



Thermal Management Techniques for Oil-Free Turbomachinery Systems

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Abstract

Tests were performed to evaluate three different methods of utilizing air to provide thermal management control for compliant journal foil air bearings. The effectiveness of the methods was based on bearing bulk temperature and axial thermal gradient reductions during air delivery. The first method utilized direct impingement of air on the inner surface of a hollow test journal during operation. The second, less indirect method achieved heat removal by blowing air inside the test journal to simulate air flowing axially through a hollow, rotating shaft. The third method emulated the most common approach to removing heat by forcing air axially through the bearing's support structure. Internal bearing temperatures were measured with three, type K thermocouples embedded in the bearing that measured general internal temperatures and axial thermal gradients. Testing was performed in a 1 atm, 260 °C ambient environment with the bearing operating at 60 krpm and supporting a load of 222 N. Air volumetric flows of 0.06, 0.11, and 0.17 m³/min at approximately 150 to 200 °C were used.

The tests indicate that all three methods provide thermal management but at different levels of effectiveness. Axial cooling of the bearing support structure had a greater effect on bulk temperature for each air flow and demonstrated that the thermal gradients could be influenced by the directionality of the air flow. Direct air impingement on the journal's inside surface provided uniform reductions in both bulk temperature and thermal gradients. Similar to the direct method, indirect journal cooling had a uniform cooling effect on both bulk temperatures and thermal gradients but was the least effective of the three methods.

Introduction

The first commercial use of foil air bearings dates back to the early 1970's in air cycle machines that provided cabin pressurization in aircraft (ref. 1). This relatively benign application was a perfect fit for these rudimentary bearings since loads and temperatures were low. Now, with the advent of more advanced, higher load capacity designs (ref. 2) and high temperature solid lubricant coatings (ref. 3), foil bearings are becoming increasingly targeted as replacements for conventional, oil-lubricated bearings in current and future high speed turbomachinery systems, as evidenced by the commercially available Oil-Free microturbine generators and turbocompressors (refs. 4 and 5). Additionally, the world's first foil bearing supported onboard auxiliary power unit (APU) for small aircraft is currently undergoing field

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testing by a major aerospace company to assess its performance and viability (ref. 6). It is expected that these successes will provide the motivation to push foil bearing technology into higher performing systems, specifically small gas turbine engines in the 4.5 to 13.3 kN thrust class.

Foil bearings are a type of hydrodynamic fluid-film bearing that operate on a very thin gas film ($\sim 10^{-4}$ m) generated by the natural pumping action of the rotating shaft that drags fluid into a convergent wedge-shaped gap. The foil bearing designs currently available employ either a smooth, continuous top foil or overlapping leaf foils resting on an elastic support structure. The top foil's purpose is to form and contain the hydrodynamic pressure and the elastic foundation provides a compliant substructure that permits local deflection of the top foil in response to pressure changes. This compliance allows foil bearings to accommodate significant centrifugal and thermal growth of the shaft, high levels of misalignment and an improved tolerance to dirt and dust contamination.

A major advantage to foil air bearings is their ability to perform at temperatures that are unsuitable for most other types of fluid lubricated bearings. In turbomachinery systems, for example, the conventional rolling element bearings supporting the rotating components operate over a fairly narrow temperature range mainly due to the temperature limitations of the required liquid lubricant. These liquid lubricants are specially formulated for this application but can only withstand approximately 200 to 250 °C before they begin to thermally oxidize and lose their lubricating properties. As a result, a complex, closed-loop, scavenging system consisting of pumps, sensors, radiators, tubing, filters, etc. is added to the engine to manage the thermal stress on the oil and prevent oxidation. Unfortunately, this added hardware increases engine weight and maintenance costs.

Foil bearings, on the other hand, rely on air as their lubricant which is not degraded by extreme temperatures. Consequentially, incorporating foil bearings provides an immediate weight savings to a system of around 15 percent, and maintenance cost reduction of 50 percent since a closed loop support system is not needed (ref. 7). Higher temperature operation also gives gas turbine engine designers the freedom to place bearings in close proximity to high temperature components, such as the turbine, combustor or compressor. Proof of this capability can be found in the commercially available microturbine mentioned previously where the foil bearing located next to the turbine operates at a temperature near 540 °C.

The ability of foil bearings to operate at high temperatures does not mean they are impervious to the thermal environment. On the contrary, foil bearings have their own unique, thermal limitations that must be addressed to ensure bearing performance is not compromised. One such issue is the effects heating has on the material used in constructing the bump foils. Each individual bump acts as a small spring with the material's elastic modulus and geometry combining to define its spring rate. Collectively, the bumps form a compliant substructure that supports the top foil and defines the bearing's performance characteristics, such as load capacity, stiffness and damping. Current high temperature foil bearings are made from a nickel-based superalloy because it retains useful spring properties up to 650 °C. Even so heating still causes the superalloy to soften which lowers the elastic modulus and, consequentially, the performance levels of the bearing, such as the load capacity (ref. 8). At first, this behavior seems counterintuitive since the higher air viscosity due to heating should lead to increased bearing load capacity. At high loads, though, the air film becomes very thin and stiff, and the bearing's performance becomes dependent upon the elastic properties of the bump foils. Furthermore, there is evidence that this softening of the elastic foundation alters the bearing's stiffness and damping properties which can adversely affect the desired rotordynamic behavior of the system if not taken into account in the design stage (ref. 9).

Another area of concern with foil air bearings is the formation of internal thermal gradients. The compliant support structure is designed with spatially variable stiffness properties to retain hydrodynamic air pressure and limit side leakage. This creates a variable air film thickness within the bearing and also entraps a slug of air that undergoes constant viscous shearing (ref. 10). These combined conditions promote non-uniform viscous heating that will manifest into thermal gradients that, if allowed to grow, can lead to thermal distortion or warping of the top foil causing bearing failure. One notable case of catastrophic failure was described by Dykas who performed an analytical investigation on a test that resulted in complete destruction of the bearing and a dime-sized hole in the journal (ref. 11). Results from

his computer model of the bearing-journal system suggested that combined effects of journal growth due to centrifugal and thermal gradients were the major cause for the failure. Data in the open literature on acceptable levels for thermal gradients are scarce but recent tests in the author's lab on bearings of the type in reference 2 have shown that there is a risk of failure when the axial temperature gradient begins to surpass approximately 2.17 °C/mm. This limit, however, should only be regarded as an estimate since testing also suggests that other factors may play a role, such as the initial amount of radial clearance (preload) built into the bearing and the time rate of change of the gradient increase. Radial clearance is an important physical design parameter that sets the "slop" or additional space present in the bearing available for centrifugal and thermal expansion of the journal. A more detailed explanation of radial clearance and its effects on bearing performance can be found in reference 12.

Only through the implementation of an effective thermal management strategy can the excessive material temperatures and thermal gradients be controlled. Whereas current turbomachinery systems rely on the circulated lubrication oil to provide thermal management for rolling element bearings the most probable scenario for foil bearings will consist of a dedicated, open loop system that uses cooling air from the compressor to remove excess heat and ensure a thermally stable bearing. The term "cooling air" used in conjunction with foil air bearings is somewhat of a misnomer since its function is not to radically cool the bearings but to merely retain bulk temperature or thermal gradients within acceptable design limits. In fact, in a gas turbine engine, compressor bleed air is far from being cool. For instance, air bled from the low pressure stage can be around 150 °C (300 °F) whereas air from the high pressure stage can be over 343 °C (650 °F). Once heat is extracted from the bearing the air may be returned to the main flow or vented to the atmosphere. Since bleed air reduces the engine's power output, it is important to ascertain the optimum flow rates that provide adequate thermal management control. Air mass flow will have to be matched to the heat loads emanating mainly from two sources. The first contributor is the energy absorbed, either by conduction or radiation, from the surrounding environment. This can be due to the bearing's relative proximity to the turbine, combustor or the compressor. The second source is the heat that is self-generated through viscous shearing in the thin air film. Tests in the lab indicate that, depending upon speed and load, the power loss can be on the order of a few hundred watts. According to reference 13 it is estimated that 80 percent of this heat is conducted away via the shaft and bearing surface, which can increase bearing internal temperatures 200 °C higher than the surrounding environment (ref. 14).

The purpose of this paper is to report on the effectiveness of three different techniques to provide thermal management to journal foil air bearings. The study was conducted on a foil bearing instrumented with three, type K thermocouples embedded in the bearing. The candidate techniques include direct and indirect cooling of the journal and cooling of the bearing via axial air flow through the bump foil support structure. Tests were performed in an ambient environment of 260 °C with the bearing operating at 60 krpm and supporting a radial load of 222N. Simulated compressor bleed air flows of 0.06, 0.11 and 0.17 m³/min at approximately 150-200 °C were used. The effectiveness of the three techniques was based on a decrease in the bearing's internal bulk temperatures and residual thermal gradients.

Experimental Apparatus/Procedure

Instrumented Foil Bearing and Journal

Shown in figure 1 is a drawing of the Generation III foil bearing (ref. 15) used in the tests along with the location of the installed thermocouples. Generation III is a term that signifies the design complexity of the bearing's support structure and is further explained in more detail in reference 16. The bearing was nominally 50 mm in diameter and 48 mm in length. The internal temperatures were measured by three type K thermocouples in 0.81mm fiberglass sheaths placed on the foil spacer block as shown, with one in the middle and one at each edge. This axial arrangement measures the thermal gradient that acts across the half-width of the bearing, as reported in reference 14. Placing the thermocouples on the spacer block also provided the least intrusive method to monitor the bearing's internal temperatures.

The mating test journal was prepared by first plasma spraying a high temperature solid lubricant coating, PS304, onto the surface and then heat treating it in a 540 °C oven for 100 hours (ref. 17). After the heat treatment process the journal was affixed to the test rig and ground with an in-place grinding system to a final diameter of approximately 50.8 mm in order to achieve the manufacturer's specified bearing preload. Afterward the rig was dynamically balanced up to 60krpm.

Direct and Indirect Shaft Cooling

Two thermal management methods utilizing journal cooling were studied, direct and indirect. These techniques are based on the concept of removing heat from the bearing by encouraging the energy to diffuse by conduction through the fluid film and into the journal where it is then carried away by convection at the journal's inner surface. An added benefit to this method is that it limits the journal's circumferential growth due to thermal expansion that can expend the available radial clearance in the bearing. To simulate compressor bleed air temperatures consistent with an engine, the air used by these two methods was heated to approximately 150 to 200 °C by an electric tube heater placed upstream of the guide tube. A thermocouple placed at the end of the tube measured the exiting air's temperature.

Direct cooling is akin to the jet impingement technique in gas turbine engines that cools turbine blades by increasing the local forced convection coefficient (ref. 18). This set-up is shown in figure 2 where a 9.5 mm inner diameter guide tube delivers the preheated air to the journal's inner surface. The distance between the tube's exit and journal surface was arbitrarily set at 8 mm which may not be optimal for producing the highest convection coefficient possible. The location of impinging air coincided with the high pressure region of the bearing that is supporting the applied load.

The indirect method, shown in figure 3, is an attempt to simulate a thermal management strategy of flowing air axially through a hollow shaft in order to increase the forced convection coefficient to remove heat from the bearing. The straight guide tube was directed at the backend of the journal where the air would impinge and then travel axially out of the journal. A more realistic approach to evaluating this method would have been to conduct the test with air traveling axially through a hollow journal but this arrangement was not possible with the current test facilities at hand. A similar attempt of this method was described in reference 19 where fins were brazed to the ID of a rotating shaft to remove bearing friction losses by improving the conduction heat transfer through the shaft.

Axial Bearing Cooling

The setup for this method is shown in figure 4 and reconstructs the more common approach of cooling a bearing by flowing air axially through the bearing's support structure. The bearing sits inside a "can" that is enclosed at one end of the journal and open at the other. A supply tube connected to the can's closed end delivers the air and the sealing capability of the hydrodynamic air film forces most of the cooling air to travel axially through the bearing's support structure. The electric tube heater was not used for this technique as the ambient environment in the can due to the furnace was sufficient to heat the incoming air. A thermocouple placed in the can approximates the cooling air temperature.

Test Apparatus/Procedure

The high-speed rig used for performing the tests is shown in figure 5 and is described in greater detail in reference 8. The rig consists of a drive shaft that is supported by two, hybrid ceramic, oil lubricated ball bearings. An air impulse turbine attached to the main shaft is capable of driving the journal to a maximum rotational speed of 70krpm. Radial loading of the foil bearing is accomplished via a vertical cable system with one end attached to the bearing in a stirrup configuration and the other to a pneumatic load cylinder mounted below the test rig. The rod extending from the bearing loader acts as a moment arm to relay the bearing torque to a load cell. The furnace consists of two halves that surround the bearing for heating and also improve operational safety by acting as a scatter shield.

The PS304 coating in an as-ground condition has a surface roughness of $0.8\text{ }\mu\text{m}$ and will undergo break-in cycles during initial operation. To ensure that surface roughness does not influence the data the bearing and journal system were subjected to 1700 start-stop cycles at approximately $375\text{ }^{\circ}\text{C}$. The start-stop cycles act as a polishing wear mechanism between the asperities of the top foil and coating, resulting in a smooth, glossy, oxide-rich surface on the coating and the development of a solid lubricant layer on the foil surface (ref. 14). After the break-in cycles the final coating surface roughness was approximately $0.2\text{ }\mu\text{m}$.

Each test commenced by supplying power to the furnace heaters and accelerating the bearing up to 60krpm. Once the desired furnace temperature was reached a 222 N load was applied to the bearing. When bearing temperatures reached steady-state the cooling air at the lowest flow rate was introduced. Bearing temperatures were allowed to reach steady-state before each step increase in the cooling air flow rate. Manual adjustment of the electric tube heater was required at each flow rate to maintain air in the 150 to $200\text{ }^{\circ}\text{C}$ range to simulate compressor bleed air. Temperature measurements for the three thermocouples were continually collected via a computer data acquisition program and stored in a spreadsheet file.

Discussion of Results

Journal Cooling

Results for the direct and indirect journal cooling methods are shown in figures 6 through 8, respectively. Each method's pre-test bearing temperatures are identical in both magnitude and distribution thereby allowing for direct comparisons. The temperature distributions, with the highest temperature in the middle and the lower temperatures near the edges, correlate well with the results reported in reference 14. The maximum axial thermal gradient is approximately $2.29\text{ }^{\circ}\text{C}/\text{mm}$. and acts across the half-width of the bearing, between the middle and rear thermocouples.

Before beginning the direct cooling test the baseline bearing temperatures were $345\text{ }^{\circ}\text{C}$ at the front edge, $386\text{ }^{\circ}\text{C}$ at the middle and $331\text{ }^{\circ}\text{C}$ at the rear edge. Introducing $203\text{ }^{\circ}\text{C}$ cooling air at the lowest flow rate of $0.06\text{ m}^3/\text{min}$ caused the bearing's internal temperatures to begin decreasing almost immediately, indicating a quick, dramatic rise in the local convection coefficient. Upon reaching steady-state the front, middle and rear temperatures dropped by approximately 9 percent. The thermal gradient followed suit as it decreased by 11 percent. With the cooling air at $188\text{ }^{\circ}\text{C}$ and $0.11\text{ m}^3/\text{min}$ the front baseline temperature was reduced by 17 percent, the middle, 19 percent and the rear, 18 percent. The thermal gradient's drop was 25 percent. At the maximum flow rate of $0.17\text{ m}^3/\text{min}$ and the air at $189\text{ }^{\circ}\text{C}$ there was a further decline in the bearing's internal temperatures but at a lesser extent. The net effect of this flow rate decreased the front baseline temperature by 21 percent, the middle, 23 and 19 percent for the rear. The thermal gradient was the most affected as it dropped by 44 percent. Overall, this method produced a near uniform cooling effect on the bearing's internal temperatures and was also effective at controlling the thermal gradient.

The above results were obtained with the guide tube pointing towards the high pressure region of the bearing. Since circumferential location of the radial load cannot be expected to remain fixed during operation a test was conducted to evaluate if angular orientation of the guide tube relative to the high pressure region would alter the technique's effectiveness. To answer this question the direct cooling test was repeated but with the guide tube rotated approximately 180° from the high pressure region. The result for this test is shown in figure 7. Comparison of figures 6 and 7 indicate that the technique's effectiveness is not a strong function of the guide tube's orientation.

The result for the indirect journal cooling method is plotted in figure 8. At the beginning of the test the front edge baseline temperature was $342\text{ }^{\circ}\text{C}$, the middle was $384\text{ }^{\circ}\text{C}$ and the rear edge was $329\text{ }^{\circ}\text{C}$. The thermal gradient was again $2.29\text{ }^{\circ}\text{C}/\text{mm}$. Introducing cooling air at $0.06\text{ m}^3/\text{min}$ and $199\text{ }^{\circ}\text{C}$ dropped the front, middle and rear temperatures by 5 percent. The response to the cooling air was slower than the

direct method indicating a less dramatic rise in the local convection coefficient. The thermal gradient only fell by 2 percent. At 0.11 m³/min the cooling air was 183 °C and it had almost an insignificant effect on bearing temperature as all three baseline temperatures experienced a small additional decrease over the previous flow rate of approximately 2 percent. The thermal gradient fell another 5 percent. Only when the cooling air flow at 184 °C was increased to 0.17 m³/min was there a noticeable change in both temperature and gradient, as the net effect of this flow rate decreased the front baseline temperature by 14 percent, the middle, 13 percent, the rear, 13 percent and the thermal gradient, 17 percent. By comparison this method is inferior to the direct method but it does produce some tangible thermal management benefits.

The effectiveness of the direct method over the indirect method is most likely based on its ability to focus the entire mass of the impinging air jet at one location of the thin, turbulent boundary layer that exists over the inner surface of the journal. The focused jet promotes the mixing of fresh, cooler air with the hotter boundary layer thereby increasing the localized convection coefficient. Based on the temperature results the mixing zone probably extends the entire axial length of the bearing. The indirect method, on the other hand, distributes the mass of the air jet over a wider area thereby lessening its impact. There is some question as to whether the increase in the convection coefficient of the direct method is due to the impinging jet penetrating the boundary layer sufficiently deep to reach the journal, as described in reference 20, but it is unclear if this is occurring and is beyond the scope of this research.

Axial Cooling

While conducting a preliminary test of this method with the bearing operating at 60krpm and in an unloaded condition its temperatures rose beyond 427 °C and threatened to reach the thermocouple sheath temperature limit of 482 °C. The bearing also began to demonstrate definitive signs of failure (rising torque, falling speed) due to a high thermal gradient over 3.00 °C/mm which lead to termination of the test. Review of the post test data identified containment of the bearing as the cause for the excessive temperature and thermal gradient. During operation the hot side leakage air, which would normally be expelled into the environment, became trapped in the closed end where it was circulated back into the bearing and further heated due to viscous shear. This continuous cycle of circulation in and out of the bearing produced the high temperatures and the excessive thermal gradient between the hotter front edge and relatively cooler middle of the bearing. The problem was rectified by supplying an initial 0.04 m³/min of air to the bearing. As shown in figure 9 this produced a more uniform axial temperature distribution, resulting in temperatures of 408, 438, and 423 °C for the front, middle and rear locations, respectively, and a more manageable thermal gradient between the front and middle of 1.25 °C/mm. This alone is a notable achievement since it demonstrates that utilizing proper thermal management can sustain bearing operation in an environment that would otherwise be unfeasible, such as in a gas turbine engine where the bearing's proximity to the combustor or turbine may cause a similar thermal non-uniformity.

Increasing the cooling air flow to 0.06 m³/min had an immediate effect on the bearing's internal temperatures. The front baseline temperature, which is nearest to the cooling air entry, registered the greatest change, dropping 36 percent, followed by the middle at 30 percent and finally the rear which fell by 26 percent. The cooling air temperature was 261 °C. The dramatic reduction in the front temperature had an adverse effect on the thermal gradient as it increased by 46 percent. Directionality of the cooling flow also caused the location of the maximum temperature to shift from the middle to the rear of the bearing. With the cooling air at 0.11 m³/min and 181 °C the baseline temperatures were almost uniformly halved with the front falling by 57 percent, the middle 52 percent and the rear 47 percent. The thermal gradient also fell by 20 percent but is still greater than the pre-test magnitude of 1.25 °C/mm. At this flow rate the front thermocouple was reading a temperature lower than the entering air. This may be a testing artifact caused by a stagnant air pocket in the corner of the can where the cooling air thermocouple resides. If so, the front temperature was probably more indicative of the temperature of the air flowing through the bearing. Increasing the flow rate to 0.17 m³/min caused a 66, 62, and 57 percent drop in the

front, middle and rear baseline temperatures, respectively. As indicated by the front thermocouple the cooling air temperature of 139 °C was about 45 °C less than what was supplied during testing of the other methods at this same flow rate. If this difference in cooling air temperature is taken into account an overall temperature decrease of over 45 percent in the bearing is achieved which still makes it the most effective of the three methods.

With regards to thermal gradients the test results suggest that a low air flow can be effective, especially if bearing temperatures can remain high. A low air flow should travel in the direction of decreasing thermal gradient. This will help to evenly distribute the heat in the bearing resulting in a more uniform temperature profile. This method also demonstrated that it has the potential to exacerbate thermal gradients if an improper flow rate is used. It is expected that large flow rates can overcome the directionality issues and will control both thermal gradients and bearing temperatures but at the expense of decreasing the efficiency of the system.

Conclusion

Three different methods were tested to evaluate their effectiveness at providing thermal management for journal foil air bearings. The axial cooling method demonstrated the most effective means at lowering bearing internal temperatures and it provided thermal gradient control but the latter is dependent upon directionality and magnitude of the cooling air flow which may not be achievable in practice due to engine cavity pressures. Direct cooling of the journal reduced both bearing internal temperatures and thermal gradients and was not a strong function of guide tube orientation. Indirect journal cooling did provide some degree of thermal management but was only capable of a minor cooling effect. Bearing internal temperatures were reduced but control of thermal gradients was limited. Directionality of the cooling flow may also be an issue with this method but was not studied.

Future thermal management requirements for Oil-Free turbomachinery systems will most likely focus on controlling thermal gradients and/or minimizing internal temperatures to retain proper bearing characteristics (system rotordynamics) and performance. Only by having a full understanding of both the static and transient thermal environment during bearing operation, either from experimentation or modeling, can the appropriate cooling methodology be implemented into the system.

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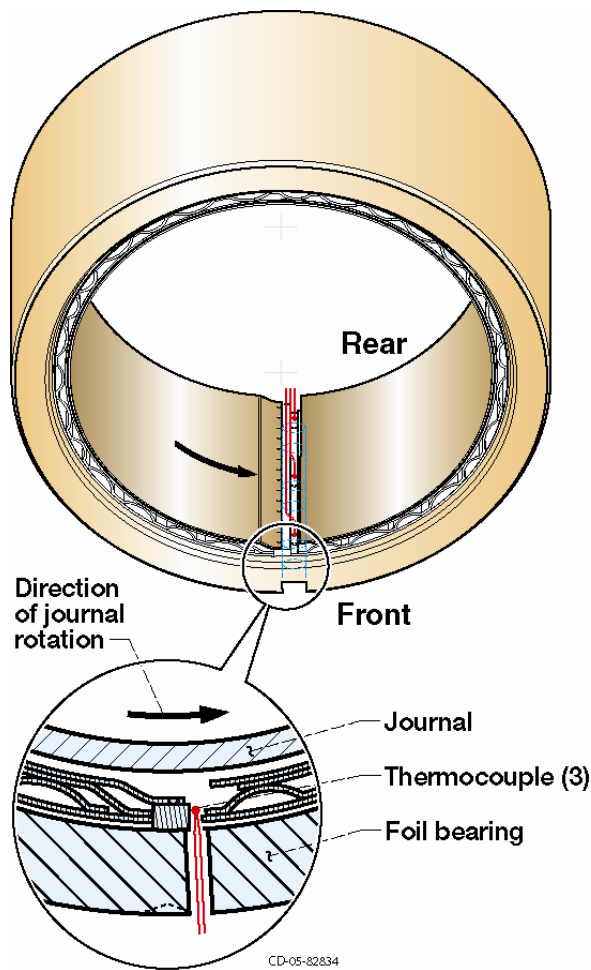


Figure 1.—Generation III journal foil air bearing with three thermocouples placed axially on spacer block.

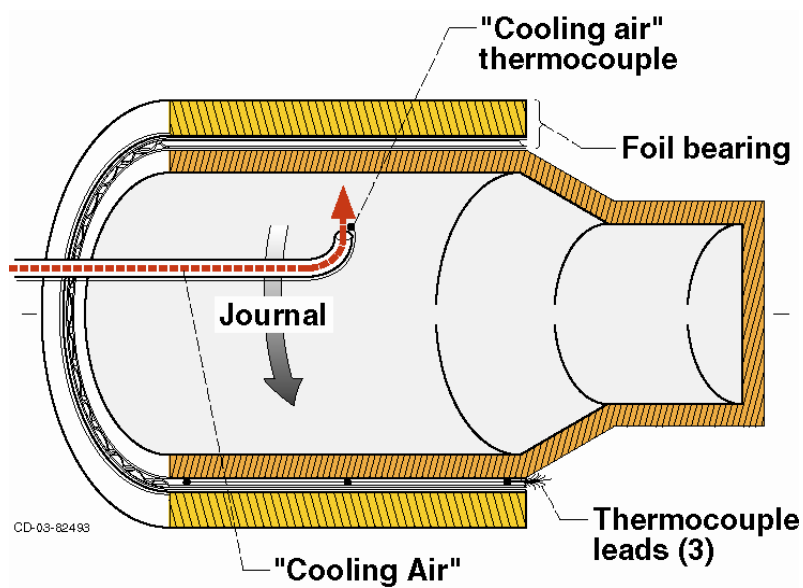


Figure 2.—Direct cooling method with guide tube oriented towards journal ID surface.

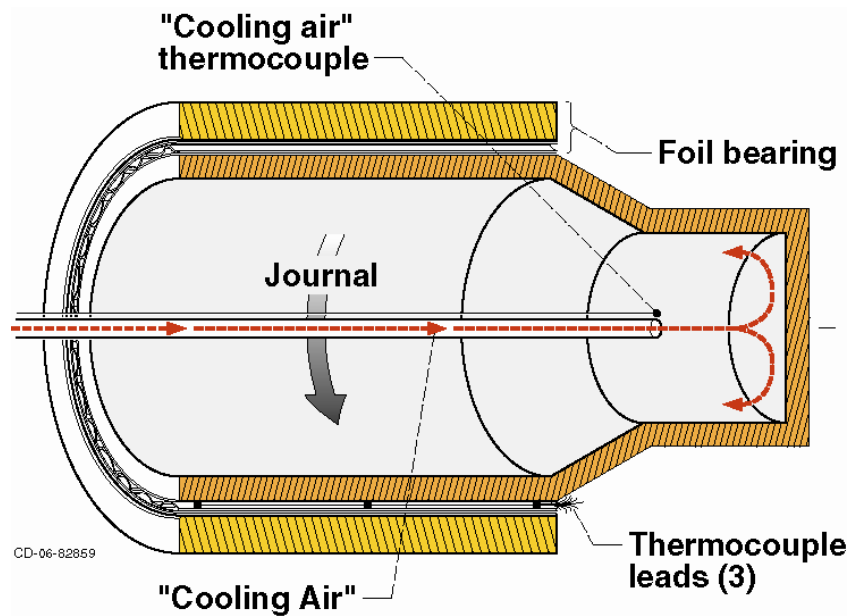


Figure 3.—Indirect cooling method with guide tube oriented towards the back face of journal

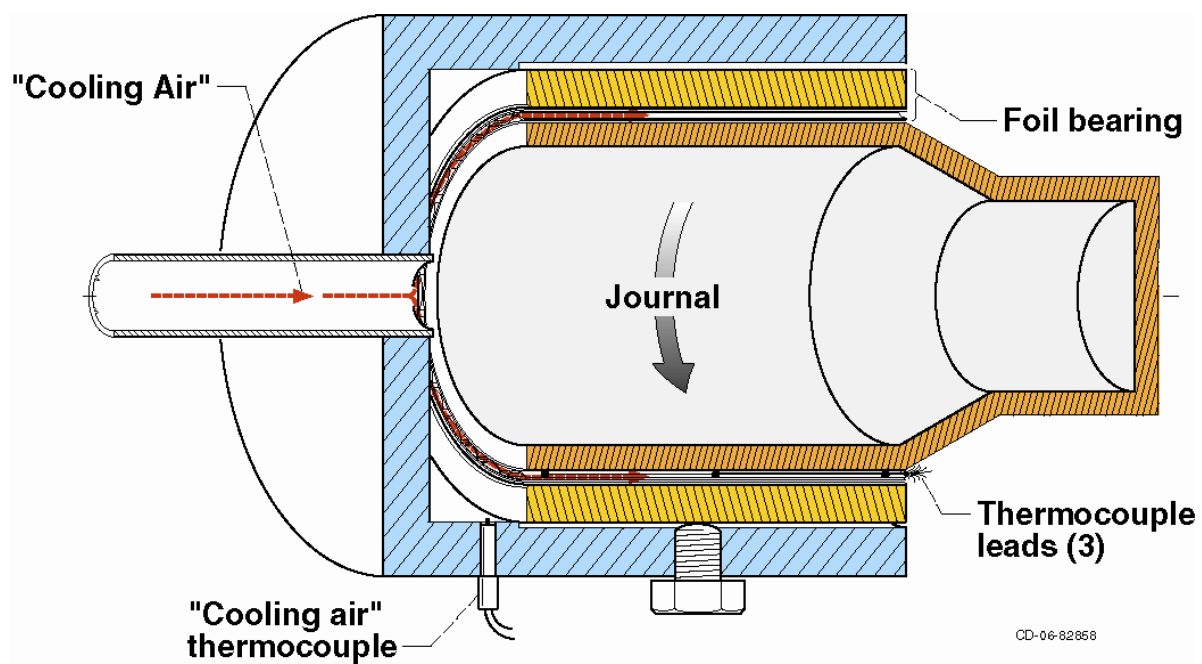


Figure 4.—Axial cooling method with bearing inside can. Air flow is through bearing support structure

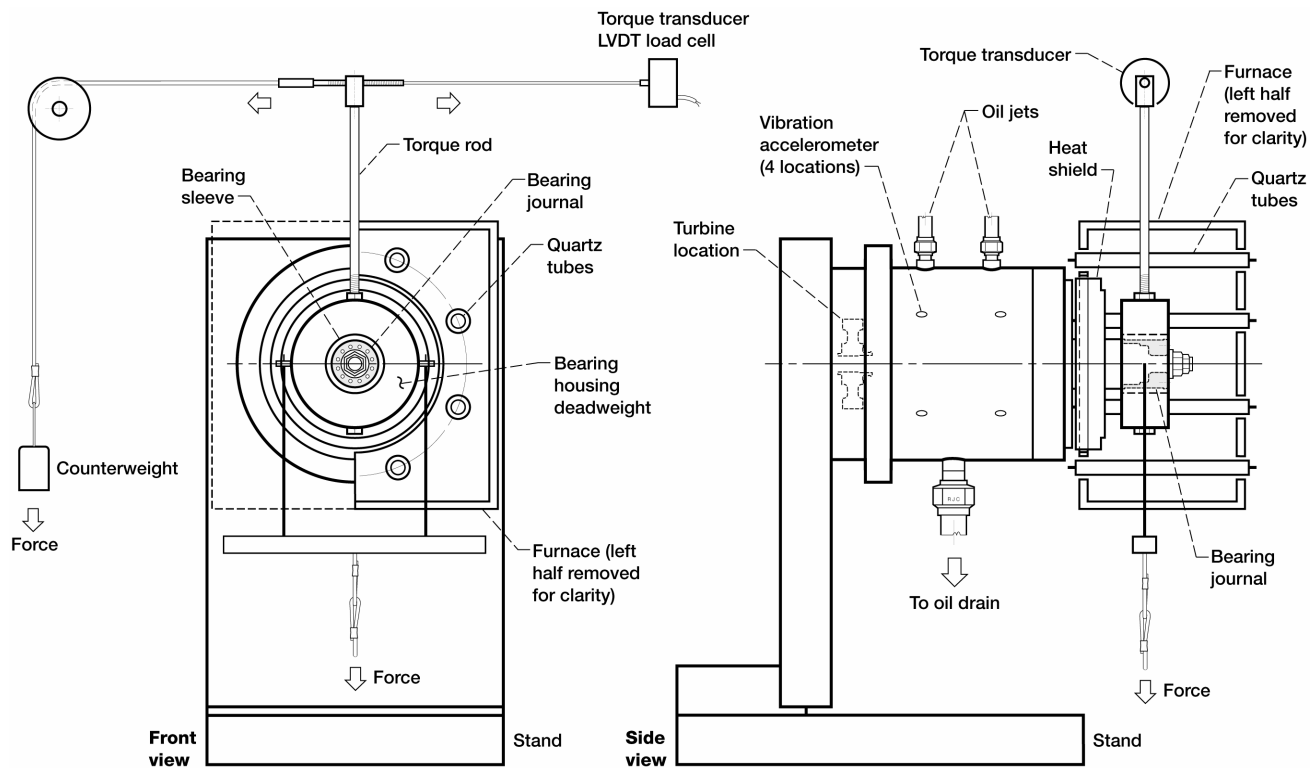


Figure 5.—High-temperature, high-speed journal foil bearing test rig.

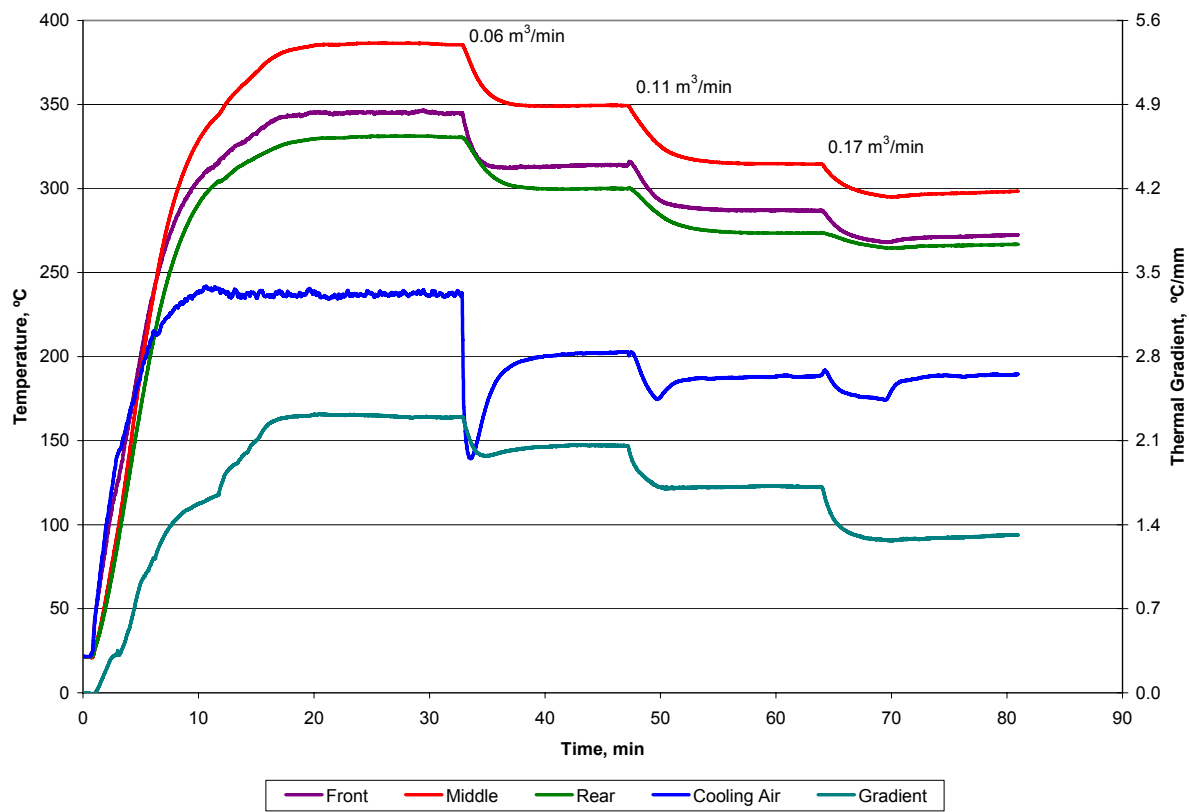


Figure 6.—Test results of direct cooling method with 0.06, 0.11 and 0.17 m³/min cooling air. Guide tube oriented toward high pressure region.

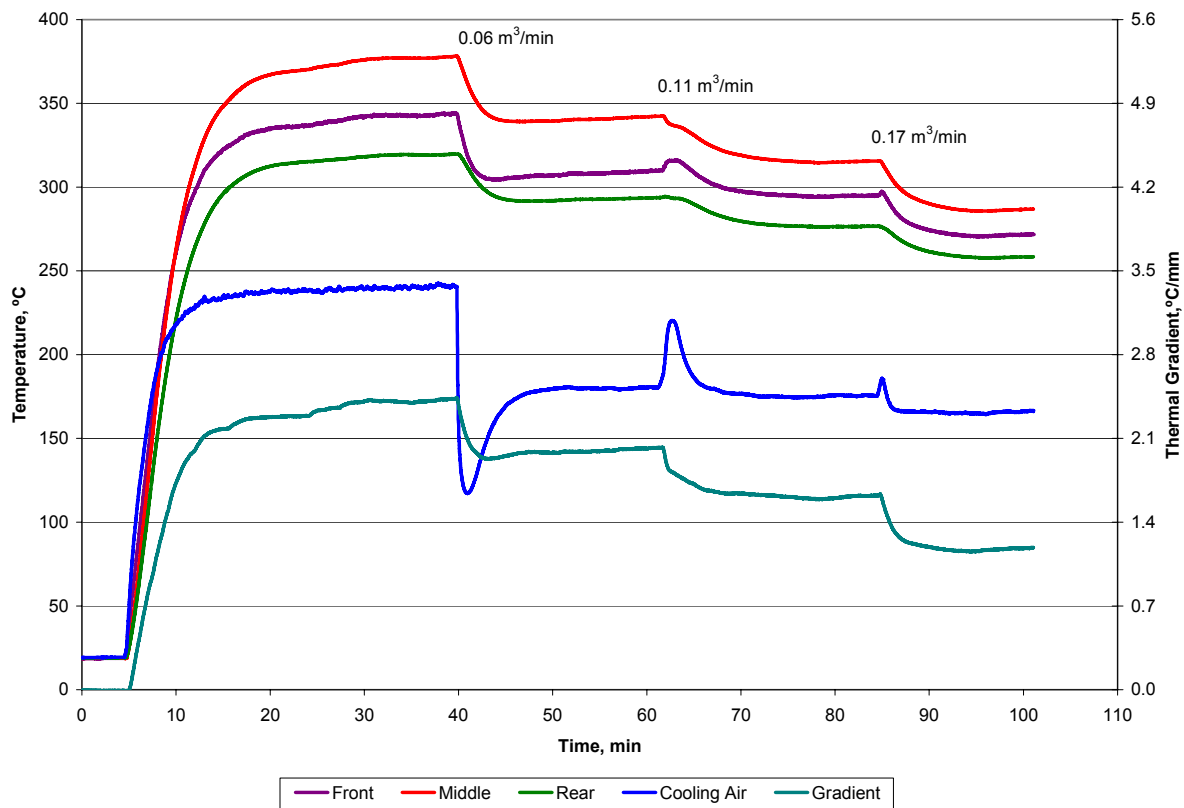


Figure 7.—Test result of direct method with guide tube oriented 180° from high pressure region.

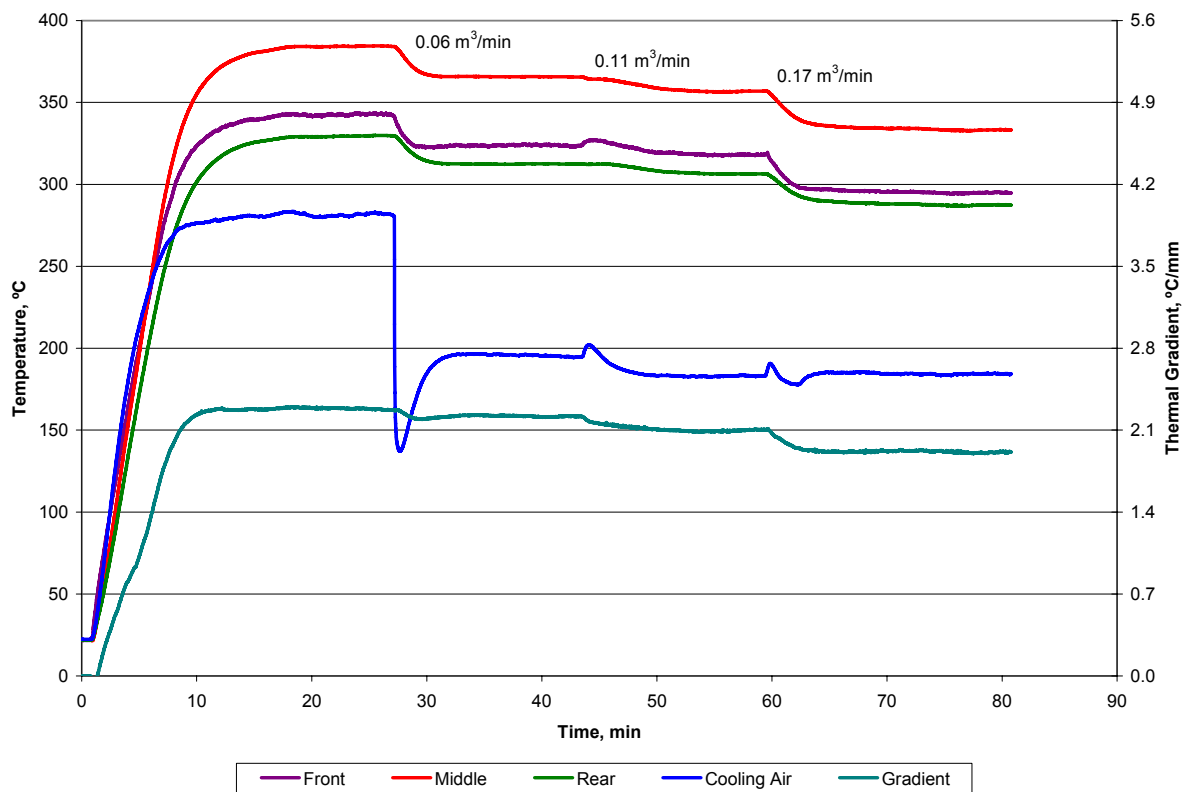


Figure 8.—Test results of indirect cooling method at 0.06, 0.11, and 0.17 m³/min cooling air.

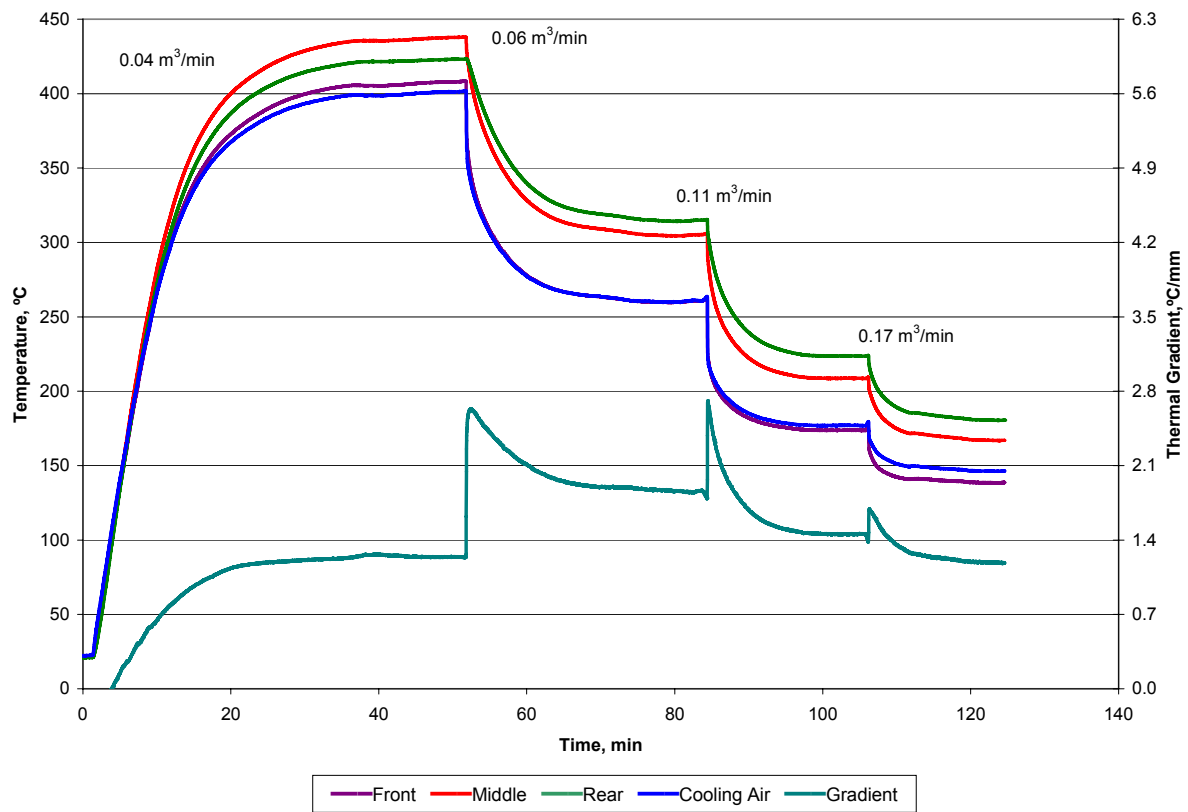


Figure 9.—Test results of axial cooling method at 0.06, 0.11 and 0.17 m³/min cooling air.

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13. ABSTRACT (Maximum 200 words) Tests were performed to evaluate three different methods of utilizing air to provide thermal management control for compliant journal foil air bearings. The effectiveness of the methods was based on bearing bulk temperature and axial thermal gradient reductions during air delivery. The first method utilized direct impingement of air on the inner surface of a hollow test journal during operation. The second, less indirect method achieved heat removal by blowing air inside the test journal parallel to the shaft axis to simulate air flowing axially through a hollow shaft. The third method emulated the most common approach to removing heat by forcing air axially through the bearing's support structure. Internal bearing temperatures were measured with three, type K thermocouples embedded in the bearing that measured general internal temperatures and axial thermal gradients. Testing was performed in a 1 atm, 260 °C ambient environment with the bearing operating at 60 krpm and supporting a load of 222 N. Air volumetric flows of 0.06, 0.11, and 0.17 m ³ /min at approximately 150 to 200 °C were used. The tests indicate that all three methods provide thermal management but at different levels of effectiveness. Axial cooling of the bearing support structure had a greater effect on bulk temperature for each air flow and demonstrated that the thermal gradients could be influenced by the directionality of the air flow. Direct air impingement on the journal's inside surface provided uniform reductions in both bulk temperature and thermal gradients. Similar to the direct method, indirect journal cooling had a uniform cooling effect on both bulk temperatures and thermal gradients but was the least effective of the three methods.				
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