OPTIMIZATION OF THE LUBRICATION CONDITIONS FOR MINIMIZATION OF THE OPERATING TEMPERATURE OF THE HIGH-SPEED BALL BEARINGS

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Bearing lubrication for high speed application (above 20,000 rpm) has significant influence on the bearing heat generation and consenquently bearing operating temperature. For application in aerospace industry the selection of lubricant type, its amount and lubrication method should be carefuly conducted in order to achieve desired life and minimal weight.

In this paper, experimental research was carried out in order to check whether and to what extent an open loop system can achieve similar bearing thermal behaviour as in a closed loop system. Tests were conducted on high-speed bearings of a turbo shaft engine. The experimental results showed that the open loop system, with oil mist lubrication, can achieve similar bearing operating temperature as the closed loop system, which operates with oil circulation, with the advantage of having a significantly lower mass. Also, a variation of lubrication conditions for open loop method, air pressure and oil amount, was carried out in order to minimize the operating temperature of high speed angular ball bearings. It has been shown that there is a combination of these two parameters that gives the least heating of high-speed bearings.

Key words: *High speed bearings, Operating tempearature, Oil mist, Closed loop lubrication*

1. Introduction

Rolling bearings are vital elements for transmitting motion and ensuring the accuracy of the axes of rotation of machine assemblies. They are characterized by high load capacity and high speed of rotation, which is limited by mechanical and thermal factors. On the other hand, they generate noise during operation due to rolling, require high manufacturing accuracy and precise assembling. These features are especially evident when it comes to high-speed bearings. With these bearings, the number of revolutions is greater than 10,000 rpm. In such working conditions, the aspects of generating and removing the amount of heat come to the fore. In [1], the load equilibrium model of angular contact ball bearings with thermal expansion was established to calculate the bearing loads. Thereafter, the coolant/lubricant, radial and axial structural constraints and assembly constraints were fully taken into account to study the heat generation and transfer of bearings, and then a novel multi-node thermal network model for angular contact ball bearings was proposed.

Since thermal performances are key factors impacting the operation of angular contact ball bearings [1], special attention must be paid to the lubrication of such bearings. There are several different ways to lubricate high speed bearings: oil sump, oil jet, oil mist and grease lubrication. Grease lubrication is the simplest design method, but it has significant limitations that make it rarely used in turbomachinery [2]. From the tests conducted in [2] it has been shown that angular contact bearings can be operated up to a speed parameter of $0.87 \cdot 10^6$ DN (D is the bore diameter, N is revolutions per minute) with grease lubrication, and the life of the grease lubricated bearing at 100,000 rpm should be limited to 10 h without replenishment of the grease. These tests, however, were conducted with a light axial load. For the high-speed angular contact ball bearings, this can be beneficial since the results show that the applied axial load significantly influences the behavior of skidding due to the changes of internal load, orbital and rotation speeds of ball under different operating conditions; appropriate axial load can be determined to avoid severe skidding [3]. The phenomena that occur in high-speed oil-air lubricated bearings have not yet been studied in detail. For example, the conventional calculation of working life based on fatigue is questionable, because a statistical analysis in [4] showed that abrasion and not fatigue is the main failure mode for such bearings. The domination of abrasion failure mode in rolling bearing under oil-air lubrication might be due to the existence of haze particles in the lubrication air.

In [5], tests were conducted to compare the performances of oil mist and conventional oil sump lubrication in terms of operating temperature and friction with variation in load and speed. The two methods of lubrication were compared directly under endurance test conditions. The oil mist lubricated high-precision angular contact test bearings ran cooler by about 10°C. Research performed in [6] with 45 mm angular contact ball bearing run to speed of 135 000 DN run with sump lubrication and air-oil mist found that the air-oil mist method resulted in lower bearing temperatures on the order of 12 to 15°C. Also, the oil mist lubricated bearings had about 25 percent less friction. In addition to savings in energy, oil mist lubrication provided better wear and fatigue protection to the tested bearings [5]. Even lower bearing temperatures and higher power losses are obtained for high-speed, small-bore, angular-contact ball bearing with oil-jet lubrication in comparison to air-oil mist lubrication [7]. Given that the amount of lubricant supplied to the oil mist system is one of the most influential parameters [8], current research [9] is focused on the extreme minimization of the amount of lubricant, which is now measured in individual drops.

Optimization of different types of parameters is performed in order to design the most efficient oil mist lubrication system. In [10], the influence of the spray angle on oil-air two-phase flow and gas field in bearing chamber was analyzed in detail. The variation in structure parameters including the spray angle, tilt angle and diameter of pipe lead to a design of a new oil-air lubrication unit with low temperature rise, vibration acceleration and noise for ultrahigh-speed grinding machine tools. With the same goal, in addition to design parameters, many authors chose to vary different lubrication parameters (air pressure, amount of lubricant, lubricant flow rate) [7, 8, 11, 12] or operating parameters (dominantly speed) [3, 7, 13, 14].

One of the main obstacles for using the oil mist system is the environmental factor, however significant design improvements have been made in that field as well, so this system is successfully used even in electric motors [15]. Considering its advantages compared to other methods of lubrication, it has found a purpose in a wide variety of applications. In [16], the development of an ultrahigh-speed bio-generator with a power of 10 kw and an rpm of 30,000, which uses air mist for lubrication, was presented. Review paper on oil mist lubrication in metal cutting applications is given in [17].

In this paper, two different methods of lubrication of high-speed bearings used in the aviation industry were applied. An attempt was made to replace the proven closed loop lubrication system of high-speed bearings with an open loop (oil mist) lubrication system. During testing with an open loop lubrication system, air pressure and the amount of lubricant were varied in order to achieve similar thermal behavior of the bearings. It has been shown that lubrication parameters can be defined for gas generator rotor for turbo-shaft engine that enable thermally safe operation of the bearings even at the speed parameter of $1.24 \cdot 10^6$ DN.

2. Case study

The case study in this paper is a bearing assembly of a gas generator rotor for turbo-shaft engine TSE-200 Phoenix, Fig.1, developed by company EDePro – Engine Design and Production. This engine has 200kW of output power, which places it in the group of relatively small power engines, but on account of that it has a very high rotational speed of 62,000 rpm of the rotor. The rotor assembly consists of a compressor and turbine impellers. This extremely large rotational speed demands the usage of hybrid high precision bearings, special attention in design process of the bearing assembly [8,12] and a specific method of their lubrication. Bearings are installed in two sets for each of shaft support. First set is ordered as TFT combination comprised of SKF angular type bearing with designation 7004 ACE/HCP4A TFT L. This set acts as a fixed support point of the rotor. For axial free support two bearings with mark 71904 CD/HCP4A DB in back-to-back arrangement but with a brass cage, are used.

Fig. 1 – Turbo-shaft engine TSE-200 Phoenix on the left and its rotor with compressor and turbine on the right

The two different lubrication methods were applied on the engine:
• Lubrication by low viscosity oil in closed loop lubrication,
• Lubrication by mixture of oil and air i.e. oil mist in open loop lubrication.

The oil used in both systems is Aeroshell 500 turbine oil. It is synthetic ester oil, with kinematic viscosity 4.90 to 5.40 mm²/s.

2.1. Closed loop lubrication

This is a repeatedly tested and proven lubrication method for the engine in question. It includes the oil supply lines and nozzles for each bearing. It also implies several oil drainage lines in order to suck out used and excess oil from the bearing housing assembly. Beside drainage lines, seals are also necessary to retain the oil in the system. For these purposes, various non-contact seals were developed [18, 19]. The bearing housing assembly for oil circulation lubrication with low viscosity turbine oil Aeroshell 500 in close loop lubrication is presented in Fig. 2.

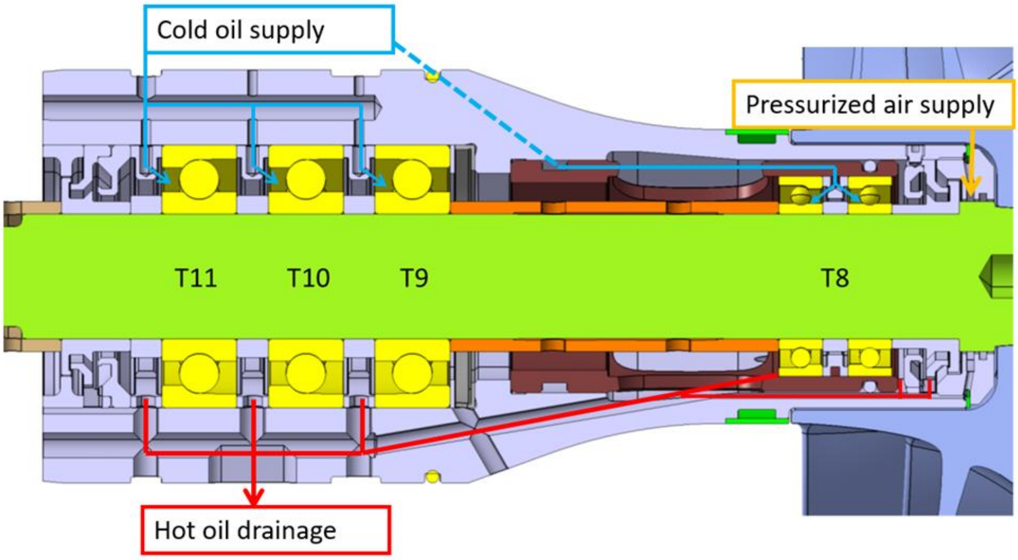

Fig. 2 – Closed loop oil lubrication, cold oil supply lines are presented with blue color, pressured air with orange color while suction lines with hot oil are shown with red lines

Another problem that needs to be solved is underpressure on the compressor intake side which tends to suck out the oil from the bearing housing. In order to overcome this problem and to compensate compressor vacuum, the overpressure is introduced through channels on the right side, see Fig. 2. This overpressure is delivered from the engine combustion process from the compressor section.

To achieve oil circulation, one drive pump is needed. For the oil drainage from the bearing housing assembly, another suction pump is used. The oil is afterwards led into the cooler in order to cool it down before leading it to the oil tank.

2.2. Open loop lubrication

In comparison to lubrication by low viscosity oil in closed loop lubrication, the oil mist lubrication method has low requirements for additional aggregates such as suction pumps, coolers, overpressure supply, valves and etc. It is a cost efficient and simple solution which allows the highest possible attainable speed of bearings rotation. For oil mist lubrication, the rotating bearing does not churn oil, which results in lower friction (hence lower energy loss) and lower operating temperature. Also, as the oil mist at room temperature flows over the hot operating surfaces of the bearings, some heat is removed by forced convection from the bearing, which results in lower operating temperatures [5].

For application in aircrafts, like in this case, it is usually used in spill out option, which means that after lubrication and cooling of the bearings, all oil mist gets induced into the combustion chamber and burned. This means that certain amount of oil is intended to be expandable. This amount is very small and is acceptable from the point of view of the flight duration. On the other side, best advantages of such systems are their simplicity and low price. There is an additional need for a maintenance unit for mixing the air and oil. The previous closed loop circulation lubrication system is revised by replacing only the sealing elements with simple cylindrical bushes in order to increase the flow surface between and around the bearings, while everything else is left the same, see Fig.3.

The compressor vacuum is used to create the air suction and air flow through the system. The previously mixed oil mist is introduced through nozzles in front of the bearing additionally.

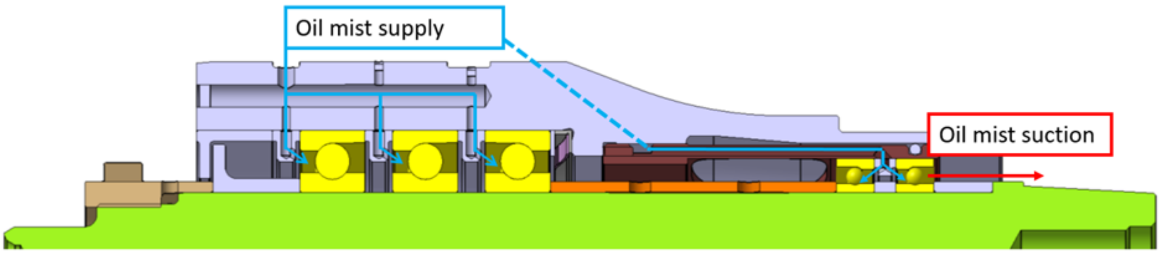

Fig. 3 – Oil mist lubrication method, oil mist supply is colored in blue and suction due to compressor vacuum is marked with red arrow

The analysis of the influence of air pressure and the amount of oil in the process of oil mist forming on the bearing operating temperature for open loop lubrication method is conducted in this paper. Also, the comparison of bearing temperatures with oil mist and oil circulation method was conducted.

3. Experiment setup

In order to determine optimal air pressure/oil amount ratio in the oil mist lubrication method, a series of tests was conducted. For that purpose, a custom-made test rig was developed.

The test rig is designed for testing several subsystems before installing the housing in the engine. For example, the same test rig is used for testing the lubrication of bearings and for rotor dynamics [20], couplings, bearings, electro starters and connections. The test assembly is placed on two supports which lock the housing in lateral and axial directions. The test setting was identical for both the lubrication method i.e. oil circulation and the oil mist method.

In the lubrication tests a Lehner 1950 electromotor is used for driving the system. The power voltage and amperage are measured with a supply unit in order to approximately determine the power consumption which is quite a good indicator of friction resistance which occurs in the bearings. Power increase can also indicate if there is too much oil in the system i.e. if there is oversaturation of oil. Excessive amount of lubricant creates rotational resistance and leads to even higher bearing operating temperature. Another reason why the temperature may rise is the existence of pockets in the system where oil remains trapped and consequently increases the temperature in the system.

In order to carry torque from the electromotor to the shaft, a R+W Miniature bellows coupling was used. This elastic coupling can damp the vibration and can calm the dynamic behavior of the shaft in the system. All elements (bearing housing assembly, coupling and electromotor) are connected with two carrier parts, shown in Fig. 5, connected with flange and bolts.

Thermocouple probes K types, wires CH+ AL-, from OMEGA supplier for temperature measurement were used. Their temperature range is from -200 up to 1,250°C with maximum error of ± 2.2 °C. Three thermocouple probes are places inside the bearing housing right next to the outside rings for each bearing of TFT set at fixed support, see Fig. 5.

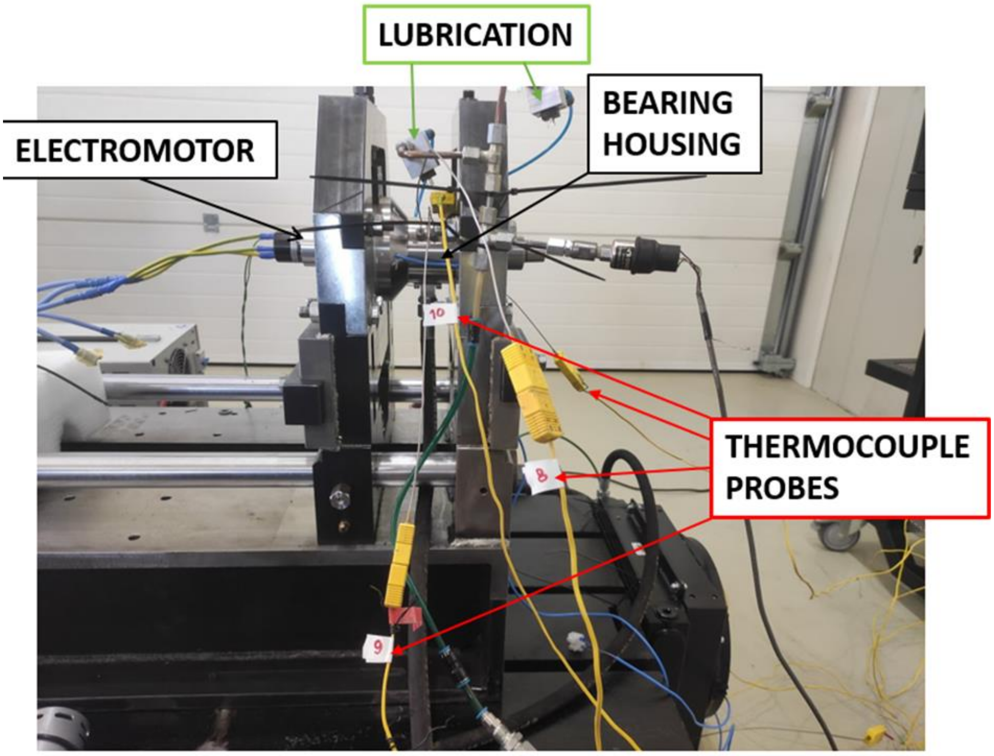

Fig. 4 – Test stand for bearing housing lubrication experimental investigation

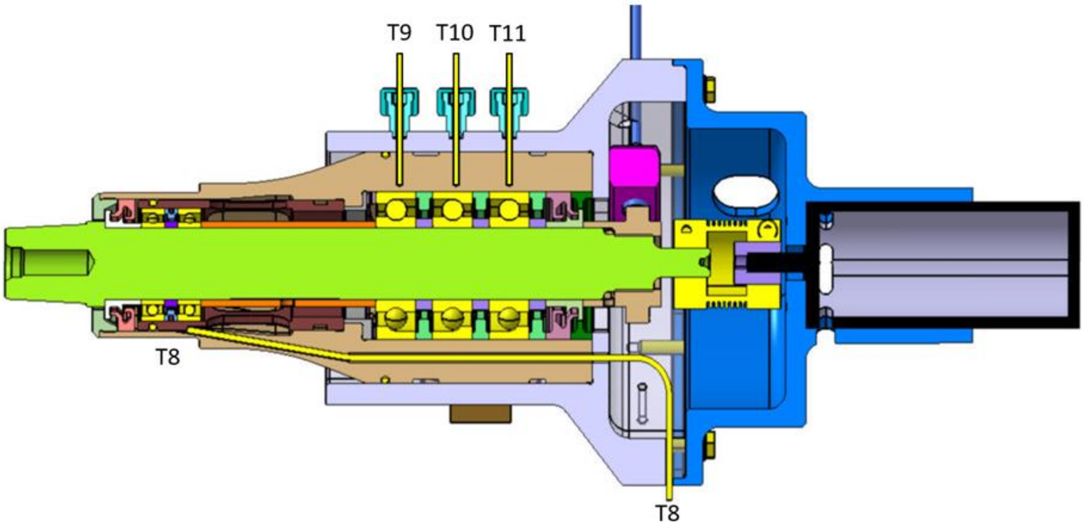

Fig. 5 – Four thermocouple probes and their install points for measure of the temperature for each bearing from TFT set of bearings in fixed support and DB set of elastic sliding support bearing set

One temperature probe is used to measure the temperature behavior of sliding support bearings, Fig. 5. It is installed through the elastic support, known as the squirrel cage support [21]. The position of this probe is adjusted to come as close to the outside ring of one of the bearings in the axial free (sliding) support.

Another two thermocouples were installed on the oil supply and drainage installation outside of the test article in order to measure input and output oil temperature for the oil circulation lubrication method.

So, as the output of the experiment, there were 4 measured temperatures – 3 temperatures for each bearing of the stiff support, named as T9, T10, T11 and 1 temperature used for both bearings of the elastic support, named T8.

Speed, i.e. rpm is measured by a non-contact inductive sensor DW-AD-405-04-290 of Contrinex Company with frequency of measurement of 10 kHz. It is placed radially with 0.4mm of clearance along the width of a specially designed nut with four teeth, see Fig.6.

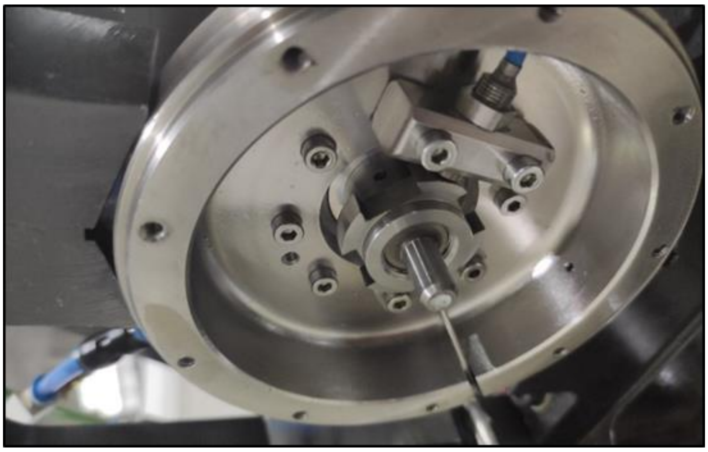

Fig. 6 – Toothed nut installed on the shaft and RPM probe installed on the housing flange

The pressure intensity, as well as the oil amount, were adjusted manually by means of "Metabo" oil mist maintenance unit, Fig. 7. The oil is introduced through the tip of the device in means of droplets, so the oil amount – flow, is measured by number of drops in second.

Fig. 7: Oil mist maintenance unit

All other test instruments like pumps, flow meters, pressure sensors and etc. were present during all tests. As they are not the aim of this paper there is no need for their detailed analysis and device description.

4. Results and analysis

The first tests were conducted with the oil circulation lubrication method, which is already designed, tested and adopted by the company Ederpo for their turboshaft engines. During these tests, main goal was to achieve operating temperature of the bearings below 80°C at the maximum allowed speed of the engine, 62,000 rpm. As lower engine operation speed generates less power and heat there was not a need for monitoring the lower regimes. Oil pumps, push and sunction, are driven to certain level and maintained constant during engine running. Consequently, at lower regimes temperature would be lower compared to maximum engine rpm as oil mass flow is kept constant. Same principle of managing the system is used for testing the bearing housing and lubrication method shown in this paper. Similar results of temperature changes were obtained at the measured locations, so one representative test and its results are presented in Fig. 8 and 9.

The test was conducted on maximum operating rotational speed of 62,000 rpm of the rotor for 900 seconds in order to achieve steady state of the bearing temperature as a result of heat balance. The results of this test were used for comparison with the results of the open loop system.

Pressure in oil system (up to 4.7 bar) and oil mass flow on TFT set (10.2 g/s) and DB set (5.6 g/s) during the test are presented in Fig. 8.

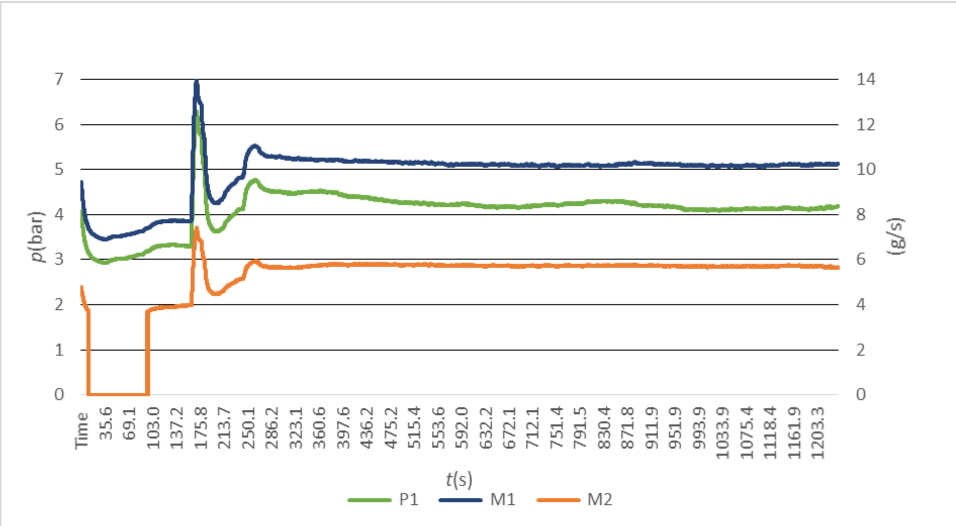

Fig. 8: Pressure in oil system (green line), and mass flow of oils on TFT set (blue line) and DB set (orange line)

The temperatures of the bearings, listed in Fig. 5 and 6, are shown in Fig. 9. The maximum operating temperature of 72°C was temperature T8 (combined elastic support bearings). As these bearings were with a brass cage it was as expected to have the highest temperature in the bearing system.

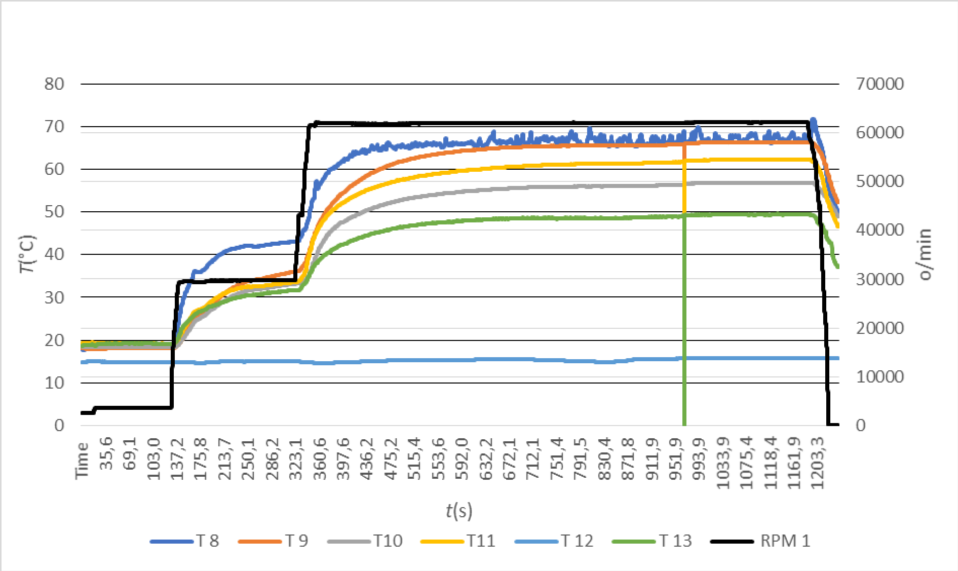

Fig. 9: Temperature of DB set T8, temperatures of TFT bearing set (according to Fig. 2) T9, T10, T11, supply oil temperature (after the cooler) T12, drainage oil temperature T13 and RPM value during the test.

These figures show only transient processes during the speeding up system to a desired operating point, i.e. maximum rpm of engine. This acceleration was done manually with very rough potentiometer so there some sudden changes, as at the 175 second in the Fig. 8, so this is the error by operator. As he produce very sudden increase in pump rpm all parameters follow that rise, and it was noticed in a few seconds same reaction was in order to lower it at the acceptable level.

On the other hand, as non-contact specially design centrifugal seals for preventing the oil leakage from the bearing housing have efficiency only above some speeds of revolution, process of speeding up the system is done in two stages. First stage implies the starting up the oil push pump to mass flow of a 4g/s for one and 7g/s for other one bearing set. This is a level where suction pump can achieve non leakage situation. Than electromotor is driven up to above 30,000 rpm, Fig. 9, where seals has enough centrifugal force to achieve sealing function. In second stage a system is driven up to maximum rpm and hold for a few minutes in order to achieve steady state of all parameters and heat balance.

The second set of tests were conducted with the open loop method, i.e. the oil mist method. A total of 12 tests were conducted, i.e. 12 combinations of air pressures and oil amounts were experimentaly carried out, until satisfying results were reached. During these tests a combination of air pressure of 1, 2, 3 and 4 bar was varied with oil amounts of 0.5, 1 and 2 drop/s. These tests were performed in 3 sets of 4 tests, by retaining the oil amount on one of previously mentioned 3 levels while varying the air pressure in the system. The temperatures of each bearing during the test can be seen in Fig. 10, 11, 12 and 13. Temperature T8 is the temperature of the elastic support (combined temperature for both bearings), while T9, T10 and T11 are temperatures of the stiff support bearings (according to Fig. 2).

Analyzing the diagrams, it could be seen that all bearings are having the best performance (lowest temperatures) when being lubricated by the oil mist with 4 bar of air pressure (the highest applied pressure during these tests) and 0.5 drop/s of oil (the lowest applied amount of oil). This could be explained with the fact that bearings are cooled by air and the oil only lubricates them. If the amount of oil is too big, it would increase the resistance to motion between the balls and raceways, increasing the operating temperature. Only these lubrication parameters in the oil mist method gave better results than the closed lubrication system. Thus, it was shown that an open loop lubrication system can have similar thermal behavior as a closed loop system, which was the goal of this work. More tests with further combinations of higher pressure of air in the system and lower amounts of oil were not carried due to the fact that lower temperatures of bearings, below optimal range of 60-80°C, could have an influence on its clearances, interferences and operating conditions which can produce some negative effects in this case.

For an identical pressure value, the technology of oil/air lubrication that contains an extreme-pressure and anti-wear additive improved the operational ability of a helicopter transmission system that is out of oil [11]. It consumed the least quantity of oil and produced the least wear width, the least rise in temperature and the best surface wear quality if the lubrication system of a helicopter reducer is compromised, i.e. its gears and bearings are at a non-lubricating oil work state [11]. The results of this test show that increasing air pressure can affect the minimization of the temperature rise in bearings. Although increasing air pressure contributes to the reduction in bearing temperature, care must be taken because saturation might appear after the pressure level reached a certain extent [20].

In general, the highest temperature of 66°C is observed on the first bearing of the stiff support, but the bearings of the elastic support are just a little bit less heated and have a temperature of 65°C. The second and third bearings of the stiff support are usually notably colder than the first stiff and elastic support bearings, with temperatures of 60°C and 64°C respectively.

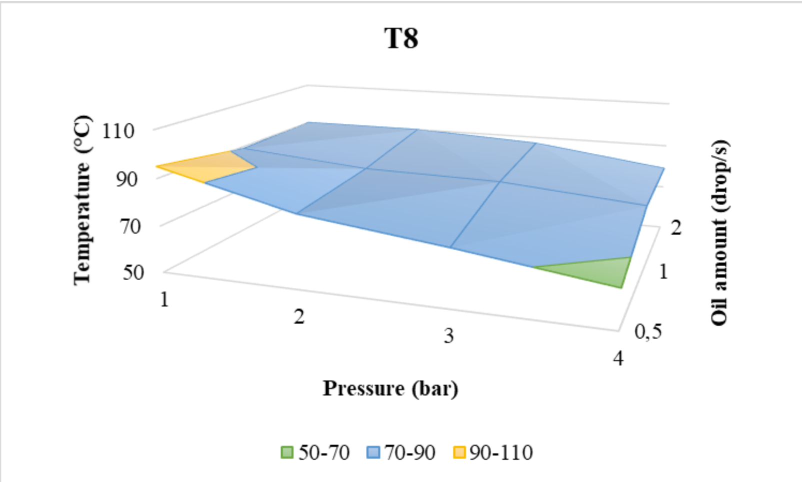

Fig. 10. Combined temperature graph of both bearings of the elastic support with respect to air pressure and oil amount

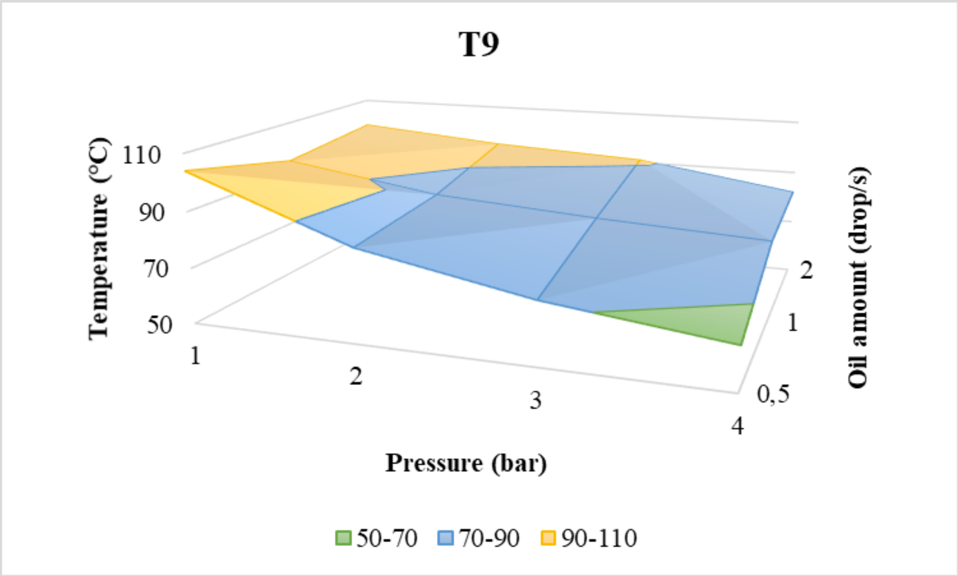

Fig. 11. Temperature graph of the first bearing of the stiff support with respect to air pressure and oil amount

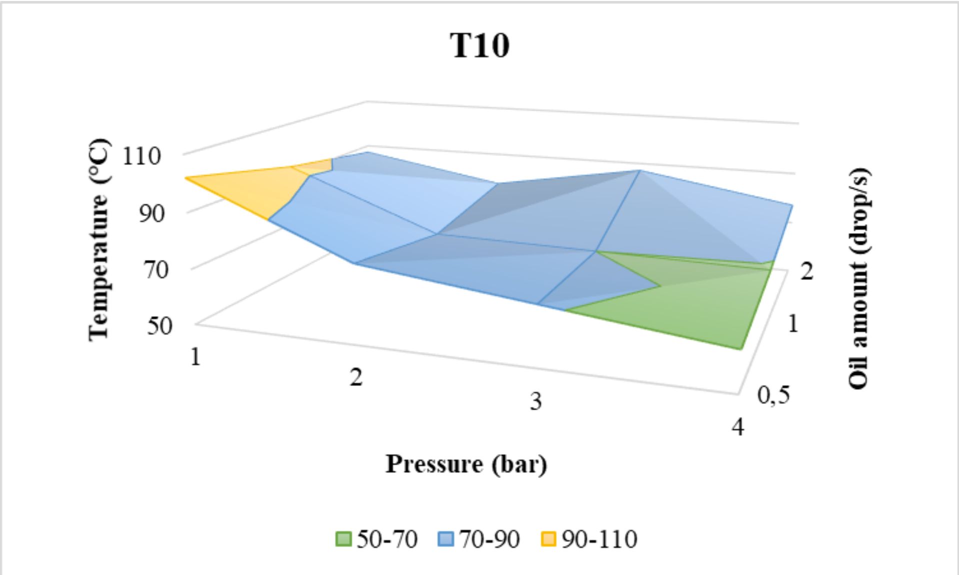

Fig. 12. Temperature graph of the second bearing of the stiff support with respect to air pressure and oil amount

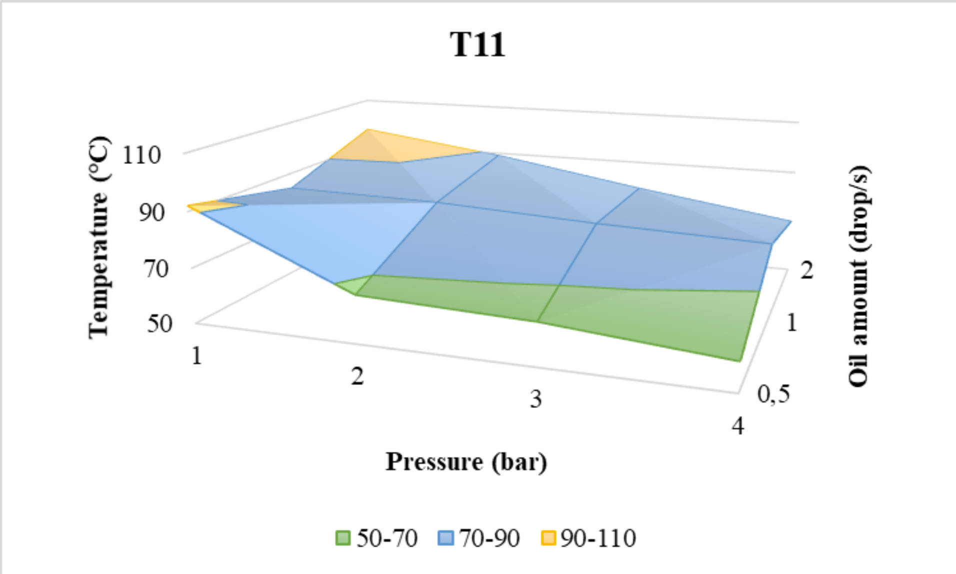

Fig. 13. Temperature graph of the third bearing of the stiff support with respect to air pressure and oil amount

In order to make it easier for the reader to analyze the diagrams, the exact values for the maximal temperatures of each bearing in every working regime is given in Tab. 1.

Tab. 1: Tabular representation of the results

Air pressure [bar]	Oil amount [drop/s]	T8 [°C]	T9 [°C]	T10 [°C]	T11 [°C]
4	2	82	77	70	79
4	1	79	69	78	78
4	0,5	65	64	60	66
3	2	91	87	79	87
3	1	82	70	80	83
3	0,5	72	71	65	74
2	2	94	77	89	90
2	1	86	71	83	84
2	0,5	83	78	67	81
1	2	99	87	97	90
1	1	95	93	84	89
1	0,5	104	102	92	95

With iterative variations of the oil mist parameters, good results in thermal behaviour of high speed bearings could be obtained for the open loop system. For example, 4 bar of air with 0.5 drop/s of oil resulted in temperatures of bearings very similar to the temperatures in the closed loop. The comparison of the bearing temperatures in the closed loop (oil circulation) lubrication method and open loop (oil mist) lubrication method is given in Tab. 2.

Tab. 2: Comparison of bearing temperatures in the closed loop and open loop lubrication systems with combination of 4 bar of air and 0.5 drop of oil per second

Temperature [°C]	Close loop	Open loop
T8	72	65
T9	66	64
T10	58	60
T11	63	66

Open loop lubrication has a much simpler design, which makes it cost effective and more reliable in operation, if compared to a closed loop. Even though certain amount of oil is irretrievably spent, the previously mentioned advantages make the open loop lubrication very competitive to the closed loop. The amount of spent oil is very low and it ends up in the combustion chamber, burning out together with the fuel.

5. Conclusion

In this paper the lubrication of the high-speed bearing assembly of the turboshaft engine TSE-200 Phoenix, a custom-made engine produced by EDePro company from Serbia, is analyzed. Two lubrication (closed loop and open loop) methods are tested and compared. It has been shown that with adequate lubrication parameters (air pressure and lubricant flow), in an open loop lubrication system, operating temperatures of the bearings can be comparable to, or even better i.e. lower than a closed system can achieve. In addition to thermal behavior, other advantages of the open loop lubrication method are:

- Much simpler design which makes production easier,
- Lower mass of the system since there is no need for the oil coolers,
- Better reliability due to absence of seals. In the closed loop system there are complex contactless seals with very narrow gaps between the sealing rotor and stator [23] which increases the possibility of failure.

Taking all these facts into account and comparing the results of the tests which showed that temperatures of the open loop bearings could be very similar to the closed loop bearings if oil mist parameters are adequately adjusted, it could be concluded that open loop lubrication can be integrated for the application in turboshaft engines, without significant changes in thermal behavior. But, further investigation on the influence of the open loop lubrication on bearing life should be considered.

The main drawback of the open loop lubrication method is wasting the oil and the requirement of pressurized air. Both these facts are compatible with turbojet and turboshaft engines for drone applications, as air could be supplied from the engine itself while the oil could be burned in the combustion chamber section. The amount of oil required to be carried and further spent for relatively small flight duration against the much simpler construction and lower mass justifies the application of open loop lubrication method for application in such engines.

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