turn, negatively affects the lubricating oil film within journal bearing. In such a condition, the propulsion system suffers complete failure, due to the misalignment of the crank shaft with the intermediate shaft and the propeller shaft. The serious impacts of this condition lead to unsafe navigation and big losses, represented in unnecessary delays and high maintenance costs. This ultimately leads the ship to become off hire.

Several research studies have been focused on the performance of journal bearing related to diesel engines. Most of them have especially investigated the different factors that could contribute to the failure of the main bearing. Forces resulting from diesel engine combustion, wear friction, misalignment, overloads and critical operational factors were probably the most prominent research areas among them. For enhancing journal bearing performance in real operating conditions Estupinan and Santos [1] evaluated the various strategies for applying controllable radial oil injection to main crank shaft journal bearing, observing the operational factors of minimum film thickness, maximum film pressure, friction losses and maximum vibration levels. Based on the conducted study, it was found out that the lubrication performance of main engine bearing could be enhanced via combining conventional hydrodynamic lubrication with controllable radial oil injection. Thomsen and Klit [2] proposed a flexure journal bearing design for enhancing operational behavior and hydrodynamic performance was evaluated based on oil film thickness, pressure as well as temperature. The operating conditions involved a rotational speed of 1500 rpm, and a load of 225 kN and the lubricant oil grade viscosity VG 32. Considering the predicted minimum film thickness, the proposed flexure journal bearing was also found to be able to operate at three times the misalignment compared to the stiff bearing. Liu et al. [3] investigated the lubricating properties of diesel engine main bearings at a speed of 2100 rpm via different applied loads. The derived outcomes ascertained that single cylinder misfire exerted a greater impact on the two adjacent main bearing loads and the axis orbits related to the two adjacent main bearings. Sander et al. [4] analyzed the behavior of automotive journal bearings under severe loading conditions. The research considered elastic deformation of the components under high pressure and at high shear rate. The operational shaft speed ranged from 1000 rpm up to 7000 rpm, whereas the applied load was from 40 kN up to 80 kN. Zadorozhnaya et al. [5] utilized the calculation of hydro-mechanical characteristics to trace the impact of the transient regime of the internal combustion engine on the resource of crank shaft bearing at different operating conditions. The study has made it possible to predict resource (wear) the bearings of the crankshaft of the internal combustion engine at different operating conditions. Xianbin and Jundong [6] obtained the elastohydrodynamic lubrication of the main bearing depending on the maximum dynamic pressure, the oil film thickness and friction power. The study established the dynamic model of marine four-stroke diesel engine body. The parameters of the main bearing comprised a journal speed of 900 rpm and the oil grade SAE 15W40. The research efforts could introduce a reference regarding the optimization design of the main bearing of 6-stroke diesel engine. Marey et al. carried out a series of research programs for investigating and enhancing the oil film lubrication within journal bearing in marine application. Marey et al. [7] involved the design and setup of a journal bearing test rig (JBTR). The study made it possible to trace the oil film pressure distribution at different speeds and constant load. Marey et al. [8] conducted a numerical study to investigate the oil film pressure profile within journal bearing. A new Computational Fluid Dynamic (CFD) model has been built for coupling future experimental test trials with computerized ones. Marey. [9] Utilized different oil grades for experimentally investigating the pressure behaviour of different lubricants within the hydrodynamic journal bearing, at different speeds ranging from 50 to 400 RPM at constant load. Marey et al. [10] enlarged the capabilities of journal bearing to contain much more sophisticated experimental test trials, via comprehensive and continuous modifications. The modifications involved adding a hydraulic loading system and full monitoring process via Supervisory Control and Data Acquisition (SCADA) system. The integrated systems have turned the structure into a Universal Journal Bearing Test Rig (UJBTR) that allowed for more extensive experiments for enhancing the performance of journal bearing and testing the most critical operational factors. Uncertainty and validation measurement analysis of UJBTR Marey et al. [11] has been carried out for ensuring the accuracy of the obtained outcomes. Li et al. [12] utilized the V8 engine model to study the lubrication performance of different poly-condensation of main bearing and concluded that oil film pressure, average filling rate of oil and misalignment of main bearing were the most important factors affecting the lubrication performance of bearings. Wan et al. [13] introduced a new method for monitoring the lubrication conditions of journal bearings in a diesel engine based on contact potential. It was concluded that asperity contact could be accurately monitored utilizing the contact potential. Additionally, monitoring the lubrication condition of a bearing utilizing contact potential was verified. Garcia et al. [14] have developed a numerical model to investigate the impact of wear and misalignment on the bearings of a stationary diesel engine at a shaft speed of 2000 rpm via changing the surface roughness as well as the bearing load. Increasing the load by 25 % was found to double the hydrodynamic pressure in the bearing. Nataraj [15] worked on enhancing the evaluation method of journal bearing performance in heavy-duty diesel engines at different engine speeds ranging from 800 rpm up till 2400 rpm under different applied load conditions. The study recommended the inclusion of multiple surface patches for changing the distribution and magnitude of bearing performance parameters such as pressure, friction losses and clearance height. Apresai et al. [16] utilized model equations to investigate diesel engine bearing wear at different engine speeds ranging from 720 rpm up to 1000 rpm and varying loads from 2000 N up to 10000 N. The main bearing was assured to ease the rotations of the crankshaft and to hold the forces created on the piston via the combustion of the mixture of compressed air and fuel in the combustion chamber.

Based on the above survey of previous research efforts, it is obvious that the operational conditions of lubricating oil film within journal bearing, and regarding marine slow speed diesel engines under overloads, were not given sufficient investigations.

2. Crank Shaft Main Bearing Assembly Forces

Circumferential Grooved Bearings (CGB) have been ascertained to provide the optimal solution in regard to marine modern large slow speed diesel engines. The reasons why such bearing type represents the best alternative involve firstly its ability to absorb strong shock-like loads arising from combustion gases and weight parts. This fact may be attributed to the lubricant oil film between the grooved bearing and journal shaft which represents a highly loadable bearing. Additionally, grooved bearings have proved their suitability regarding slow speeds and they are also characterized by their long operating life that often lasts an engine's entire life span.

The rotation of crank shaft related to slow speed diesel engine is driven by the forces resulting from the combustion gases. These gases are distributed on the area of the piston crown. The forces move to the piston rod, then they are transferred to cross head and finally they work on the connecting rod of each cylinder. The analysis of the forces on the main journal bearing is involved in Rui et al. [17]. The crank shaft is mainly subjected to forces from the big end of the connecting rod and the main bearings. These two types of forces ensure the balance of the forces working on the shaft. The forces working on the main journal bearing could exceed the required limits in case of heavy loads that result from incorrect injection timing.

3. Effects of Heavy Loads on Main Journal Bearing

Crank shaft of main journal bearing in marine diesel engine often suffers failure due to heavy loads resulting from the problems of poor Heavy Fuel Oil (HFO). The use of HFO leads to incomplete combustion of fuel oil. The problem of early injection leads to increasing the maximum pressure of combustion gases. All of the previously mentioned problems negatively affect the lubricating oil film in the crank shaft main journal bearing as outlined in Figure 1. The main risk occurs when oil lubrication moves from the hydrodynamic region to the boundary region, incurring the wear. The resultant wear leads to a very high risk resulting from the misalignment between crank shaft, intermediate shaft and propeller shaft. It can also incur the failure of the main bearing with the result of increased friction losses and hence the mechanical efficiency of the engine will decrease. The negative consequences of such a condition involve increased levels of emissions resulting from the increase in fuel consumption leading ultimately to decreasing the ship energy efficiency.

All these consequences are so much detrimental regarding the shafting system responsible for the safe navigation of the ship. Based on the previous facts, the work at hand is focused on the investigation of the effect of heavy loads on the performance of main journal bearing at different speeds, representing the loading program main engine on board ships.

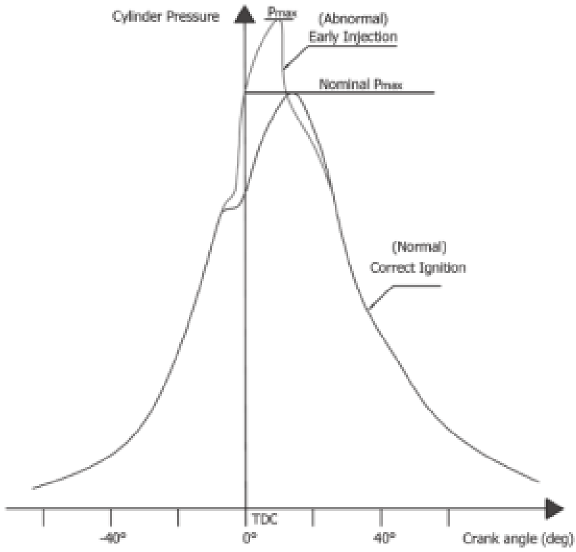

Figure 1: Typical faults shown on marine diesel engine draw card.

4. UJBTR Arrangement

UJBTR Marey et al. [10] utilized to accomplish the experimental tests, has involved four intrinsic systems, ensuring the efficient operation and accurate outcomes Figure 2. The shafting system consists of the drive motor and the drive shaft connected by the flexible coupling. Mounted on the stand, the shafting system is provided with two foundations for the supporting journal bearings and one thrust bearing. The main lubricating oil system comprises the lubricating oil pump unit, the filters and the lubricating oil cooler. All of the regulator valves, pressure gauges and sensors are included in the system. In addition, it is provided with the necessary thermocouples and oil hoses. The hydraulic oil system, in its turn, contains the hydraulic power pack unit, the filters as well as two hydraulic pistons. It also integrates the necessary hydraulic oil hoses, the proximity sensors, the pressure gauges and sensors. White metal is the material from which Circumferential Grooved Bearing (CGB) of UJBTR is made. The nominal diameter, width and wall thickness are 116 mm, 105 mm, 58 mm and 5.5 mm respectively. The main bearing clearance is 0.1 mm, whereas L/D ratio of Circumferential grooved bearing is 0.55. CGB is horizontally mounted on main journal bearing and contains pressure sensors (14No's) and thermocouples (14No's) Figure 3, for measuring the oil film pressure and temperature distribution within journal bearing. The determination of the distribution of the pressure sensors and thermocouples is based on a sensitivity study and in accordance with design criteria requirements. The flexible coupling connects journal shaft and drive shaft, and journal shaft is driven by AC motor with a maximum speed of 1450 rpm. Also, VFD controls shaft speed and rotation direction. For preventing misalignment and vibration of the shaft, the shafting system contains two supporting journal bearings. The lubricating oil system is operated to supply the main journal bearing and the supporting journal bearings with the lubricating oil. Further, the inlet port of the oil supply pressure is located on the upper part of journal bearings and the required pressure is determined according to the shaft weight and the applied lateral loads. Moreover, the lubricating oil system is regularly checked for ensuring proper operation and for avoiding any leakage, abnormal noise or vibration. Oil supply temperature is kept constant throughout the experimental test trials. The set point of the oil supply temperature is set to 40 °C on the SCADA system. The cooling system is then operated automatically via the closed loop system using the PID controller. The oil heater is turned on and the Thermocouple (TC) on the oil sump tank sends a feedback signal to the PLC. Here, the PLC sends a signal to the Zelio Controller to determine the cooling fan speed, whether it is low, medium or high. The cooling fan speed is controlled by the VFD in order to keep the oil temperature constant at 40 °C. In addition, the PID controller helps reduce the overshoot resulting from turning the oil heater on and off and also from turning the cooling fan on and off. The dead band width is also reduced to ± 1 °C via the cooling fan display page to keep the oil temperature between 39 °C and 41 °C, as neither the cooling fan nor the oil heater works in this range.

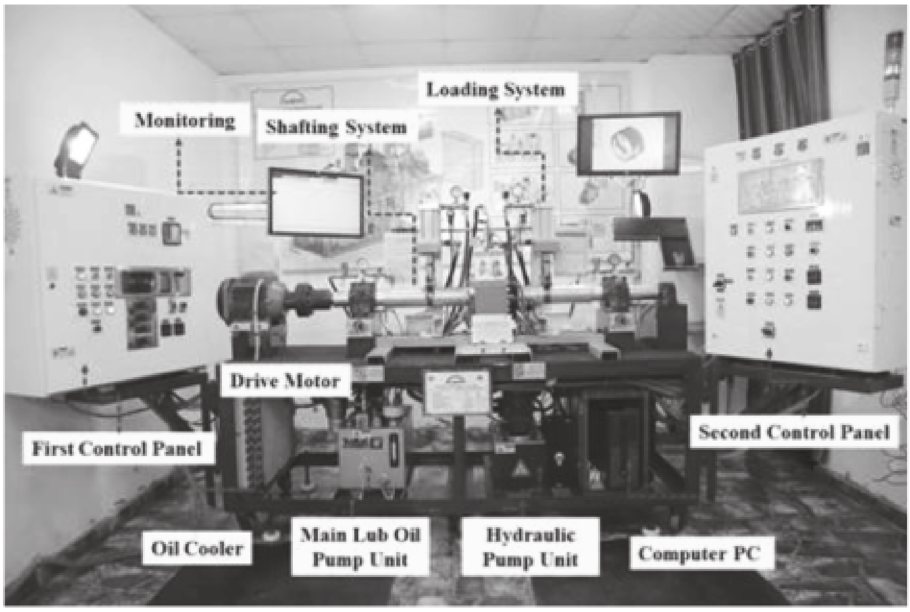

Figure 2: A schematic of UJBTR with operating systems [10].

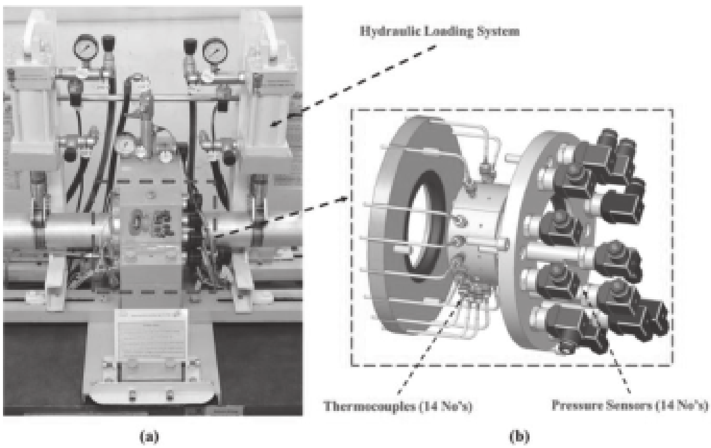

Figure 3: (a) UJBTR main journal bearing and (b) Pressure sensors and thermocouples distributed on grooved bearing circumference [10].

Finally, for ensuring the accurate performance of the whole UJBTR structure and its related systems, it is provided with a fully control system. It is operated via the advanced and highly precise SCADA control system for ensuring all procedures and experiments are accurately and efficiently manipulated and free of errors. Figure 4 shows an overview of UJBTR instrumentation and data acquisition system represented in the SCADA system integrated into UJBTR.

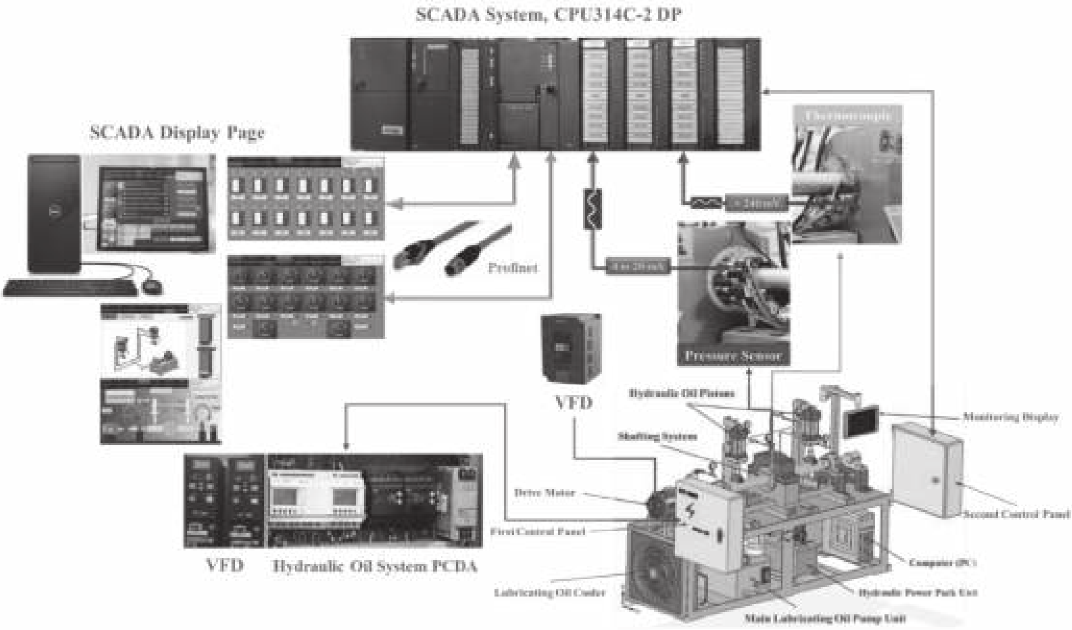

Figure 4: UJBTR Supervisory Control and Data Acquisition (SCADA) system (adapted from Marey et al. [10]).

5. <u>Experimental Test Trials on Main Journal Bearing</u>

Journal shaft of UJBTR has been utilized as a scale model (scale 3:16), of a marine slow speed main engine crank shaft type (SHD-MAN B&W 6S50MC) [18], to simulate heavy loads working on the main journal bearing crankshaft and resulting from the combustion gases of the combustion chamber. Further, the forces produced from the combustion gases have been represented by the simulated hydraulic loading system related to the UJBTR. Heavy load test trials have been carried out in accordance with the loading program related to the marine slow speed diesel engine. The test trials were conducted at different speed limits including 40 rpm, 85 rpm and 105 rpm, representing part load, Normal Continuous Rate (NCR) and Maximum Continuous Rate (MCR) respectively. At each of the previously mentioned speed ranges, the performance of the oil film pressure profile has been examined without load as well as under both optimal loads and heavy loads. Technical data and operational parameters of UJBTR are provided in Table 1. Further, specifications and properties related to the different oil grades 20W-50, 5W-40 and 0W-30 are shown in Table 2.

Table 1 Technical data and operational parameters for UJBTR.

Parameters	Value
L, Bearing Length	58 mm
D, Inner Diameter For Grooved Bearing	105.05 mm
Φs Shaft Diameter	104.97 mm
r, Radius for Journal Shaft	52.425 mm
C_0, Total Clearance	0.104 mm
C, Radial Clearance	0.052 mm
L/D ratio	0.55 mm
Eccentricity	0.032 mm : 0.015 mm
Operating Speeds	40, 85 and 105 rpm
W, Applied Loads	491 N : 6377 N

Table 2 Parameters and specifications for the different oil grades.

Parameters	Oil Grade Properties		
	20W50	**5W40**	**0W30**
Density at 15 °C	0.88 g/ml	0.85 g/ml	0.838 g/ml
Kinematic Viscosity at 100 °C	19 mm^2/s	14 mm^2/s	11.8 mm^2/s
Kinematic Viscosity at 40 °C	161 mm^2/s	84.7 mm^2/s	61 mm^2/s
Viscosity Index	136	171	193
Flash Point	260 °C	236 °C	217 °C
Pour Point	-24 °C	-36 °C	-42 °C

5.1. Test trial procedures

UJBTR is operated under full control of SCADA system, and the lubricating oil system is checked to ensure all journal bearings were properly fed with the lubricating oil with no leakage. The inlet port of oil supply pressure is located on the upper part of journal bearings, where the pressure is adjusted based on applied lateral loads. Oil supply temperature is kept constant during test trials. The following procedures illustrate the use of the cooling system for keeping oil temperature and for preventing fluctuations during experimentation. The temperature of the oil supply pressure is set to 40 °C on the SCADA system. The cooling system is then operated automatically via the closed loop system using PID controller. On turning the oil heater on, a feedback signal is sent by the Thermocouple (TC) on the oil sump tank to PLC. After that, a signal is sent by the PLC to the Zelio Controller to determine the cooling fan speed, whether it is low, medium or high. The cooling fan speed is controlled by the VFD, so that the oil temperature could be kept stable at 40 °C. As for the determination of the oil supply pressure, it depends on the lateral loads applied on the journal shaft. That is, oil supply pressure is continuously adjusted so that it can be consistent with the applied loads at the different speed ranges related to the experimental trials. Noteworthy that the test trials have been conducted repeatedly for each individual oil grade and the related readings were obtained via SCADA system.

6. Results and Discussion

In this paper, a comprehensive analysis of hydrodynamic performance of heavily loaded journal bearings and utilizing UJBTR was experimentally carried out. It comprised versatile operational conditions comprising different grade oils, where test trials were conducted according to the loading programs of marine slow speed diesel engines. The effect of such critical operational factors on the oil film pressure distribution profile within journal bearing will be discussed.

6.1. Experimental results with oil grade 20W50

Figures 5, 6 and 7 illustrate a number of outcomes that were derived based on conducting test trials utilizing oil grade 20W50. The pressure values obtained without load at journal shaft speeds of 40 rpm, 85 rpm and 105 rpm were observed to be very low, where they recorded 0.06 bar at the angles of 108° and 126°. In this region, the cavitation occurs due to the decreased oil pressure values, which negatively affects the journal bearing performance. However, when test trials were conducted under optimal loads, the pressure values obtained at the same angles were noted to rise due to the increased oil supply pressure. Consequently, the cavitation phenomenon disappears due to the increased pressure which obtained the value of 0.46 bar under the optimal loads 2943 N and 3336 N at both journal shaft speeds of 40 rpm and 85 rpm respectively, whereas at a speed of 105 rpm it assumed the value of 0.58 bar under 4817 N. Additionally, the recorded values of the maximum oil film pressure P_{Max} under optimal load and at an angle of 198° were 4.16 bar, 5.12 bar and 9.46 bar at speeds of 40 rpm, 85 rpm and 105 rpm respectively. While the obtained values of P_{Max} were 2.73 bar, 4.04 bar and 9.2 bar at the same previously mentioned angles and speed ranges but under heavy loads of 3993 N, 4170 N and 5592 N respectively. Hence, based on the conducted test trials it is noted that the differences in the values of P_{Max} under the impact of the heavy loads at the angle and speeds outlined before were 1.43 bar (34.8%), 1.08 bar (21.1%) and 0.26 bar (2.75%) respectively. Based on the previously mentioned outcomes, it is clear that operating at the slow speed of 40 rpm under a heavy load for long periods, represents a risk on journal bearing resulting from the lubrication moving from the hydrodynamic region to the boundary region. Consequently, the performance of journal bearing in marine diesel engines is negatively affected under heavy loads at part load operating conditions.

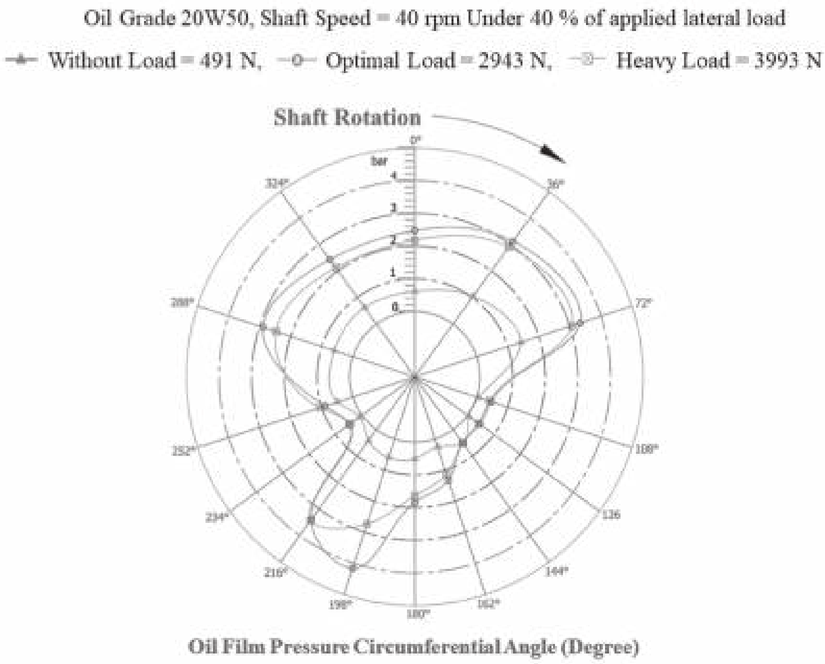

Figure 5: Variation of oil film pressure along the circumference of CGB.

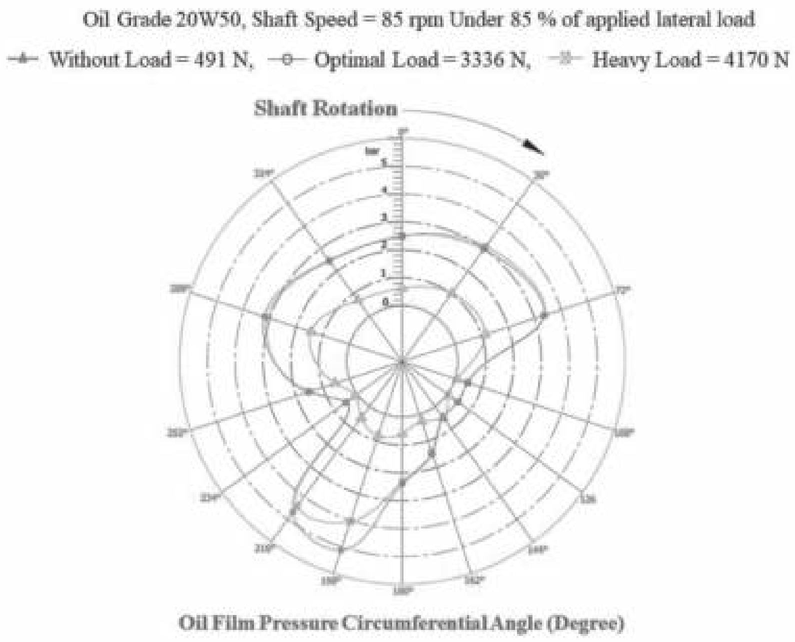

Figure 6: Variation of oil film pressure along the circumference of CGB.

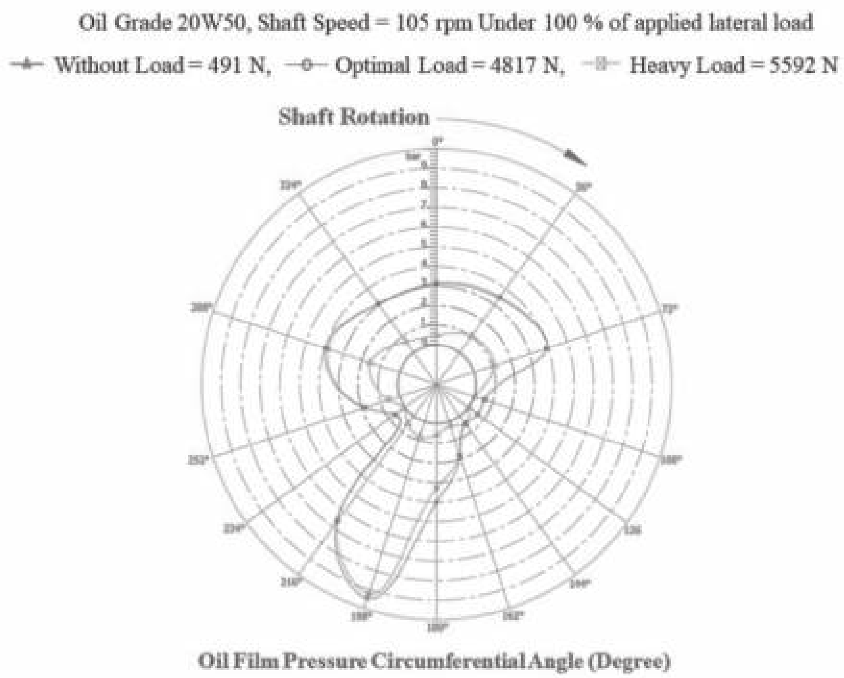

Figure 7: Variation of oil film pressure along the circumference of CGB.

6.2. Experimental results with oil grade 5W40

Another group of test trials on journal bearing were carried out also under heavy loads but utilizing the oil grade 5W40. Figures 8-10 show the impacts of heavy loads at different shaft speeds on the performance of the lubricating oil film within journal bearing in such a case. The cavitation phenomenon still exists at the shaft speed ranges of 40 rpm, 85 rpm and 105 rpm under the low oil pressure of 0.06 at the angles of 108° and 126°.

6.2.1. Performance of oil grade 5W40 Vs oil grade 20W50

The differences in the values of the P_{Max} in case of 20W50 grade oil and those recorded at an angle of 198° when 5W40 oil grade was utilized under the optimal load were -0.53 bar, 1.57 bar and -1.56 bar. In comparison, those differences in case of applying heavy load were 0.5 bar, 2.23 bar and -2.38 bar. The differences in the values of P_{Max} can be attributed to the variance in the degree of oil viscosity, which was 0.161 Pa.s for oil grade 20W50 whereas it was 0.0847 Pa.s for 5W40. The second reason for such difference is the different applied loads in case of oil grade 5W40, which were 4905 N, 5592 N and 5837 N respectively under heavy loads. Thus, the viscosity of oil grade 5W40 has the capability of tolerating higher loads than that of oil grade 20W50, but the value of the P_{Max} at the speed of 105 rpm under heavy load is lower by 26%.

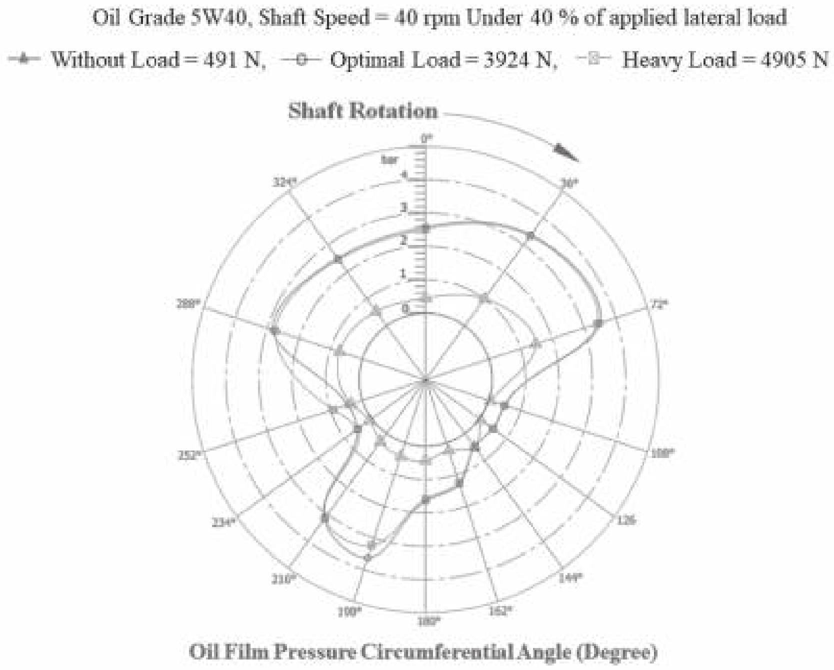

Figure 8: Variation of oil film pressure along the circumference of CGB.

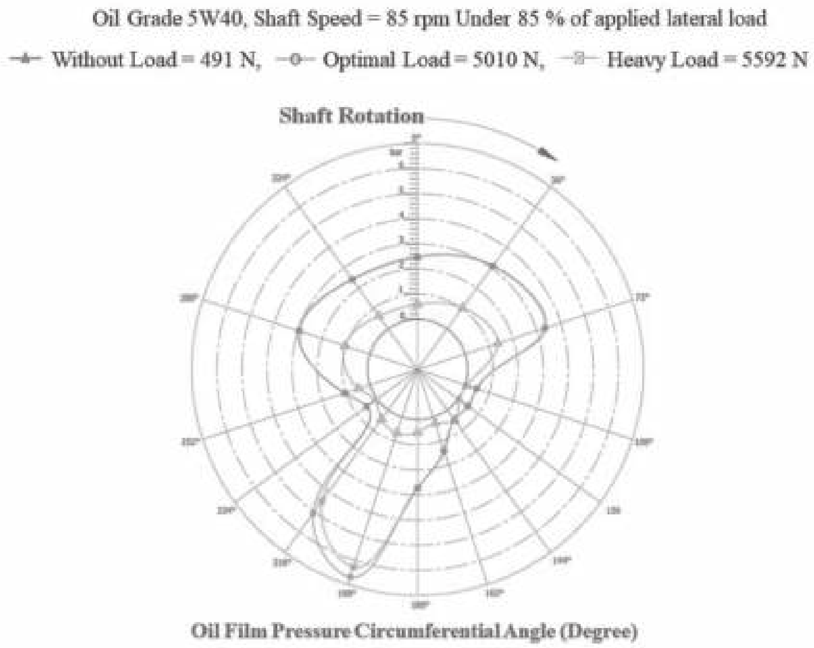

Figure 9: Variation of oil film pressure along the circumference of CGB.

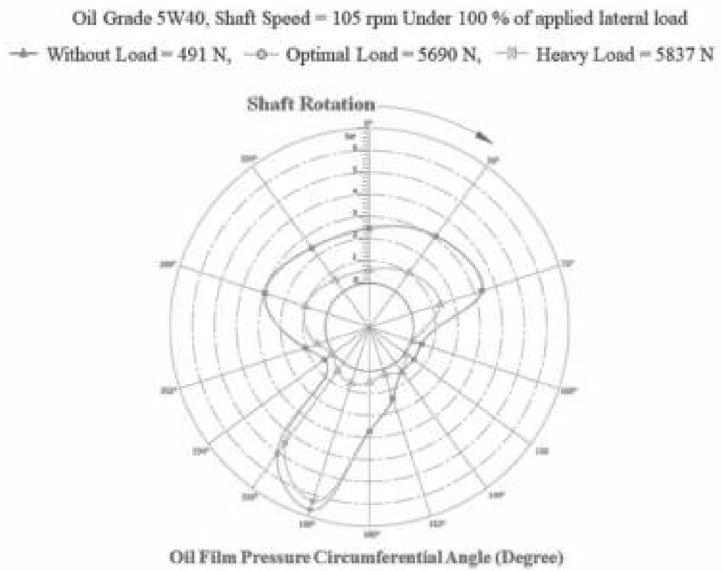

Figure 10: Variation of oil film pressure along the circumference of CGB.

6.3. Experimental Results with Oil Grade 0W30

In comparison with the results obtained with 20W50 and 5W40, experimenting with oil grade 0W30 shown in Figures 11-13 has yielded additional significant outcomes in regard to the impact of heavy loads on the behavior of the lubricating oil film within journal bearing. Again, the cavitation phenomenon which appeared while testing with the previously mentioned oil grades continues to exist with oil grade 0W30 at the same shaft speeds and angles, where the oil film pressure was 0.05 bar.

6.3.1. Derived observations with oil grade 0W30 Vs oil grades 5W40 and 20W50

As for the P_{Max} at the angle of 198° and the optimal load at 40 rpm, it is observed that both have obtained the values of 3.01 bar and 2747 N respectively. Those are lower values than those acquired when experimenting with 20W50 and 5W40. In relation to those values when obtained under heavy load, they have assumed the values of 2.7 bar and 4022 N, which are both lower values if compared with their peers obtained with oil grade 5W40. Therefore, it turned out at the slow speed of 40 rpm which represents the part load in marine diesel engine, oil grade 0W30 whose viscosity is 0.061 Pa. s will not the optimal option for operation under both optimal and heavy loads. The reason is that such oil grade will not tolerate the applied loads effectively and thus will represent a risk on the performance of journal bearing. It is also noted that at shaft speed of 85 rpm which represents NCR, the values of the optimal and heavy loads (5396 N and 6082 N) were higher with oil grade 0W30 than those obtained with oil grades 20W50 and 5W40. Also, with oil grade 0W30, the value of the P_{Max} at the angle of 198° was lowest in case of both optimal and heavy loads than it was with oil grade 5W40, assuming the values of 5.86 bar and 5.6 bar respectively. Additionally, at shaft speed of 105 rpm, it is observed that the value of heavy load was higher with oil grade 0W30 where it was 6377 N than it was with oil grades 20W50 and 5W40. Whereas the value of the P_{Max} recorded under heavy load was less than its counterparts obtained with oil grades 20W50 and 5W40, where it was 6.6 bar.

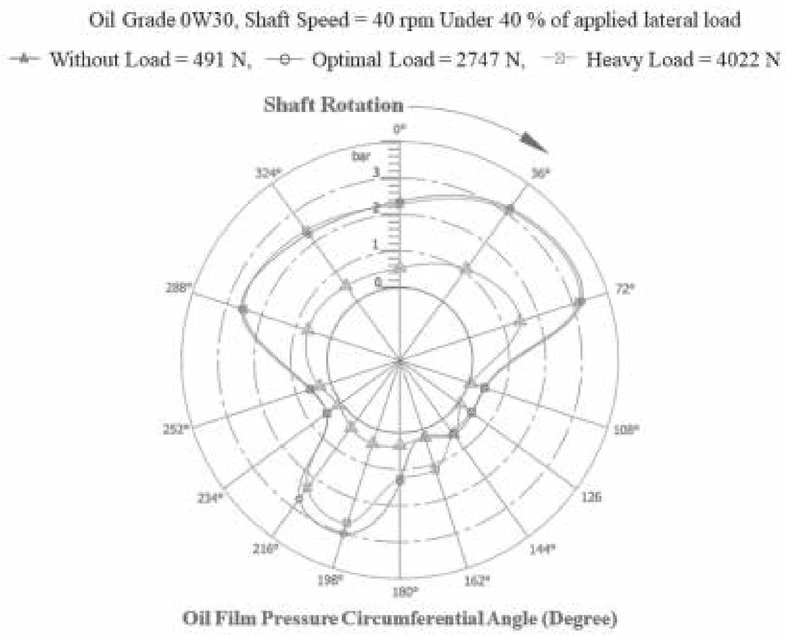

Figure 11: Variation of oil film pressure along the circumference of CGB.

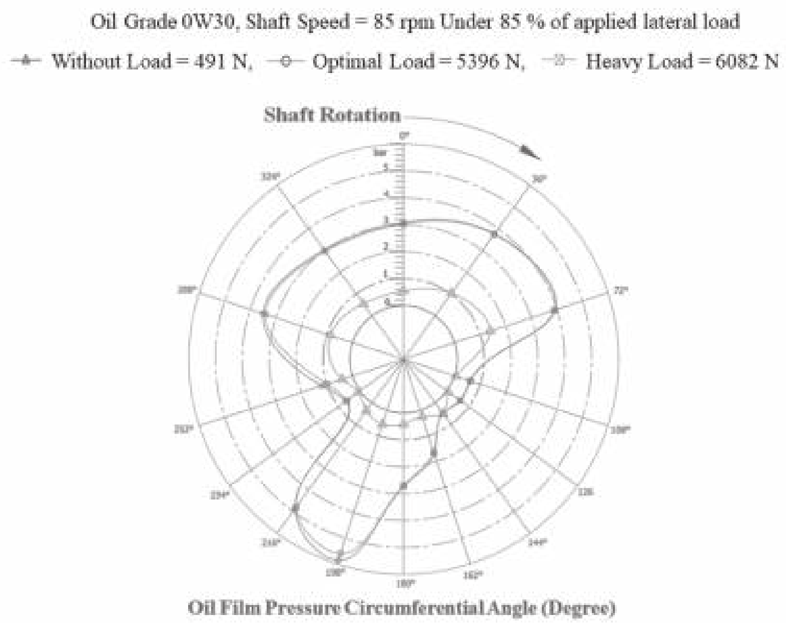

Figure 12: Variation of oil film pressure along the circumference of CGB.

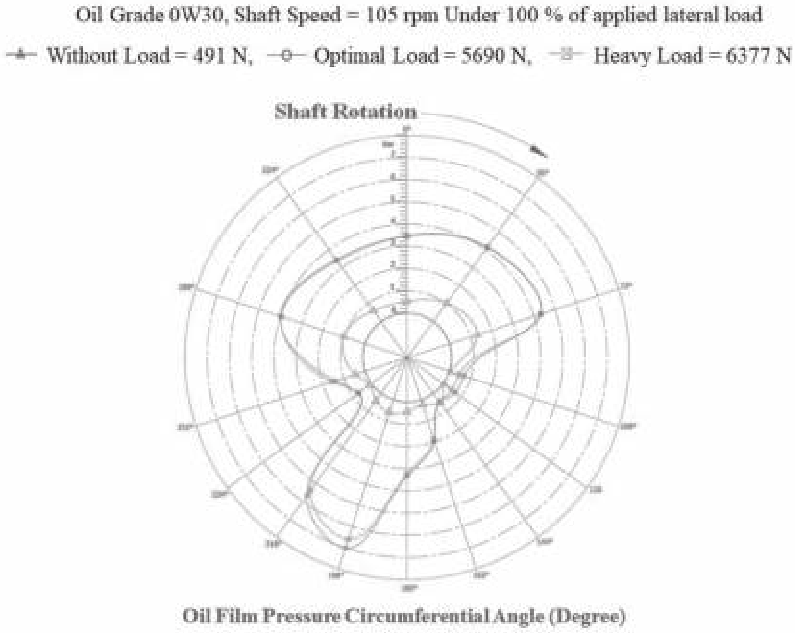

Figure 13: Variation of oil film pressure along the circumference of CGB.

The load diagrams Figure 14 illustrate the relation between shaft load and shaft speed in case of optimal load and heavy load. While Figure 15 shows the relation between shaft power and shaft speed at the same loading conditions. From the outlined loading diagrams, it is obvious that there is a positive relation those two operational factors throughout the course of the conducted experimental test trials utilizing the different oil grades previously mentioned.

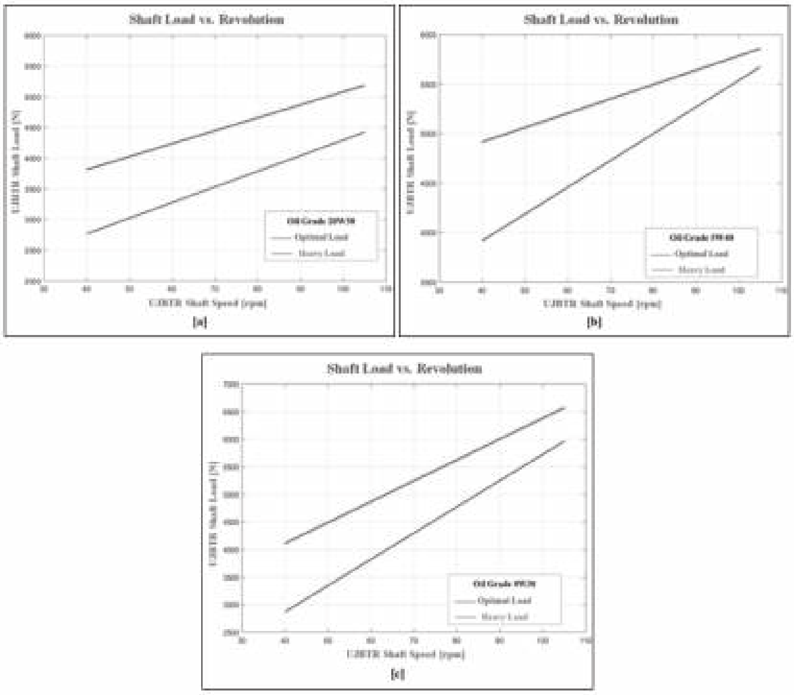

Figure 14: Load diagrams of UJBTR (Shaft Load vs. Revolution) for oil grades. (a) 20W50 (b) 5W40 (c) 0W30.

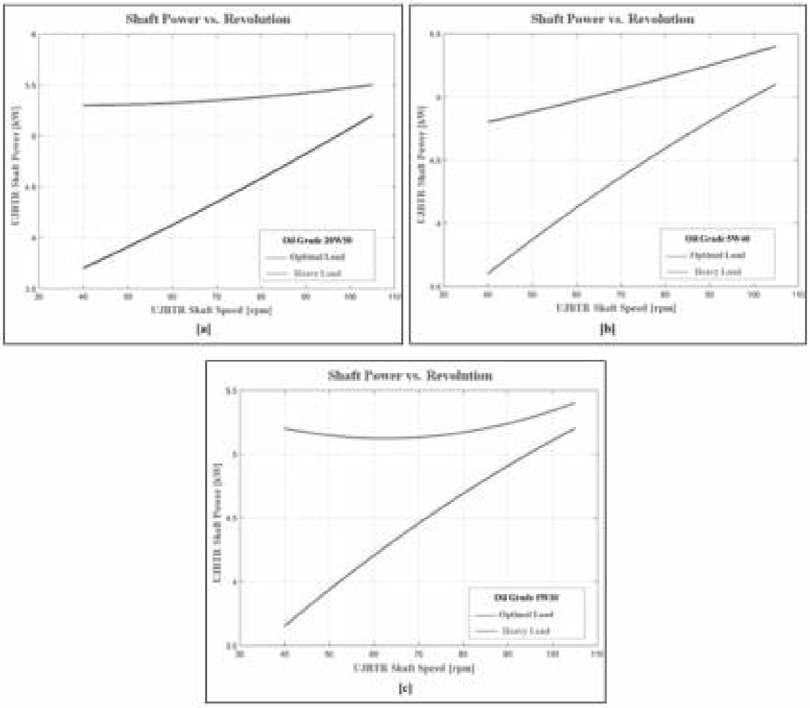

Figure 15: Load diagrams of UJBTR (Shaft Power vs. Revolution) for oil grades. (a) 20W50 (b) 5W40 (c) 0W30.

Conclusion

To realize the most possible optimal operating conditions regarding journal bearing in marine applications, the present work investigates the impact of heavy loads on the performance of journal bearing at different speeds, which represent the loading program for slow speed marine diesel engine. Further, the scope of research has extended to cover the influence of different oil grades (20W50, 5W40 and 0W30) in such operating conditions. Based on the obtained research outcomes and comparative studies of the registered characteristics, the following conclusions can be drawn:

- Throughout test trials conducted without load at all speed variations and different oil grades, the cavitation phenomenon occurred at both angles of 108° and 126°. This is due to the low values obtained in regard to the oil pressure profile at this specific region.
- With the increase of shaft speed and applied loads, the oil supply pressure increases with the result of drifting from the cavitation region.
- At increased shaft speeds the lubricating oil film within journal bearing is capable of tolerating the lateral applied loads.
- While oil grade 0W30 of lower viscosity 0.061 Pa. s at the slow speed of 40 rpm is not efficient regarding tolerating the applied loads, it has the capability to stand the exerted loads at increased speed ranges of 85 rpm and 105 rpm.
- Under heavy load condition, oil grade 20W50 has obtained the least ability in regard to tolerating the applied loads at all experimental test trial speed ranges. The high viscosity properties characterizing this oil grade results in high friction between the oil molecules, leading in turn to increased friction losses and the inability to apply further lateral loads.

- Throughout all conducted experimentation for optimal and heavy loads, oil grade 5W40 was concluded to be the most efficient oil grade in operational conditions. It has offered the optimal alternative among all tested oil grades as it provided the least loss in the P_{Max} value when moving from the optimal load to the heavy load test trials at all shaft speeds.
- According to the previously outlined load diagrams, at 105 rpm for 100 % of load in case of heavy load, it is concluded that the higher the viscosity grade of the lubricating oil, the less the capability of the lubricant to tolerate extreme loads and the higher gets the power of the shaft. Accordingly, selecting the oil grade with adequate viscosity will ensure safe operation in both cases of optimal load and heavy load.

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Nomenclature

C_0	total clearance	mm	PLC	programmable logic controller
C	radial clearance	mm	Profinet	process field net
D	inner diameter for grooved bearing	mm	PS	power supply
L	bearing length	mm	PT	pressure transmitter
L/D	bearing length/inner diameter for grooved bearing		SCADA	supervisory control and data acquisition
N	shaft speed	rpm	SM	signal module
P	motor power	kW	TC	thermocouple
P_{max}	maximum oil film pressure	bar	UJBTR	universal journal bearing test rig
P_0	nominal bearing pressure	bar	VFD	variable frequency drive
r	Radius for Journal Shaft	mm		
T	temperature	℃		
W	applied load	N		
Φ_s	shaft diameter	mm		

Dimensionless Group

P_0/P_{max}	maximum film pressure ratio

Greek Letters

μ	dynamic oil viscosity	Pa.s
ρ	lubricant Density	kg/m^3

Abbreviations

CGB	circumferential grooved bearing
CP	communication processor