EXPERIMENTAL INVESTIGATION INTO GRAVITY DRAINED JOURNAL BEARINGS

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ABSTRACT

Turbomachines with large & heavy rotors have journal bearings that utilize thin hydrodynamic oil film to maintain a gap between the shaft & bearings. They are fed with a continuous supply of lube oil at a high rate to maintain the oil film and remove the heat generated. The shaft imparts high rotational velocities to the oil as it passes through the bearing. Due to high kinetic energy of oil leaving the bearing, gravity drained bearing housings generally have a big sump near the bottom dead center to collect and reduce the kinetic energy of the oil. This facilitates smooth drain of oil back to the oil tank. The use of gravity to facilitate the draining results in a simple and cost-effective bearing system. The size of sump is determined by the oil flow rate in the bearing housing which itself is a function of rotor load, speed & temperature. In absence of this oil sump (in applications where there is little or no room for a large bearing housing) the swirling oil in the bearing housing doesn’t get enough time to slow down. The rapidly swirling oil therefore fails to drain into the drainpipe(s), and eventually floods the housing and leaks out through the shaft seals. The failure to drain can be attributed to multiple reasons like air pressure fluctuation, oil vortex formation, oil frothing, etc.

This paper focusses on the design of a journal bearing for gas turbines without an oil sump due to design space restrictions. The flow fields in the bearing are chaotic and difficult to analytically predict without experimental validation. Therefore, a bearing rig was constructed, and multiple tests were conducted to understand the flow characteristics inside the bearing housing. Based on the understanding of the flow characteristics, design modifications were made and validated to enable the design of a sumpless gravity drained bearing housing. This paper discusses the methodology and findings from these rig tests which led to the design solutions that solved the issue of draining the high energy oil back to the oil tank without the need of having a traditional oil sump.

NOMENCLATURE

Table:

- \( A_{CH} \): Cross-sectional area of chamber  
  - \( \text{mm}^2 \)
- \( A_{FP} \): Profile area of the film  
  - \( \text{mm}^2 \)
- \( A_{ps} \): Area per pad per side  
  - \( \text{mm}^2 \)
- \( A_R \): Film area ratio  
- \( \beta \): Pad sector angle  
- \( \text{BDC} \): Bottom Dead Center  
- \( \text{CFD} \): Computational Fluid Dynamics  
- \( C_p \): Machined radial clearance  
  - \( \text{mm} \)
- \( \text{CS} \): Compressor Side  
- \( D \): Shaft Diameter  
  - \( \text{mm} \)
- \( H_{CH} \): Chamber height  
  - \( \text{mm} \)
- \( \text{HUN} \): Hydraulic Uplift Number  
- \( L \): Bearing length  
  - \( \text{mm} \)
- \( \theta \): Flow angle with respect to tangential direction  
- \( n \): Number of pads  
- \( Q_s \): Non-dimensional side flow  
- \( q_s \): Side flow rate  
  - \( \text{mm}^3/\text{s} \)
- \( R \): Local radius  
  - \( \text{mm} \)
- \( R_H \): Housing inner radius  
  - \( \text{mm} \)
- \( R_S \): Shaft outer radius  
  - \( \text{mm} \)
- \( \text{TS} \): Turbine Side  
- \( U_0 \): Tangential velocity of air in chamber  
  - \( \text{mm}/\text{s} \)
- \( \text{VOF} \): Volume of Fluid  
  - \( \text{mm}^3 \)
- \( V_a \): Axial velocity  
  - \( \text{mm}/\text{s} \)
- \( V_t \): Tangential velocity  
  - \( \text{mm}/\text{s} \)
- \( \delta_{MAX} \): Max film height  
  - \( \text{mm} \)
- \( \delta_R \): Film height ratio  
- \( \omega \): Shaft angular velocity  
  - \( \text{mm}/\text{s} \)
1. INTRODUCTION

A bearing housing used in gas turbines is a protective casing that supports bearing components as well as the rotating shaft seals. It can be considered as a small oil tank that temporarily stores oil and facilitates drainage. It has a sump at the bottom to collect and reduce the kinetic energy of the oil prior to its drainage. A bearing housing for a low energy journal bearing can be easily sized and designed similar to a regular oil tank. The oil level in a gravity drained oil tank is directly proportional to the inlet flow rate and inversely proportional to the cross-sectional area of the outlet pipe. Designing a bearing housing for a high energy journal bearing is challenging by itself. The challenge is magnified in cases where the design space does not allow creation of a substantial sump. This makes it difficult to analytically estimate the steady state oil level and oil film thickness. The flow in such bearings is a combination of highly rotating multiphase and free surface fluid. CFD models using volume of fluid (VOF) models are utilized to analyze such conditions [1, 2]. These models generally require very fine mesh with several million elements. Also, the analytical results don’t always agree with the test data. The low confidence in the analytical accuracy of the computational method necessitates rig testing to reduce the design risk to an acceptable level. A flexible rig test setup allows changes to the bearing housing’s internal features and other design variables.

Rig tests are generally performed to test the performance of a system or a subsystem before they are put into actual operation. But when the system is not fully designed, the rig needs to have flexibility to change design features until desired results are achieved. A flexible rig was therefore designed that allowed easy attachment and removal of the 3D printed parts. This rig also allowed testing the sensitivity of the design to various design variables like oil and air flow rates. This helped the designers explore the design space and thereby ensure reliable engine operation.

The main purpose of the tests was to design and validate which feature or combination of features would achieve the lowest liquid level at the bottom of the rig. This is commonly referred to as residence volume. The residence volume is the amount of liquid instantaneously present in the chamber/housing at any time. A high residence volume leads to poor performance caused by the heat generation from windage or churning. This heat must be removed to prevent oil coking and thus the work done towards this represents a parasitic loss to the engine [3]. An efficient bearing housing configuration minimizes the intermixing of oil and air in the sump which facilitates fast draining of oil, thereby minimizing the residence volume. This would prevent the oil level from reaching the seals (see Figure 1 for schematic), thereby preventing any potential leakage out of the bearing housing that would cause damage to other engine components.

2. CHOICE OF TEST RIG: ROTATING VS STATIONARY

A rotating rig mimics the geometry and the moving interfaces inside a bearing housing and thereby is the most accurate representation of the domain. A stationary rig, on the other hand, mimics the rotating fluid domain. In a stationary rig the oil flowing out of bearing is simulated by multiple nozzles and has no rotating parts. A stationary test rig was chosen over the rotating shaft rig because of the following reasons:

- A stationary rig provides more stable operating system with less complexity than a rotating rig. It has comparable risk mitigation which is within 5% of a rotating rig.
- This 5% is based on comparative evaluation of small gas turbine bearing housings with stationary and rotating rigs. While flow fields differ in actual distributions, which impacts the bearing housing thermal map, sealing and clearances, the draining characteristics are sufficiently predicted which was the key concern for this design.
- It is easier to design and maintain, and is nearly 50% lower in cost due to absence of rotating components and their drivers (motor, gearbox, couplings).
- Due to faster setup time it allows faster design changes that help in cutting the overall testing schedule by almost half.
- The authors’ organization has experience in both rotating and stationary bearing rigs. Based on the historical testing and engine correlations a stationary rig was deemed the most appropriate for meeting the test objectives with the adequate quality and in meeting the test schedule.

3. TEST SETUP

The rig components were first designed to simulate the actual bearing and bearing housing geometry as far as possible. This is shown in general schematic Figure 1 and the cross-sectional view Figure 2. Then calculations were performed to find velocities and directions of the oil flowing out of the two bearing sides into the bearing housing. Accordingly, choice of test fluid, number of nozzles, their orientations and flow rates were decided. The rig was setup in such a way to allow easy assembly and disassembly of the different design concepts.
A schematic view of auxiliaries and measurement system is shown in Figure 3. The stationary test rig was supplied with glycerol-water mixture from a liquid tank equipped with copper coolers that were fed with water from regular tap water supply. From the liquid tank the fluid was fed to the nozzles by a low-pressure system consisting of one multistage centrifugal pump (variable flow) and a high-pressure system consisting of 4 pump units that were switched on and off as per the flow requirements. The drain fluid from the rig went through the two drainpipes into a receiving tank and was pumped back to the liquid tank by a return pump. The receiving tank was divided into two volumes, one for the drain with dummy oil supply pipe and the other for the drain without the dummy pipe. Bottom valves between the tanks had flexibility to be closed to measure the difference in flow between the two drains. The buffer air was taken from the shop air system with maximum supply rate capability of 400 gram/sec. It entered into circular manifolds in the end walls of the test rig and then into the bearing housing volume through seals designed as the real oil seal in the turbine. The end walls were made with solid transparent plastic (commonly referred as Perspex). The buffer air was eventually sucked out from the receiving tanks to the atmosphere by a vacuum fan mimicking a similar operation in the actual gas turbines.
3.1. HOUSING GEOMETRY

The bearing housing geometry was constructed to have all of the major rotor and stator interfaces. This ensures that the flow encounters the appropriate geometry including the draining features. The manufacturing of the rig was achieved with a combination of conventional and additive manufacturing methods. The test rig consisted of an inner metal part that distributes the liquid to the nozzles. The metal part sat in a printed plastic part in white as can be seen in Figure 4. The circular outer mid-section consists of a Perspex tube with inner diameter matching the journal bearing diameter. The end walls, as can be seen in Figure 5, were machined from solid Perspex blocks and polished to have ordinary glass visual quality. The Perspex parts being transparent helped in performing visual checks inside the bearing housing. The surface roughness / elastic deformation of the plastic components in rig were compared with the actual bearing metal parts. But its impact was deemed insignificant since the temperatures inside the bearing housing are below the oil flash point temperature and the internal surfaces in the bearing housing, especially in the lower half, are mostly machined. Several add-on parts were manufactured in printed plastic, colored in red to make them easy to identify on pictures/movies.
3.2. FLUID SELECTION & NOZZLE ARRANGEMENT

The viscosity, density and surface tension of the oil are the key drivers which determine the drainage characteristics of the oil. The bearing housing was simulated by a mixture of water and glycerine due to the risks associated with oil, like leakage or fire. Also, the use of oil would have increased the complexity of the rig. The percentage of glycerine was altered to match viscosity, density and surface tension of the oil at different operating temperatures. The fluid outflow from the bearing was created by high pressure nozzles directing fluid at an appropriate angle. Calculations were performed to estimate the exit oil flow rate from the actual bearing. Axial and tangential components were calculated (see next section) and accordingly the initial oil flow path was estimated considering the impact of oil on the rotor shoulder and subsequent flow acceleration. Therefore, two sets of nozzles were added on each side of the simulated bearing: 10 HP and 10 LP nozzles on each side, in total 40 nozzles. The HP nozzles have a tangential outflow while the LP nozzles have a small axial component. The low-pressure pump is connected to the turbine side (TS) and the high-pressure pumps are connected to the compressor side (CS), as shown in Figure 5. The fluid from the HP pumps enters the compressor side and then is conveyed to the turbine side through multiple pipes radially outboard of the LP pipe, as shown in Figure 5. The HP pumps simulate the oil that after exiting the bearing pads is further accelerated by the rotor shoulders, whereas the LP pump simulates the remaining oil flow.

There was an assumption made that about 10% of the oil leaving the bearing with this speed would impact the rotor shoulder on the compressor side and the turbine side. The rotor would then impart even more centrifugal velocity to this oil which would greatly enhance its tangential velocity. The HP nozzles were designed to mimic this high kinetic energy oil.

The positioning of nozzles is done uniformly around the bearing housing. For the HP nozzles it is assumed that the oil after impacting the rotor would be whirled around in a uniform manner in the confined space. The LP nozzles simulate the flow that comes in contact with the rotor and is ejected from both sides with a radial velocity matching the journal surface speed. This could mean a circumferentially varying outlet speed on the LP nozzles which would most accurately represent the flow field (due to differential oil velocity profile between upper and lower pads). This was however not done to simplify the feeding circuits and the impact (to drainage) was deemed negligible when compared to the HP turbulence.
3.3. ESTIMATION OF FLOW VELOCITY & ANGLE

The axial component of the oil velocity leaving the four tilting pads was estimated using the calculations in [5] which provided the values for Sommerfield number S and the side flow non-dimensional number Qc. The side flow rate q was first calculated from eq. (1) with the known values of rotor diameter D, angular velocity ω, machined radial clearance C_p and bearing length L:

\[ Q_c = \frac{q_s}{(D/2 \cdot \omega \cdot C_p \cdot L)} \]  

Then the side area per pad A_p (for a non-eccentric bearing) was calculated

\[ A_p = C_p \cdot D/2 \cdot (\pi \cdot \beta / 180) \]  

where β is the pad angle as defined in Figure 7.

Axial velocity:  \[ V_a = \frac{(q_s \cdot (2\cdot n))}{A_p} \]  

where n is the number of pads

Tangential velocity \( V_t \) was approximated to be same as rotor tangential speed.

Tangential velocity  \[ V_t = \frac{D}{2} \cdot \omega \]  

This is the maximum tangential velocity that can be imparted by the journal to the oil. All the oil leaving the bearing does not attain this velocity, but that assumption was made which is considered conservative for the purposes of this study. The LP nozzle angle is the flow angle with respect to tangential direction. It was estimated using the ratio of axial velocity and tangential velocity:

\[ \theta = \frac{V_a}{V_t} \]  

The axial velocity of the oil exiting the bearing pads normalized by the journal surface speed is 1.2x10^{-3} and the average axial component is 0.28 (ratio of \( V_a \) to \( V_t \)). As it can be seen (Figure 1) the geometry of the rotor consists of a step change in diameter on both sides of the journal. Some of the oil exiting the journal bearing is very likely to impact this shoulder and gain kinetic energy. The oil impacting the shoulders on both sides undergoes a velocity transformation to complete tangential. An assumption is made that 10% of the oil exiting the bearing impacts the rotor shoulder further gains kinetic energy which places the velocity ratio to about 1.0. An overall kinetic energy balance was also performed to ensure that the LP and HP pump output velocity summation matched the predicted kinetic energy sum of the bearing housing.

3.4. INSTRUMENTATION AND LAB MEASUREMENTS

The following test instruments were continuously monitored at a sample rate of 60Hz.

| No. | Instrumentation and Lab Measurements          |
|-----|---------------------------------------------|
| 1   | Fluid flow rate – high pressure supply       |
| 2   | Fluid flow rate – low pressure supply        |
| 3   | Fluid Pressure – high pressure supply        |
| 4   | Fluid Pressure – low pressure supply         |
| 5   | Fluid temperature – high pressure supply     |
| 6   | Fluid temperature – low pressure supply      |
| 7   | Fluid flow rate at each drainpipe            |
| 8   | Buffer air supply flow rate                  |
| 9   | Buffer air supply pressure                   |
| 10  | Buffer air supply temperature                |
| 11  | Bearing compartment / Rig internal pressure  |
| 12  | Liquid tank temperature                      |
| 13  | Receiving tank pressure                      |
| 14  | Ambient pressure                             |

Laser Doppler Velocimeter (LDV) was used for:
1. Fluid film thickness on the bearing compartment wall
2. Fluid film speed

Some other parameters were measured either in the laboratory or visually assessed and documented using photo/video:
1. Viscosity and surface tension verification
2. Foaming
3. Flow pattern at drain entrance

3.5. HOUSING CONFIGURATIONS / FEATURES TESTED

In order to capture the oil and assist it in draining into the drainpipes, multiple design concepts were created and tested. The main intent in all of these concepts is to block and redirect the high energy fluid flow in the bearing housing towards the drainpipes. The following design concepts were investigated:

FIGURE 7: Schematic drawing of tilting-pad bearing with four pads and load between pads [5]
1. Baseline
2. Hood/Hat Blocker
3. Hood/Vertical Blocker
4. Vertical Walls
5. Scoops
6. Symmetrical end walls

3.5.1. Baseline – No Added Features

The baseline design without any added features is shown in Figure 8 below.

![Baseline concept](image)

**FIGURE 8:** Baseline concept

3.5.2. Hood / Hat Blocker Concept

In this concept the fluid is trapped near the oil drain using a flat plate protruding horizontally from 7 o’clock position obstructing the flow below it as shown in Figure 9. The intent is to stop the flow just downstream of the drainpipe so that oil naturally drains due to loss of kinetic energy.

![Hood blocker concept](image)

**FIGURE 9:** Hood blocker concept

3.5.3. Hooded Vertical Blocker Concept

This concept is an enhancement of the Hood Blocker concept where the fluid is not only blocked but also guided towards the oil drain cavity using a curved vertical wall (Figure 10).

![Hood vertical blocker concept](image)

**FIGURE 10:** Hood vertical blocker concept

3.5.4. Vertical Walls Concept

In this concept the fluid flow is obstructed using vertical walls on both sides of the drainpipes as shown in Figure 11. A small gap of approximately 5mm has been left between these walls and the bearing housing’s side walls (not shown in Figure 11) to allow a small amount of oil to flow down and get trapped near the bottom and drain. This concept tries to create a sump between the two vertical walls thereby trying to mimic a traditional bearing housing.

![Vertical walls concept](image)

**FIGURE 11:** Vertical walls concept

3.5.5. Wide Scoop Concept

In this concept the fluid flow is obstructed by a single scoop shaped wide blocker (covering both compressor and turbine sides), situated right above the second drain hole (in the
direction of flow) as shown in Figure 12. Multiple variations of the scoops with different heights were tested.

The scoop concept was based on the concepts (Section 3.5.3 and 3.5.4) tested above, that simple blocking and encapsulation of the oil flow did not produce the desired results. The failures of concepts (Section 3.5.3 and 3.5.4) can be attributed to the high kinetic energy and the volume flow rate and the position of the drains. The scoops were designed so that it could drain the required amount of fluid into the drainpipes by a combination of encapsulation and flow redirection.

The height and the width of the scoop are critical in determining the volume of the fluid that is to be encapsulated and drained. An optimization was done in the rig to identify the appropriate height to width ratio to ensure adequate drainage while not overwhelming the scoop system. The optimized scoop design had a semi-circular form with diameter (width) twice and height about half of the drainpipe inner diameter as shown in Figure 12.

3.5.6. Symmetrical End Walls Concept

Finally, in addition to feature changes near the drainpipes, a change in the design of end walls is made to make the two sides somewhat symmetrical. A special insert was added to this effect as shown in Figure 13.

3.6. DESIGN VARIABLES INVESTIGATED

In addition to different types of blockers/redirectors, other design variables were investigated. These tests allowed to determine the design robustness of the down selected configuration. The results of these tests are discussed in Section 4.2.

3.6.1. Size of the Drainpipes Diameter

As part of the sensitivity testing, drainpipes with smaller inner diameter were tested. This was done by adding Perspex inserts in the drainpipes. The reduced pipe diameter made the liquid block the outlet for the air flow and the pressure inside the rig increased. The increased housing pressure resulted in reduction of the fluid supply to the bearing which is undesirable.

3.6.2. Oil Supply Temperature Variation

The oil temperature variations were simulated by changing the fluid viscosities which was achieved by changing the percentage of glycerin in the mixture. This was done for two temperatures other than the standard operating temperature (~90°C). The temperature results are shown in Figure 23.

3.6.3. Nozzle’s Broom Angle

Tests with zero broom angle (a straight jet of liquid) and wider angles were performed. The zero-broom angle test had a
higher velocity when hitting the wall, making the liquid splash more at the wall. No other major difference was observed. The larger broom angle test looked a lot like the nominal 15º spray results. Based on these tests the 15º nozzle angle was chosen for all locations as it allowed better visualization to the flow field without compromising the result.

### 3.6.4. Fluid Swirl Speed Variation

The swirl speed is the average tangential speed of the fluid flow spinning within the bearing chamber. Larger nozzles were deployed which reduced the velocity to nearly half keeping the same volumetric flow rate. This resulted in a marked reduction in the resident swirling fluid volume. The swirling volume has an inverse impact on the draining functionality of the bearing housing. This is due to the fact that the tangential velocity prevents the oil from draining down the gravity supported drain.

### 3.6.5. Fluid Flow Rate Increase

The flow rate was nearly doubled keeping the swirl speed nearly the same (marginal increase). The result was similar to reduced drainpipe size i.e. the air flow was blocked and the pressure inside the rig increased. This reduced the fluid supply to the bearing which is undesirable. This is due to the fact the bearing housing doesn’t have a direct air venting system and the drainpipes are used to vent the air in addition to draining the oil.

### 3.7. FILM THICKNESS ASSESSMENT

Laser Doppler Velocimetry (LDV) was used to measure the velocity inside the transparent wall of the full-scale model of the bearing housing. It uses two intersecting laser beams to give a pulsating frequency when particles or interface surfaces pass the measurement volume. The optical probe is mounted on an adjustable mechanical frame to reach all measurement positions as shown in Figure 14.

The liquid height above the bearing housing’s cylindrical wall in the lower half was estimated by measuring oil velocities at different radial points of a particular circumferential location. The LDV was used to measure the velocity of the oil radially inboard at each location until a sharp drop in the velocity was detected which indicated the oil air interface at that location. Oil film thicknesses were measured by LDV on both compressor and turbine sides of the bearing housing along the lower 180 degrees. Discrete readings were taken at every 9 degrees leading to 20 discrete point measurements on both sides.

As discussed earlier, the main purpose of the tests was to find out which design feature or combination of features would lead to the minimum residence volume. The residence volume was difficult to calculate due to internal geometry complexities and complex flow domains. Instead, a good predictor of residence volume is the film thickness along the housing walls. Large film thickness can cause entrainment onto shaft which may lead to increased leakage and increased power-loss (excessive oil heat-up). Therefore, maximum allowable oil film thickness on the wall was determined that avoided oil entrainment onto shaft using Taylor-Couette flow equation with stationary outer cylinder [4]:

$$ U_\phi(R) = \omega \left[ \frac{R_H^2}{(R_H^2 - R_s^2)} \right] \left[ \frac{R_H}{R} - \frac{R_s}{R} \right] $$  \hspace{1cm} (6)

Where:
- $R_s$ - Shaft outer radius
- $R_H$ - Housing inner radius
- $R$ - Local radius
- $\omega$ - Shaft angular velocity
- $U_\phi$ - Tangential velocity of air in chamber

Average minimum air velocity in the housing was calculated at ~20m/s which resulted in maximum allowable film thickness to chamber height ratio of ~0.2. The failure point of the bearing housing is the least of the two oil heights:

- Height of the bearing housing seals from the bottom
- Height of the oil beyond which there is a high risk of entrainment (calculated by eq. (6))

Of the two, the oil height as calculated by the Taylor-Couette flow equation is the lower. This was used as a guiding criterion to pick the optimum configuration, as is evident from film thickness chart Figure 18 and Figure 19.

![LDV Probe](image_url)
4. RESULTS

A criterion was developed to measure the relative drainage capability of different design schemes. Two ratios, Area Ratio ($A_R$) and Height Ratio ($\delta_R$) were defined as described below:

$$A_R = \frac{A_{FP}}{A_{CH}}$$ (7)

$$\delta_R = \frac{\delta_{MAX}}{H_{CH}}$$ (8)

The chamber height ($H_{CH}$) is the maximum radial distance between the chamber and the oil seal as shown in Figure 15. $\delta_{MAX}$ is the maximum measured film height. It has been observed that the film height is maximum around $+90^\circ$ because the flow rotation and gravity work against each other in this region causing maximum accumulation.

FIGURE 15: Parameters description

The film profile area ($A_{FP}$) is the projected area of the film in the lower half of bearing housing in the plane perpendicular to the turbine axis as shown in Figure 16 (see Section 4.1 for more details on film profile area calculation method).

FIGURE 16: Film Profile Area ($A_{FP}$)

Similarly, chamber area ($A_{CH}$) is the cross-sectional area perpendicular to the turbine axis of the lower half of housing between the chamber cylinder and the oil seal as shown in Figure 17.

FIGURE 17: Chamber Area ($A_{CH}$)

Thickness measurements were made for all design configurations in the lower half and compared with each other as shown in the Figures 18 & 19. The graphs show how the film thickness varies with respect to the angular position in the lower half and provides a visual clue of how different configurations perform in comparison to the baseline design. As can be noted the Hood / Hat Blocker configuration looks worse than the baseline on both compressor and turbine sides.

FIGURE 18: Film Thickness ($\delta_{FP}$)

FIGURE 19: Chamber Height ($H_{CH}$)
4.1. Film Profile Area Calculation

In order to calculate the film’s profile area, the film thickness was measured in the lower half at every 9 deg increments starting at the bottom dead center covering 90 deg in both clockwise and anti-clockwise directions. Then the total film areas were calculated using two different numerical methods of area integration: Trapezoidal and Simpson’s rules [6]. The average of the two methods was then used for the subsequent comparative study.
4.2. Non-dimensional HUN comparison

Hydraulic Uplift Number (HUN) was defined as a drainage criterion to compare test results. It is defined as the product of Film Area Ratio (see Eq. (7)) and the Film Height Ratio (see Eq. (8)).

\[
HUN = A_R \cdot \delta_R \tag{9}
\]

\[0 \leq HUN \leq 1\]

A smaller HUN indicates better drainage performance. The testing suggests that certain design features improve the bearing housing drainage performance better than the others, as can be observed in Figure 20 and Figure 21. The vertical walls and wide scoop assisted the most in drainage but the major breakthrough happened with the use of an insert to make the bearing housing end walls more symmetrical.

With the end wall symmetry inserts, the axial part of the flow coming from each side of the bearing now worked together. The liquid coming out from each side of the bearing after hitting the end walls would center itself straight to the drains. This geometry together with the wide scoop gave nearly no continuous liquid level at the bottom of the bearing housing even though a lot of liquid swirled around in the housing. The liquid line was approximated using videography of the test from multiple angles.

![FIGURE 20: HUN Comparison Compressor Side](image)

4.3. HUN as a function of drain area

The impact of operating temperature on HUN was also investigated. Figure 23 shows the variation of HUN with the temperature ratio (ratio of test temperature to standard operating temperature). The results seem to indicate that outside the standard operating range the HUN tends to increase.

![FIGURE 22: HUN comparison with Drain area ratio (TS)](image)

4.4. HUN as a function of temperature

The variation of the HUN with respect to the drain blockage is an important parameter as bearing drains also provide conduit for instrumentation and other ancillary routing into the gas turbine. Figure 22 illustrates the variation of the HUN with increasing blockage of the drainpipes. The area ratio is calculated as the ratio of the experiment point compared to the baseline drain configuration.
But this is not strictly true since the change in operating temperature also changes the bearing operating point and thereby the flow rates. The tests were repeated three times for each configuration to ensure a repeatable data was arrived at.

FIGURE 23: Variation of HUN (TS) with operating temperature

5. CONCLUSION

The rig testing validated that the drainage of oil from a low volume high kinetic energy bearing housing is complex. The oil flow in the bearing housing is highly three-dimensional, and the flows coming from each side of the bearing housing interact strongly.

Based on the test results, the wide scoop concept along with end wall symmetry inserts gave the best possible drainage configuration for this bearing housing. This configuration also exhibited the least HUN in the design space sensitivity tests that included blockage studies.

This unique design of bearing housing (without oil sump) provides significant benefit where packaging space is tight and thus reduces the overall cost of the gas turbine.

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