OPERATION AND CURRENT DEVELOPMENTS OF SELF-CONTAINED, SELF-LUBRICATING THRUST AND JOURNAL BEARINGS SYSTEMS

by
Matthew M. Marchione
Senior Design Engineer
Kingsbury, Incorporated
Philadelphia, Pennsylvania

Matthew M. Marchione is a Senior Design Engineer with Kingsbury, Incorporated, in Philadelphia, Pennsylvania. He is responsible for the design, analysis, development, and troubleshooting of fluid-film thrust and journal bearings. In his 13 years with Kingsbury, Inc., he has also served as a Service Engineer and Research Engineer. Prior to joining Kingsbury, Inc., Mr. Marchione worked for Vibration Specialty Corporation, Philadelphia, Pennsylvania, and Mechanical Technology Inc., Latham, New York. In both positions he performed field service and analytical studies aimed at understanding and eliminating machinery vibration.

Mr. Marchione received a B.S. degree (General Science, 1980) from Villanova University, and is a member of the Vibration Institute. He has written numerous technical papers, coauthored and contributed to EPRI Final Reports, and has lectured on many topics.

ABSTRACT

Self-lubricating, self-contained fluid-film thrust and journal bearings have been in use since the 1920s. These bearing systems offer relative simplicity in a reliable, stand-alone package. Recent testing has shown that these systems are capable of much more, although history has shown that the workings of the system are largely misunderstood. The purpose of this paper is twofold: one, to explain the proper operation of a self-contained system and how mysterious and repetitive anomalies are easily corrected, and two, present and discuss the experimental results of testing aimed at expanding system applications.

INTRODUCTION

Simplicity and functionality in design are common goals of design engineers. With ever increasing emphasis on the bottom line, product improvements that do not increase product cost must also be a goal if a product is to remain both viable and competitive. Plant and reliability engineers are concerned with installation, maintenance and repair costs, as well as the product life cycle. Low total cost and long service life are both desirable features of an enduring product.

Self-lubricating, self-contained fluid-film thrust and journal bearings systems are such a product (Figure 1). They offer design simplicity, and used in place of external bearing lubrication skids, provide fewer parts, lower initial and long-term costs. Since they operate as soon as shaft rotation begins, they need no involved starting sequence. In addition, power outages do not affect bearing lubrication during coastdown. An external lubrication system needs to be started before the related machine(s), and requires a backup power source, battery driven pump, or gravity tank to offer lubrication during an emergency shutdown. Therefore, use of the self-lubricating self-contained system means a reduction in machine skid size, weight, complexity, and maintenance.

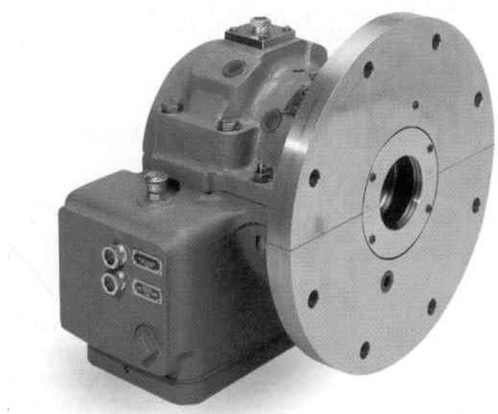

Figure 1. Combination Bearing and Housing Assembly.

The advantages of these systems are even greater through expanded applications. By refining the design and evaluating system capabilities through test and field applications, self-lubricating, self-contained fluid-film bearing systems appear to have the potential to meet many user requirements.

OPERATION, ANOMALIES, AND CORRECTIONS

Self-lubricating, self-contained hydrodynamic bearing systems are typically flange-mounted components that include a load-equalizing double-thrust bearing, a plain journal bearing, an oil reservoir, an oil circulator, and a separable or integral thrust collar. The circulator and collar form a viscosity pump. A second plain journal bearing in a separate housing may also be included (Figure 2). The viscosity pump becomes active with shaft rotation. As long as the machine shaft rotates, pressurized oil is available to the system bearings. Directional changes in shaft rotation do not affect viscosity pump operation.

Figure 2. Separate Journal Bearing Assembly.

Figure A-1 in APPENDIX A presents a cross section of the combination housing and identifies the major components. The equalizing double thrust bearing is positioned on the outboard end of the housing, and the journal shell on the inboard. The combined housing, which is typically located on the outboard or nondrive end (NDE) of the pump, is bolted to the pump housing through a mounting flange.

The bearing housing is equipped with an oil reservoir. Oil discharged from the thrust bearing flows through the heat exchanger and back into the reservoir, while oil exiting the journal shell flows directly into the reservoir.

The inboard side of the bearing housing is typically equipped with an end seal to prevent oil contamination and/or leakage; the outboard end of the bearing housing is enclosed with an end cover plate. The end cover plate supports the thrust load when the outboard thrust bearing is active. If a through-shaft is required, an end seal similar to that used on the inboard side is applied to prevent leakage along shafting extending through the end cover plate.

The separate bearing housing containing only a journal shell is also bolted to the pump housing through a full mounting flange or half flange with a spigot. Figure A-2 in APPENDIX A identifies the major components. This bearing is dependent upon the combination housing for lubrication. The housing is typically located at the inboard or drive end (DE) of the pump. The inboard and outboard sides of the housing are equipped with end seals. Figure 3 presents the combination and separate bearing housings mounted on a pump.

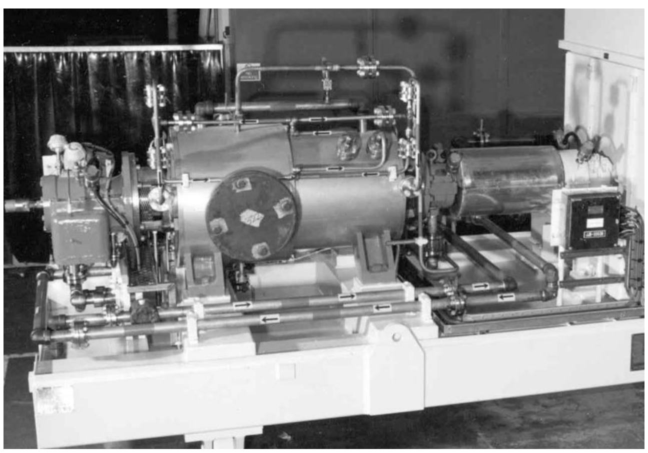

Figure 3. Self-Lubricating Self-Contained System Mounted on a Pump.

Operation

The heart of the viscous pump is the thrust collar and close-fitting circulator. Lubrication occurs as long as the shaft is rotating. Figure A-3 in APPENDIX A illustrates the oil flow path through the combination bearing housing.

With the exception of a portion of the thrust collar, the rotor sits above the oil level while at rest. The instant shaft rotation begins, the adhesiveness of the oil carries the circulator with the collar until the lug at the circulator top meets a housing stop. In this position two circulator ports align with housing ports. Oil is drawn from the reservoir and is carried around the shallow groove in the circulator bore. After almost a full rotation, the oil meets a dam in the groove and is scraped off. The oil, pressurized by oil behind it, flows through a drilled housing passage on the way to the thrust and journal bearings.

Oil traveling to the thrust bearings flows through vertical ports and enters each bearing between the two lowest shoes. Collar rotation assists in carrying the pressurized oil through the other thrust shoes, flooding the thrust cavity. Oil seal rings located in the bore of each base ring minimize the volume of oil needed to flood the cavity. The oil then flows through internal housing passages to the heat exchanger. Cooled oil returns from the heat exchanger into the reservoir.

Oil flowing to the journal bearing travels through the same initial drilled passage until it meets a cross-drilled passage leading to the bearing seat. Pressurized oil is delivered to the shell and exits both through drain slots and along the shaft. Used oil flows through drilled passages back into the reservoir, where it mixes with the cool oil returned from the heat exchanger.

If the direction of rotation is reversed, the circulator lug will run against the opposite housing stop, and oil will be drawn from a second set of suction holes. The lubricating scheme remains the same.

Several taps off the pressurized feed line are available to lubricate the separate journal bearing. Piping is connected to these taps to deliver the oil to the additional bearing.

The oil flow path through the separate journal housing is presented in Figure A-4 in APPENDIX A. A single oil supply line leads from the combination housing into the journal shell supporting rib in the single housing. Pressurized oil travels through the supporting rib to the bearing seat. The oil travels through additional passages and enters the shell at the bearing joint.

Used oil drains out both through drain holes and along the shaft. A single drain line directs the oil back to the reservoir where it mixes with the cool oil returned from the heat exchanger.

Supplying the separate bearing with oil is relatively easy since the oil is pressurized. However, returning used oil to the housing is not as straightforward.

The oil exiting the separate journal must be returned to the reservoir, but the oil is no longer pressurized. The forcing function driving the oil back to the reservoir is the change in elevation. At rest, the oil levels in both bearing housings are the same. Once rotation begins, the thrust cavity fills with oil, and the separate journal is supplied with oil. Both these events cause the oil levels in the combination bearing housing to drop. The oil level in the separate journal housing remains higher than that in the combined housing, and a small pressure is developed based on the oil head. The pressure (Cameron Hydraulic Data, 1984) can be determined by:

$$P = \frac{SG * H_{\underline{f}}}{2.31} \quad (1)$$

where:
P = Pressure (psig)
SG = Specific gravity
$H_{\underline{f}}$ = Head in feet

Oil Considerations

The quality, viscosity, temperature, and available quantity of lubricating oil are critical to the long-term operation of any fluid film bearing system. This is certainly true of the self-contained, self-lubricating system, although there is one area that deserves special attention.

Oil filters are commonplace in lubrication systems, but they present a potential hazard when used with the self-contained, self-lubricating system. As the filter element fouls, the increasing flow resistance reduces the flow through the filter. This will likely result in the redistribution of flow through the system, and depending on the filter location, this may result in oil leakage or bearing failure. An oil filter also presents an obstruction in cold weather starts.

Filtration devices may be used, but only with caution. The filter must be located in a pressurized supply line, and the expected pressure should be known in order to apply an appropriate device. In general, the supply pressures are too low for a differential gauge to provide an accurate, timely warning.

The system must initially be filled with clean, filtered oil. Since the volume of oil required to fill a complete system is relatively low—in most cases less than 10 gallons (~38 liters)—it is far easier to periodically sample the oil and change as required. Given that the system has a limited amount of oil, it is prudent to replace the oil drawn for sampling with clean filtered oil.

The heat generated by bearing operation must be removed from the oil. Depending on both ambient and operating conditions, convection, an oil to water heat exchanger, an oil to sea water heat exchanger, or forced-air cooling may be used. The heat exchanger may be mounted to, or mounted remotely from, the bearing housing. As noted above, only the oil discharged from the thrust bearing flows through the heat exchanger.

Initial Startup

The journal bearings start on residual oil from the last operating period, or from wetting the parts with oil during initial assembly. In addition to residual oil, a portion of the thrust bearings sits below the oil level, and more oil is delivered to the thrust bearing at startup by the wetted collar.

This boundary lubrication is sufficient to protect the bearings until the viscosity pump has primed and is delivering a continuous supply of oil to the bearings. The time required for a lubricant to prime varies based on bearing size, rotational speed, and oil viscosity, but in most cases full-film lubrication occurs from less than a second to several seconds.

A machine with these self-lubricated bearings may be started even after extended periods of time without bearing damage. Providing the driver supplies the required torque, the machine may also be started while the thrust bearings are loaded up to the rated bearing load.

Anomalies

With an understanding of system operation, potential problems can be easily avoided. Following are examples of situations that can lead to high bearing temperatures and possible bearing failures, with comments on prevention or solution (refer to Figures A-1 and A-2 in APPENDIX A).

Lack of Lubrication

• *Circulator burrs*—As machine rotation begins, the circulator is designed to rotate with the collar until the tab reaches a stop in the housing. This position indexes the oil suction and discharge holes. However, if the circulator is not free to rotate, the oil passageways will not align, resulting in oil supply restrictions. The circulator may bind if distorted, scratched, or damaged during handling, installation, or maintenance activity.

When accessible, check the fit of the circulator in the housing recess by grasping the lug and moving it side to side. It should move freely, but not axially.

• *Cold oil—From ambient temperature/excessive cooling water*—The cooler and more viscous the system oil, the more difficult it is to circulate through the bearings, especially if a separate journal bearing is used. If the oil temperature is too cool (<~10°C (50°F)), the relatively low pressure developed by the viscous pump may not be capable of circulating the oil, leading to marginal bearing lubrication and potential damage.

Some readily available cooling water sources have high sediment and contaminant levels. In order to prevent heat exchanger fouling, this water is often run through the heat exchanger at high velocities, even when the unit is off-line. This can create "plugs" of high viscosity oil that make circulation difficult.

To avoid these problems, use sump oil heaters and, if required, heating tape on exposed piping. Stop the cooling water flow when the machine is off-line.

• *Oil level*—The reservoir oil level is typically monitored by comparing the level in a sight glass to range plates secured to the housing. If the oil level is too low, dropping below the top of the circulator suction holes, the circulator will not prime, and pumping will not occur. If the oil level is too high, the result will be leakage while at rest and in operation. Either at rest or in operation, the leakage will stop once an equilibrium point is reached. However, if the unit leaks at rest, it may leak again once rotation begins.

To properly fill the reservoir, remove the fill cap or a breather vent. Using a funnel, pour oil into the housing until the level rises to and maintains the "FULL" level as indicated by the oil gauge plate. The oil will flow through the heat exchanger rather slowly; therefore, check the oil level frequently and add oil as required.

If a separate journal bearing is used, the supply line must be filled with oil. The easiest way to accomplish this is to add a fill location to the supply line. If the fill location is not available, the oil will fill the supply line, but it will occur rather slowly.

Ensure that the oil level reads "FULL" on the range plate when the unit is at rest. Check the oil reservoir level after the first several periods of operation. Add oil as required.

• *Rapid restart*—Once an operating machine is shut down, the lubricating oil in the thrust cavity must drain back into the reservoir. Since the thrust cavities are not vented, it may take several minutes for this to occur. If a restart is attempted prior to evaluating the reservoir oil level, the circulator may not prime, and oil pumping will not occur if the oil level is too low.

It is important to note that if the oil level drops below the top of the circulator suction holes while in operation, lubrication will continue. However, if the oil drops below the top of the circulator suction holes while the unit is at rest, the circulator will not prime.

Evaluate the reservoir oil level prior to each start. Top off the reservoir as required.

• *Collar finish*—The rotating collar is at the heart of the viscous pump and the proper finish is critical. Too smooth a finish and the collar will not adequately pump the oil; too rough, and excessive oil churning or component damage may result.

The optimum finish to operate on the bearing surfaces is 12 to 16 microinches rms (0.3 to 0.4 rms microns). The optimum finish on the collar outer diameter is 63 microinches rms (1.6 microns rms).

• *Seal rings*—Seal rings are used to limit the volume of oil needed to flood the thrust bearing cavity. If the seal ring clearance is excessive, or the seal rings are omitted during assembly, the thrust cavity may not fill with oil. This will result in a limited oil supply. The bearing temperatures will increase dramatically due to the limited oil supply and minimal flow through the heat exchanger.

Ensure the oil seal ring clearance is within tolerance and the seal rings are not omitted during assembly.

• *Circulator/collar clearance*—The clearance between the circulator and collar forms a seal, allowing the oil to be pressurized. As the clearance increases, the efficiency of the viscous pump declines. The recommended clearance is approximately 0.001 inches (0.03 mm) per inch of collar diameter.

Ensure that the circulator to collar clearance is within tolerance. Carefully evaluate the circulator surfaces for deep scratches and/or assembly damage.

• *Collar position*—It is not uncommon for the vertical centerlines of the collar and circulator to be offset during operation. However, large offsets caused by assembly errors may cause the chamfered edges of the collar to enter the circulator bore. Aside from pump-related problems due to rotor position errors, the circulator/collar sealing area is minimized and oil pumping capability is lost.

Thrust bearing end play must also be considered. End play is the free float of the thrust collar (rotor) between the two thrust bearing subassemblies. It is provided to allow for thermal expansion and formation of oil films. The proper end play is typically set by the manufacturer. However, if gasket material is used to seal the end cover plate during assembly, the end play, and more notably the rotor position, are affected. If the rotor thrusts in the outboard direction, it will move the gasket thickness in addition to the end play, potentially breaking the circulator to collar seal.

Be aware that gaskets used with the end cover plate will affect end play and possibly rotor position. Verify end play and relative collar to circulator position at assembly.

Leakage

• *Venting*—Proper venting is critical to the proper operation of self-lubricating bearing systems. The system pressure is relatively low compared to skid-mounted lubrication systems, so air-bound cavities will restrict flow and cause leakage.

The most common venting issues occur with air vents external to the housing. In older applications, an external flow tube was installed from the top of the assembly into the reservoir (Figure 4). The flow tube ended in a sight glass threaded into the housing. The purpose of the sight glass was to ensure the operator that oil was being pumped, as evidenced by the steady stream of oil flowing through the glass. In order to serve a dual purpose, the gaskets in the sight glass were removed, effectively making it an air vent. If the stream of oil flowed cleanly through the glass, operation was trouble free, but if the flow hit the side of the glass, leakage could occur. If an alert mechanic saw the leakage, or if someone noticed that the gaskets were missing even though there was no leakage, the gaskets were replaced, venting was lost, and the separate journal began to leak.

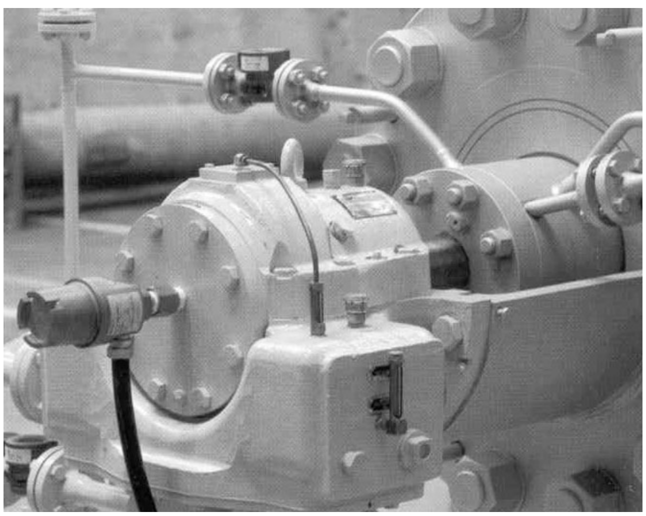

Figure 4. External Flow Tube.

Similar problems occur if the sight glass is painted or becomes clogged from dirt or contaminants.

Newer designs incorporate an internal vent that is not exposed to the environment. A “bulls eye” or similar viewport is used to provide visual assurance of oil flow.

Venting of the separate journal housing is also critical. Without proper venting, the oil will not return to the reservoir. This may occur if the housing vents are painted or excessively dirty. In some applications the separate housings do not have vents; they are equipped with relatively large end seal clearances. If the end seals are replaced with tighter-clearance seals, venting is compromised, and the oil level increases until leakage occurs.

Venting in large clearance end seals is quickly lost if the oil level reaches the shaft/seal interface. Upon reaching this region the oil is quickly carried around the shaft with rotation, filling the seal gap. This effectively seals the air vent provided by the clearance, and results in leakage.

Identify the bearing housing air vents. The vents must be kept clean and free of debris. They should never be painted. Do not replace sight flow gaskets if they are absent.

• *Cold oil—From ambient temperature/excessive cooling water*—Moving oil in the return line is critical to proper system operation. Cold oil (<~10°C (50°F)) is very resistant to the small differential head returning the oil to the reservoir. Since some oil is delivered to the separate bearing via the pressurized supply, but little returns due to the small head, leakage from the separate journal housing seals will occur.

Use heating tape on exposed piping. Stop the cooling water flow when the machine is off-line.

• *Windage*—The low pressure area created by coupling or other component rotation adjacent to the bearing housing may cause leakage.

Shield the housing from the coupling through baffles or an adequate seal. The bearing cavity may also be vented.

• *Piping*—Piping is required to both supply and return the oil from the combination to the separate bearing housing. Since the return oil is driven by low pressure, restrictions or pressure drops of any kind will hinder the return flow. The return piping must therefore be free of excessive fittings, valves, filters, or anything else that has the potential to restrict flow.

Over time the oil in an operating machine becomes aerated, therefore the return piping is critical. For this reason the separate journal discharge piping drops vertically as far as space will allow. Large bubbles, if present, will rise up the discharge pipe and vent through the separate housing, rather than get trapped in the piping.

Figure A-5 in APPENDIX A illustrates the proper piping layout for self-lubricating systems. An elbow is used to route the piping perpendicular to the shaft and exit the pump envelope. A second elbow is used to direct the flow at a positive incline relative to the shaft centerline, back toward the reservoir. A third and final elbow is used to redirect the oil flow into the reservoir, again perpendicular to the shaft.

The large run of piping must have an upward slope to prevent the entrained air from collecting in piping elbows or transitions. If the air collects in these locations, the pipe diameter will be reduced, restricting flow.

The entire pipe must enter the reservoir below the oil level or the effective pipe volume/area is lost and flow is reduced.

• *Moving/offset horizon*—The primary driver of the used oil from a separate journal bearing to the reservoir is the difference in elevation between the housings. Self-lubricating bearings are not suited to shipboard or barge applications due to a shifting horizon. Similarly, the slope of a poorly installed pump skid may be enough to prevent the return flow of oil from a separate journal shell to the reservoir.

Evaluate the relative position of the bearing housings once the pump has been installed onsite. If necessary, correct the positions.

• *Oil level*—As noted above, too low an oil level may lead to bearing starvation. However, if oil is added above the housing “FULL” mark, it may leak while the pump is either operating or at rest. The leakage will stop once the excess oil has been purged.

Examine the oil level prior to starting the unit. Reexamine the levels after the first several operating periods.

Additional Considerations

It is commonly held that some or all the above anomalies are better and more reliably resolved by equipping the self-lubricating system with a traditional lubrication system. Although possible, this approach is not recommended since the joining of these systems creates additional challenges and reliability issues.

The circulator/housing interface acts as a pipe in typical operation; however, the pressure created by an external supply will cause leakage. Depending on bearing size and shaft speed, the pressure may reverse the circulator oil flow. If the circulator is not fixed, its position may change, altering or even eliminating the oil flow paths. In addition, the high flowrates are not a benefit. Without the use of orifice plates, the increased oil supply to the journal bearings may cause leakage from the housings. Since the means of return oil flow is unaffected by the external lube skid, separate housing leakage may occur due to the now inadequate return line.

INHOUSE AND FIELD TESTS

Test Rig

In order to further enhance capabilities and advantages, a self-contained, self-lubricating bearing test stand was designed and constructed (Figure 5). The prime mover is a 60 hp (44.7 kW) variable speed AC motor capable of test speeds up to 7200 rpm. It is coupled to a 110.60 inch (2.8 m) long test shaft with a 76.00 inch (1.9 m) bearing span.

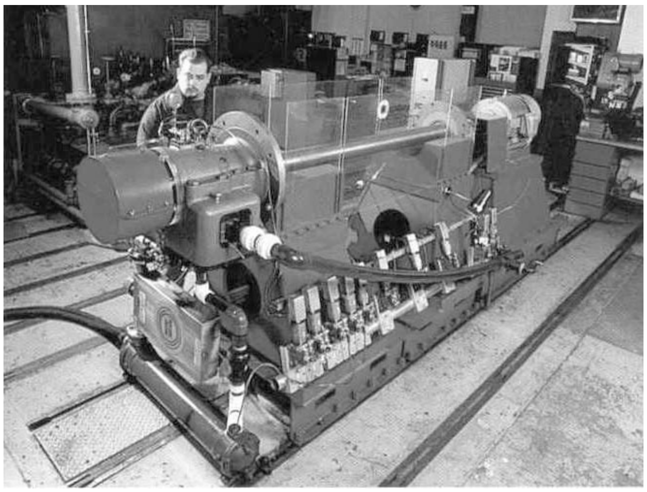

Figure 5. Test Rig.

The 450 lb (2000 N) test shaft is supported by two 4.33 inch (110.0 mm) diameter by 2.56 inch (65.0 mm) long journal bearings. The two bearing housings are mounted to separate structures. The inboard or DE housing contains only a journal shell; the outboard or NDE housing contains the viscosity pump, the oil reservoir, a journal bearing, and a double 9.00 inch (228.6 mm) equalizing thrust bearing. A heat exchanger is mounted on the floor beneath the combined bearing housing.

The inboard and outboard bearing housings are connected with two runs of clear plastic pipe. The oil supply pipe from NDE to DE housing is 1.0 inch pipe; the return line is 2.0 inch pipe. The NDE housing end cover plate and all end seals are made of clear plastic. Numerous clear plastic windows were installed in the housings as observation ports.

The rig is equipped with 10 pressure sensors, two flow meters, axial and radial displacement probes, and a wide array of thermocouples. Figure A-6 in APPENDIX A presents an overview of the test rig instrumentation.

The pressure sensors are located at the NDE journal bearing feed and discharge, the NDE housing thrust cavity, the DE journal bearing feed and discharge, the DE feed near the NDE and DE housings, the heat exchanger inlet and outlet, and the oil circulator. Holes were drilled in the oil circulator at six locations, and copper tubing was soldered to each of these locations (Figure 6). The tubes were then routed to a manifold. Valves were used so that one pressure sensor could be used to measure each pressure tap.

One 50 gpm (189.3 lpm) capacity flow meter is located between the combination housing and the heat exchanger. This meter measures the total amount of oil flowing through the thrust bearing. The second flow meter (10 gpm (37.9 lpm)) is located in the feed line to the separate journal shell. The flow to the journal shell in the combined housing is conservatively assumed to be no less than the separate journal shell.

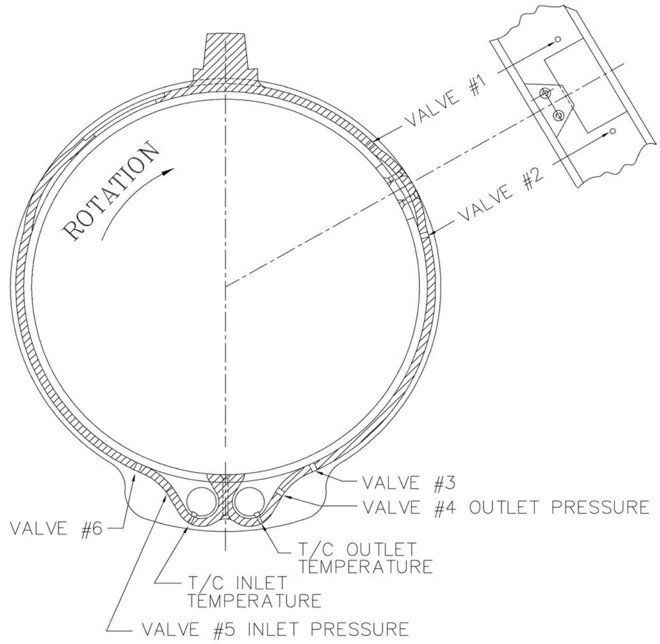

Figure 6. Circulator Instrumentation Locations.

Small J-type thermocouples are used to measure the oil inlet and outlet temperatures of the oil circulator, the thrust bearing, the internal and separate journal shells, and the heat exchanger. The thrust cavity and housing skin are monitored, along with three sensors mounted in the oil reservoir. Thermocouples are also embedded in the babbitt of the journal and thrust bearing pads. The actual thermocouple junction is positioned within 0.03 inch (0.8 mm) of the babbitt metal surface.

Radial displacement probes were used to monitor shaft vibration. An axial probe was used to monitor shaft position. Data were recorded with a data acquisition system capable of simultaneously monitoring all 43 channels of data.

A series of four sight glasses was also mounted near the NDE housing to monitor the elevation difference in the reservoir, the inboard and outboard NDE journal bearing cavities, and the DE journal bearing cavity. The levels were recorded manually.

The test rig was filled with approximately 8 gallons (30.3 liters) of ISO VG 32 light turbine oil. Initial tests were preformed in 300 rpm increments, from 300 to 6300 rpm. Long operating periods outside this speed range caused the drive motor to overheat and trip. Data were acquired outside the range, but the run and subsequent data stabilization time were lower.

Early test results indicated that the oil flow available to both the double thrust and journal bearings exceeded typical recommended flows. Figures 7 and 8 show the thrust and journal bearing flows, respectively. Subsequent testing focused on how to best utilize the additional oil.

It was believed that the benefits of providing oil to other remote bearings could be easily attained by reducing excess thrust bearing oil flow and redirecting it to the remote locations.

In order to reduce the thrust bearing oil flow, a restriction had to be added to the system. A ball valve located near the inlet of the heat exchanger was used for this purpose. The thrust bearing oil flow was held to approximately 12.0 gpm (45.4 lpm) at 3600 rpm. By closing the valve and restricting the oil flow through the heat exchanger, the thrust bearing housing cavity pressure increased, as did the oil flow to both the internal and separate journal bearings. Figure 9 presents the cavity pressure before and after the flow through the heat exchanger was restricted.

Referring to Figure 9, the thrust cavity pressure increased from 6.5 psig (0.5 barg) to almost 22.0 psig (1.5 barg) at 3600 rpm. The circulator outlet pressure was also noted to increase from 17.0 psig

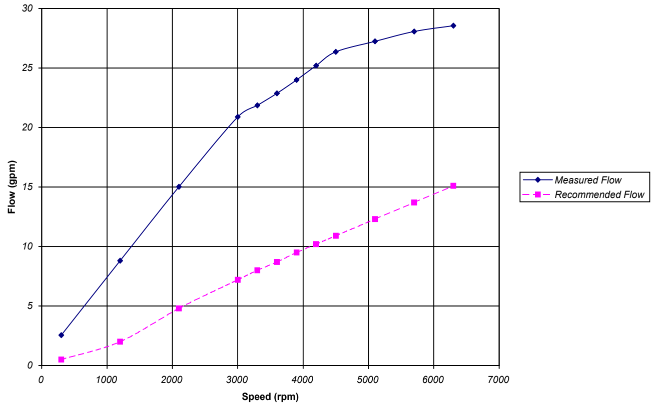

Figure 7. Comparison of Measured and Recommended Thrust Bearing Flows.

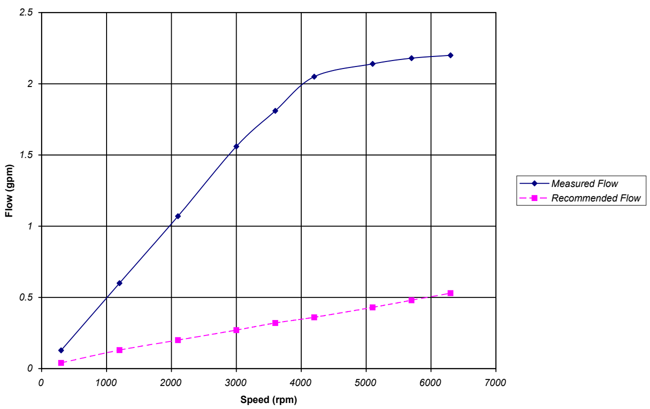

Figure 8. Comparison of Measured and Recommended Journal Bearing Flows.

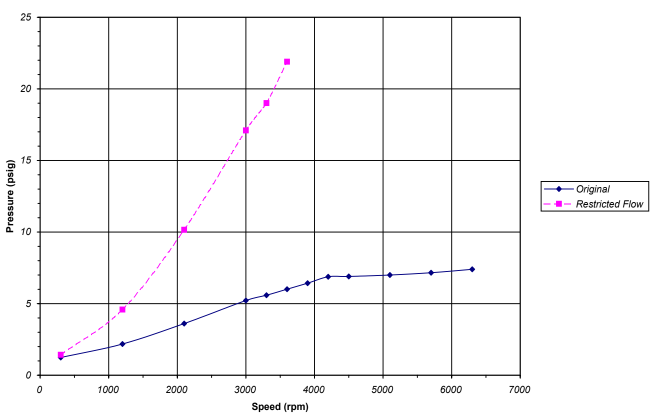

Figure 9. Comparison of Normal and Restricted Flow Thrust Cavity Pressure.

to 30.0 psig (1.2 barg to 2.1 barg) at 3600 rpm. This increase in pressure enabled the system to increase the separate journal bearing flowrate from 1.8 gpm to 2.8 gpm (6.8 lpm to 10.6 lpm) at 3600 rpm.

A reduction in thrust bearing flow was expected, since this is the nature of a viscosity pump. The result of increased pressure is lower flow. As the pressure drops, the flow rises. The thrust bearing flows decreased with the increased pressure, but more than an ample supply remained. Figures 10 and 11 compare the thrust and journal bearing flows, respectively, both before and after restricting the oil flow through the heat exchanger.

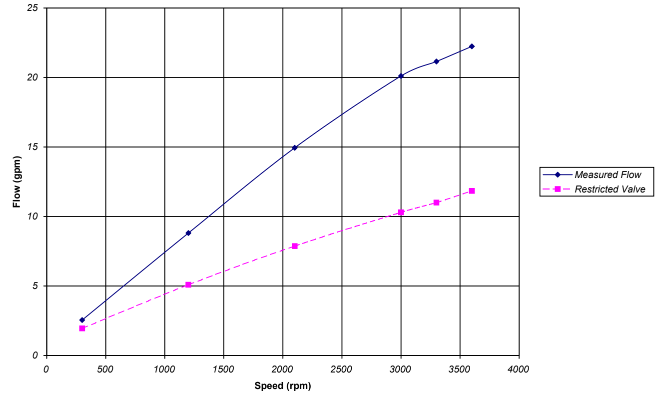

Figure 10. Thrust Flows Before and After Increasing Cavity Pressure.

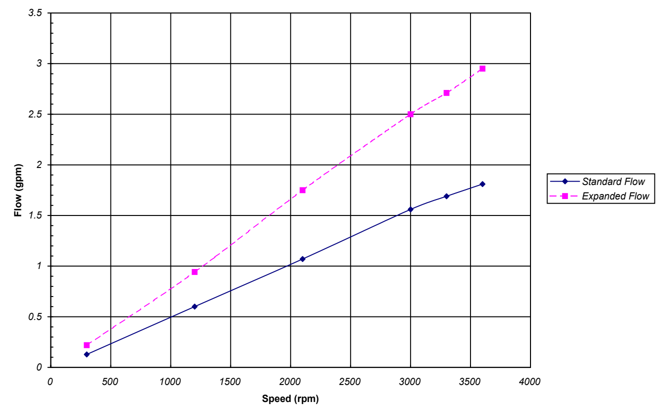

Figure 11. Journal Flows Before and After Increasing Cavity Pressure.

At this point it appeared that the viscosity pump was capable of supplying even more remote bearings. In order to test this, an auxiliary tank was piped into the system to simulate driver bearings. The tank was placed beyond the test rig motor NDE. A 1.0 inch pipe was placed from a feed available in the end cover plate (Figure A-1). The line was routed into the auxiliary tank below the standing oil line. A flange was added into the feed line to allow for the installation of an orifice plate.

The separate feed line allowed the in-place flow meter to measure the flow to the one separate journal bearing. Flow to the auxiliary tank was determined by blocking the return oil flow and measuring the increase in oil volume.

A return line was added to the bottom cover plate of the auxiliary tank. Typical of self-lubrication system oil return lines, the return line dropped straight to the floor. A piping elbow directed the pipe perpendicular to the shaft centerline, outside the driver envelope. Another elbow was used to turn the return line toward the reservoir in an uphill slope. The return line from the auxiliary tank joined the remote journal bearing return flow in the coupling area.

All the additional piping was made of clear plastic. A valve was added to the auxiliary tank supply line, and a plug to the drain line, so that the test rig could be operated without the tank.

The four sight gauges were rearranged to provide the levels of the reservoir, the "pump" NDE journal bearing, the "pump" DE journal bearing, and the auxiliary tank. The levels were recorded manually.

Initial tests began in the auxiliary test supply line. Since oil heaters were not used, the test rig was initially started at 300 rpm and then was carefully brought up to approximately 2400 rpm to warm the oil. After reaching 120°F (49°C) (as measured at the circulator inlet) the speed was returned to 300 rpm and data collection began.

The oil temperature was maintained throughout the tests. Data were recorded after the oil levels stabilized after the system reached equilibrium. At the completion of the first test, the 1/4 inch orifice plate was replaced with a 3/8 inch orifice, and the test was repeated. The results are presented in Figure 12.

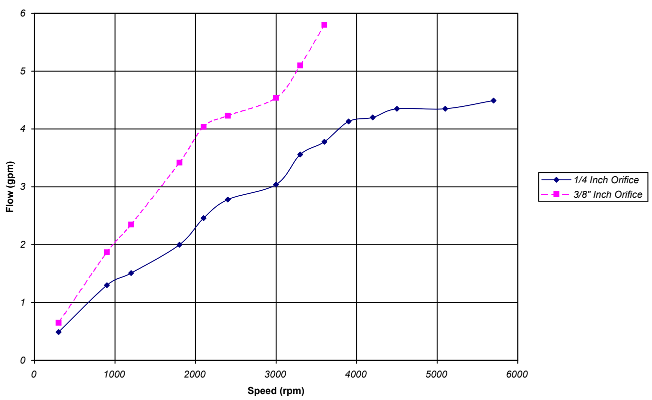

Figure 12. Auxiliary Tank Oil Flows—1/4 and 3/8 Inch Orifices.

Again, a reduction in the existing oil flows was expected when supplying the auxiliary tank with oil. Figures 13 and 14 compare the original thrust and journal bearing flows, respectively, to the measured flows after installation of the 1/4 inch and 3/8 inch orifices. As expected, use of the largest orifice resulted in the biggest change from the original levels. The flows still remained above the recommended flow values.

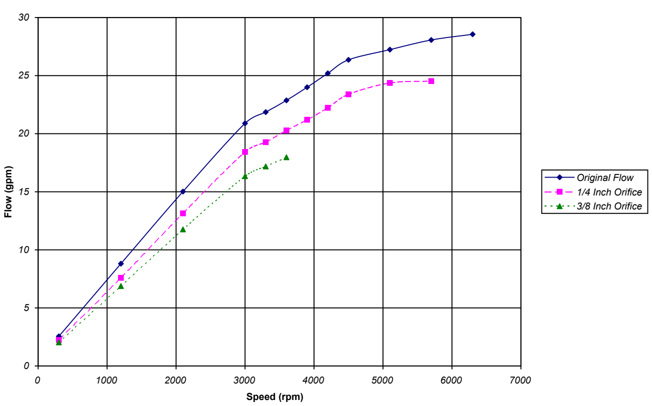

Figure 13. Comparison of Thrust Bearing Flows.

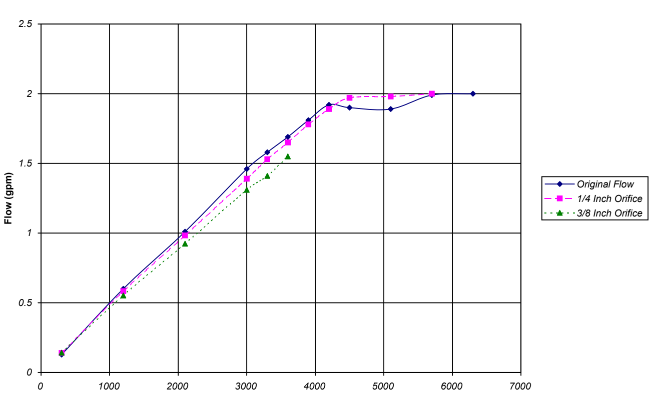

Figure 14. Comparison of Journal Bearing Flows.

Field Test

Similar results were attained when tests to supply driver bearings with oil were conducted on an actual pump in a manufacturing facility. The main goal of the test was to evaluate the ability of the oil circulation system to lubricate the driver and driven machine bearings with oil.

The pump shaft was supported by two 2.56 inch (64.9 mm) diameter by 1.90 inch (48.2 mm) long journal bearings. Similar to the test setup, the inboard or DE housing contains only a journal shell; the outboard or NDE housing contained the viscosity pump, the oil reservoir, a journal bearing, and a double 7.00 inch (177.8 mm) equalizing thrust bearing. An oil to water heat exchanger was mounted directly to the underside of the combined bearing housing. The field test setup is presented in Figure 15.

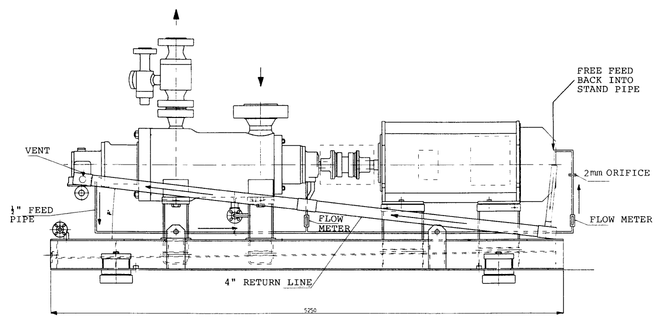

Figure 15. Field Test Rig.

The prime mover was a 550 kW (737.6 hp) electric motor. The motor was equipped with grease lubricated rolling element bearings. This was seen as an advantage in that the test unit was not at risk; it had undergone full performance testing and was scheduled for shipment.

The pump operates at 2980 rpm, and the journal bearings operate under a 1124 lb (5000 N) load.

Clear plastic tubing was fitted between the components for use as lubricant supply and drain lines. A single parallel branch was extended to the NDE of the motor. A 2 mm (0.079 inch) diameter orifice in the parallel line was used to simulate motor bearing oil flow. A clear standpipe allowed visual monitoring of the oil level as it would be in the bearing housing.

A 1/2 inch pipe was used for the oil supply line, a 2 inch pipe for the standpipe, and a 4 inch pipe was used for the return line to the reservoir. Approximately 18.5 gallons (70 liters) of ISO VG 46 was used to fill the system.

The bulk of the instrumentation was factory-equipped with the bearings and pump. Two flow meters were temporarily mounted in the oil supply line. One was positioned in the separate journal supply; the second was placed upstream of the orifice at the NDE of the drive motor. A pressure gauge was also installed in the end cover plate to measure the central feed pressure in the combination housing.

During the test, the sump temperature, standpipe and reservoir oil levels, oil flow, and oil supply pressures were monitored and manually recorded.

Performance data are presented in Table 1. The pump was started at 13:50 and set to rated conditions. The developed oil supply pressure was 11.0 psig (0.8 barg), with 3.0 lpm (0.8 gpm) delivered to the motor NDE, and 2.25 lpm (0.6 gpm) to the pump DE. By 13:55 the supply pressure had dropped to 5.0 psig (0.3 barg, delivering 2.5 lpm (0.7 gpm) to the motor NDE and 2.0 lpm (0.5 gpm) to the pump DE. At 14:10 the cooling water flow was stopped to determine the effect of increased oil temperature on both oil flow and oil level.

Upon reaching 150°F (66°C) at 14:30, cooling water was reapplied and the bearing temperatures were allowed to stabilize for approximately an hour before ending the test.

Table 1. Field Test Results.

Time	Supply Pressure (psig)	Reservoir Temperature (°F)	Motor NDE Flow (gpm)	Pump DE Flow (gpm)	Maximum Journal Bearing Temperature (°F)	Maximum Thrust Bearing Temperature (°F)
13:50	11.0	70	0.79	0.59	*	*
13:55	5.0	*	0.66	0.53	*	*
14:10	4.0	*	0.53	0.40	128.7	150.8
14:20	2.5	135	0.32	0.26	154.4	176.0
14:30	2.3	150	0.28	0.22	170.6	190.4
14:40	3.0	120	0.35	0.29	148.8	164.1
14:50	3.2	110	0.37	0.40	141.8	158.0
15:00	3.2	110	0.50	0.40	137.8	155.5
15:10	3.3	110	0.53	0.40	136.6	154.6
15:30	3.4	110	0.53	0.40	134.6	152.4

* – Data unavailable

During steady-state operation the oil level in the standpipe was measured to be between 1/4 to 3/8 inches (6.4 to 9.5 mm) higher than the reservoir. The oil levels increased uniformly with increased oil temperature, presumably from volumetric expansion.

As expected, while the oil is cool, the system supply pressure and oil flow are relatively high. As the oil is warmed and viscosity drops, the system supply pressure and flow also drop.

The test was considered a success for a number of reasons. Unlike work performed on the test stand, the field test did not provide the ability to restrict oil flow through the heat exchanger. The system was able to deliver more than the recommended amount of oil to the existing bearings during normal operation, while at the same time maintaining a steady supply of oil to the simulated bearing.

The test was accomplished with no leakage at any location, even though the 4.0 inch return line entered the reservoir through a 2.0 inch NPT fitting, which effectively served as an unwanted orifice. Also noted was the fact that the standpipe oil levels remained well below the projected shaft to seal interface. The levels measured during the test would not have led to leakage in the simulated bearings.

A concern voiced after the test was that the drive motor occasionally needs to operate uncoupled. This creates a problem if bearing lubrication is linked to pump rotation. A potential solution is the use of oiling rings, although they do have surface speed limitations.

A second solution would be to install a second circulator in the drive bearing housing, or perhaps mount the combination housing to the motor. This would allow operation of only the motor.

CONCLUSION

History has shown that self-contained, self-lubricating fluid-film bearings systems are capable of providing long-term trouble-free operation. Assembly oversights and misconceptions can and have limited successful system operation. With an understanding of how the system operates, these pitfalls can be easily avoided.

Both inhouse and field test results have verified that these systems are capable of providing more oil flow than the system bearings require at rated load. These same results have also shown that oil can be supplied and returned to additional remote journal bearings with minimal bearing system changes and negligible additional cost. The fact that the tests were conducted on units with essentially no internal oil flow path adjustment indicates that there is room to tailor system performance to meet specific needs.

NOMENCLATURE

P = Pressure (psig)
SG = Specific gravity
H_f = Head (feet)
DE = Drive end
NDE = Nondrive end

APPENDIX A

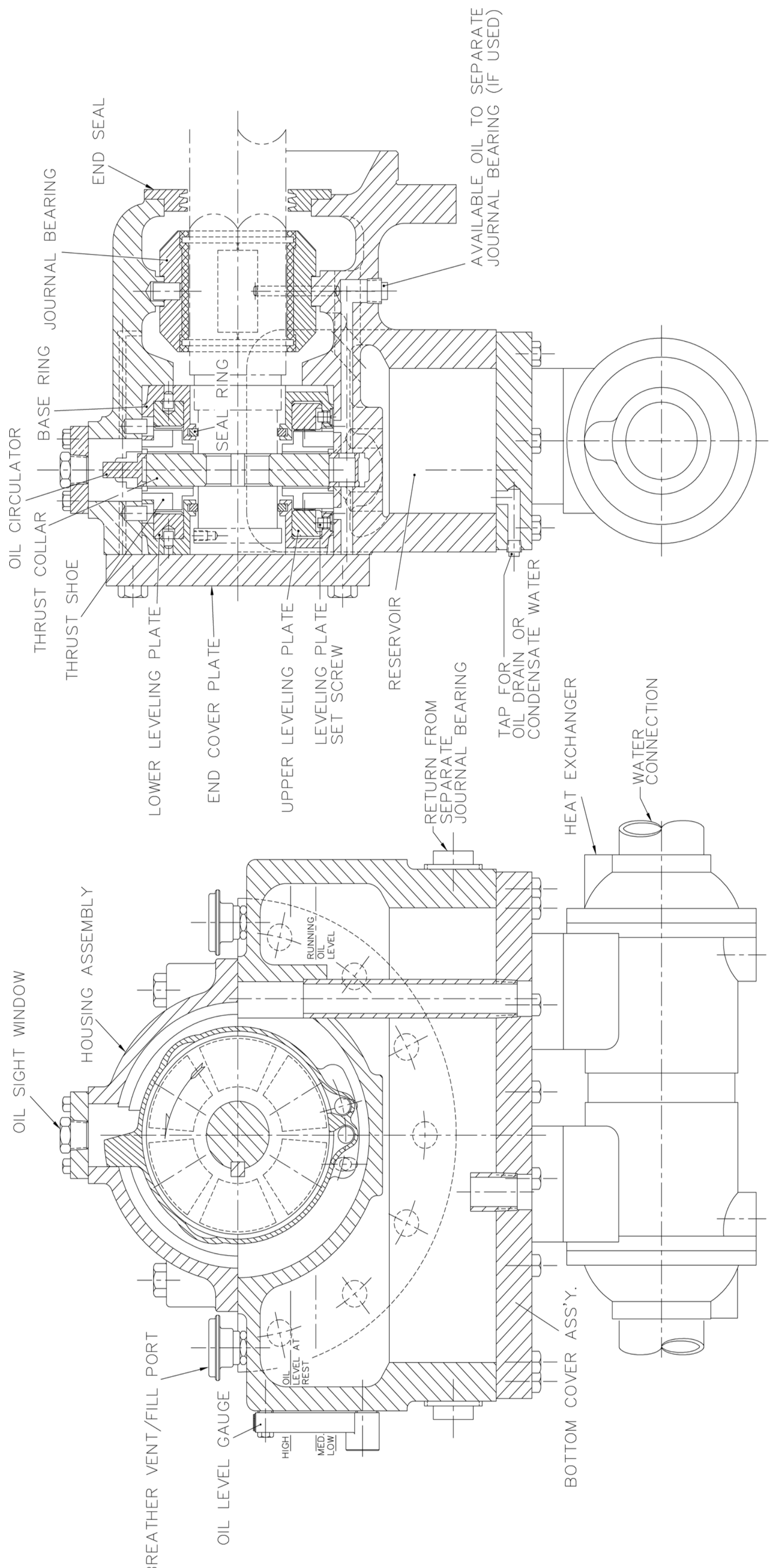

Figure A-1. Combination Housing Nomenclature.

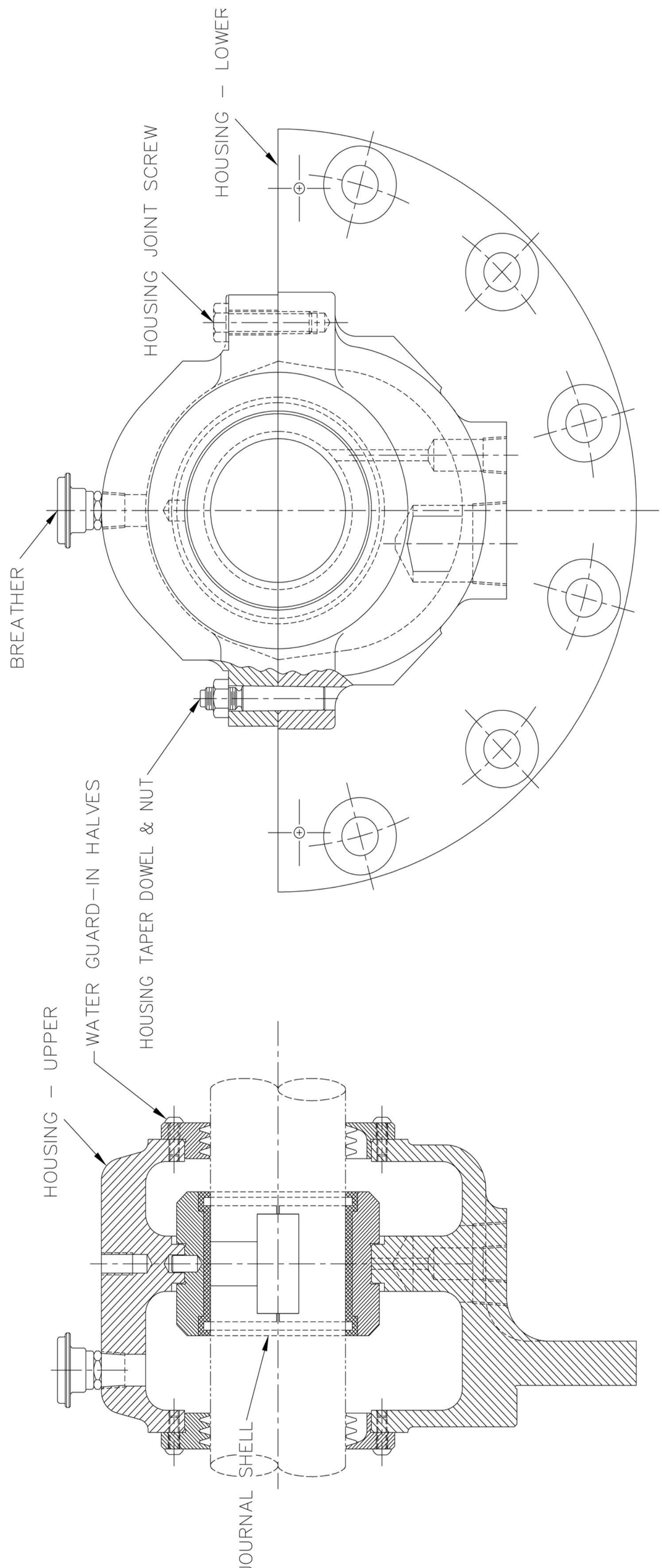

Figure A-2. Separate Journal Bearing Nomenclature.

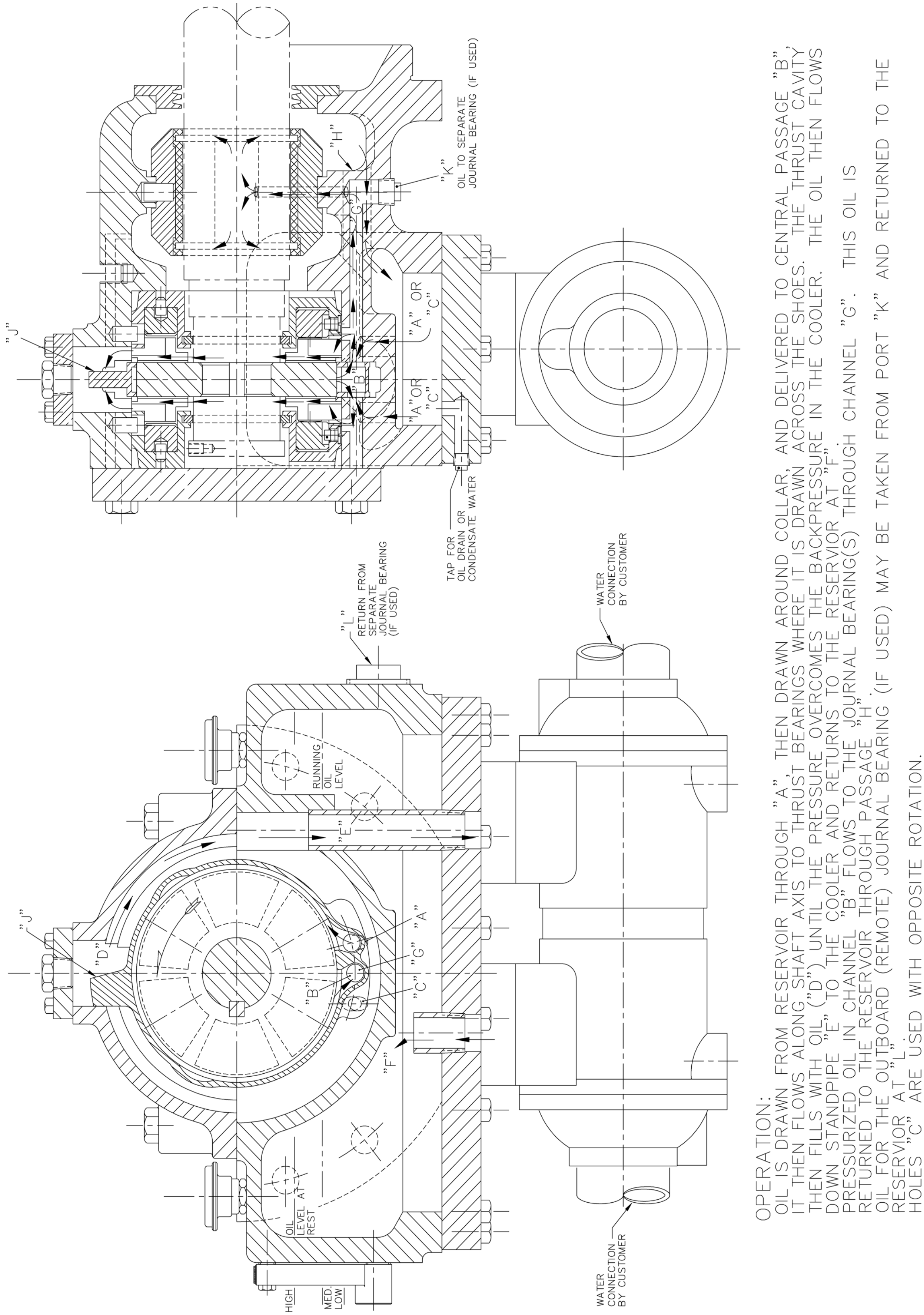

Figure A-3. Flow Path Detail through the Combination Bearing Assembly.

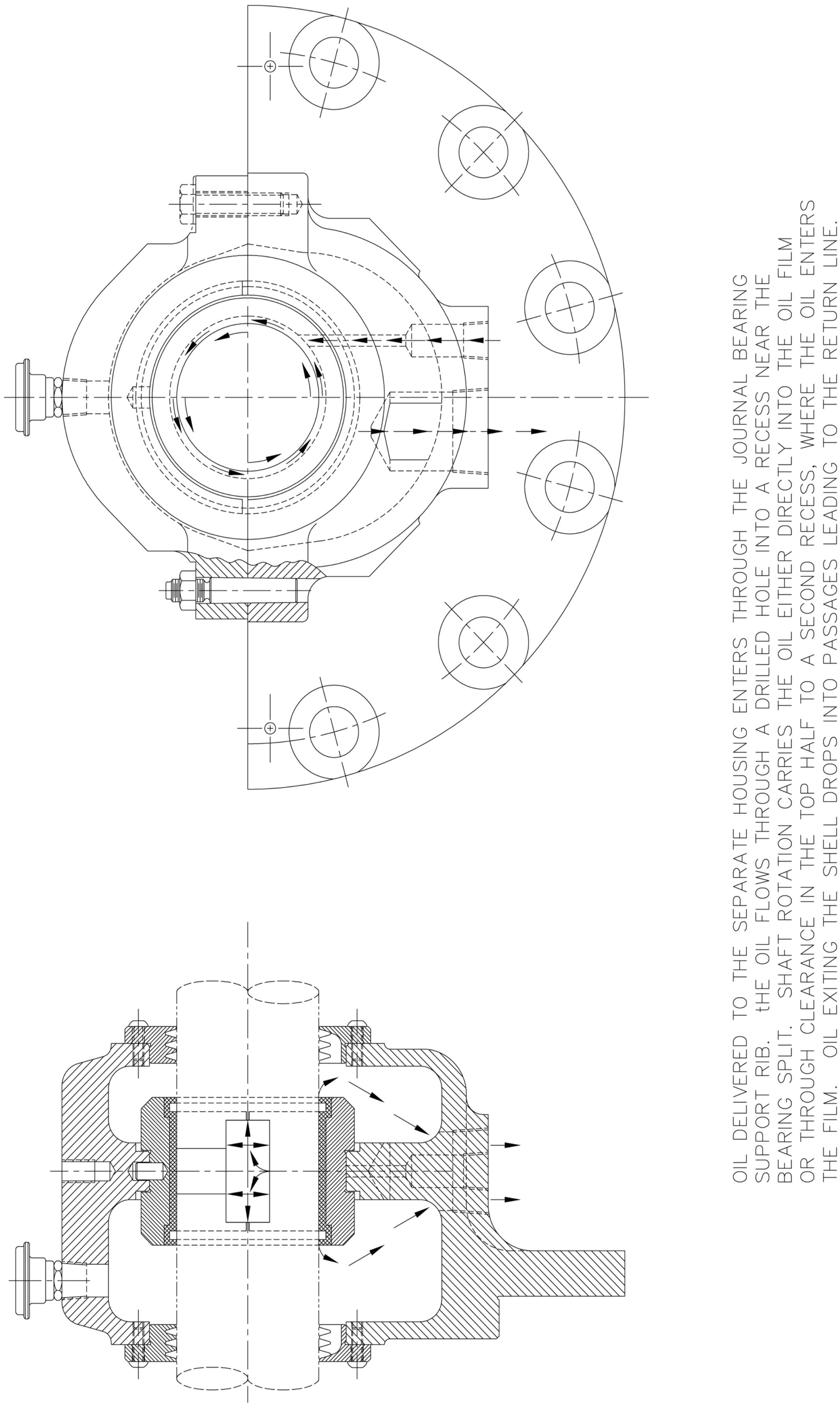

Figure A-4. Flow Path Detail through the Separate Journal Assembly.

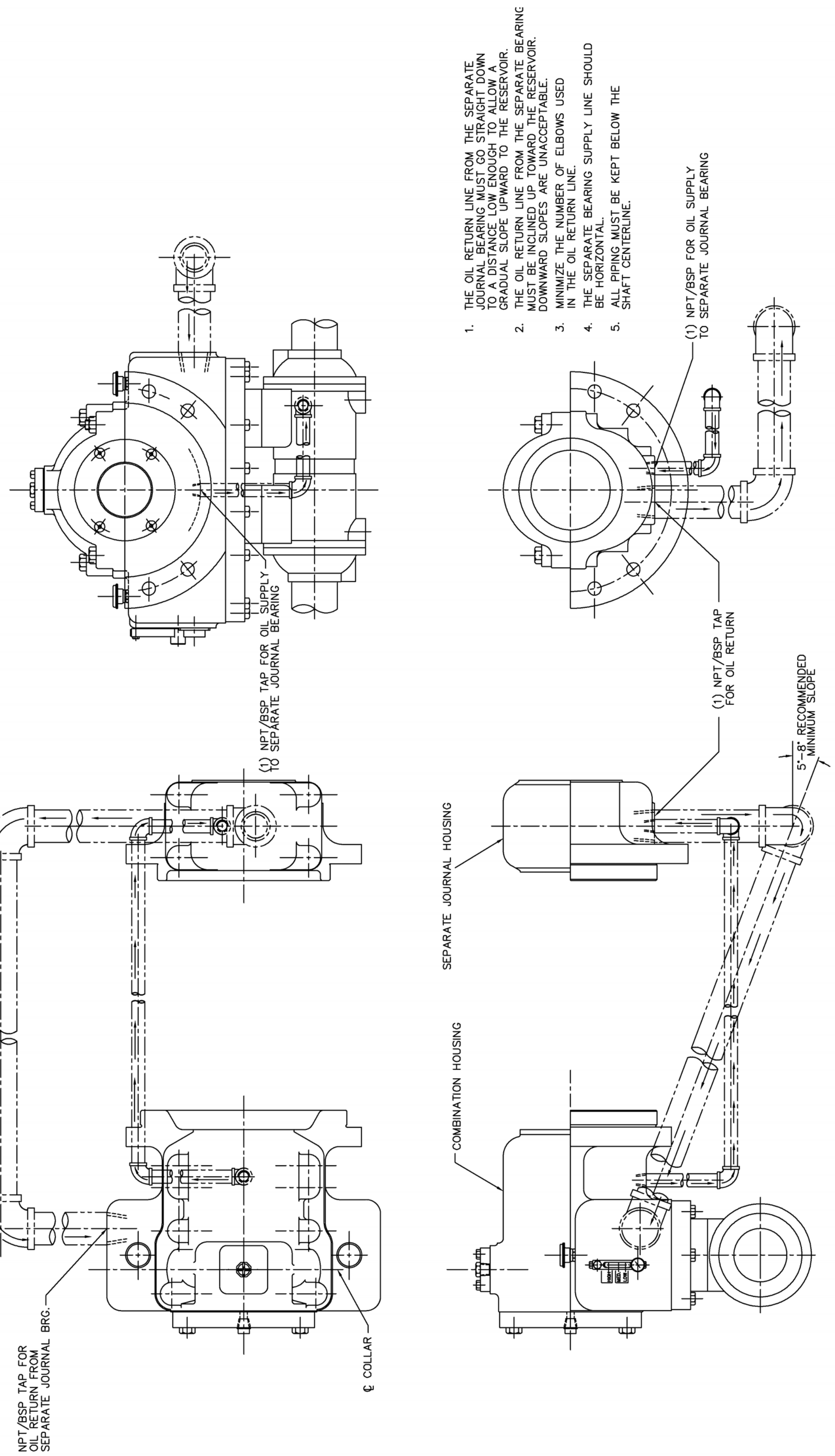

Figure A-5. Supply and Return Piping Detail.

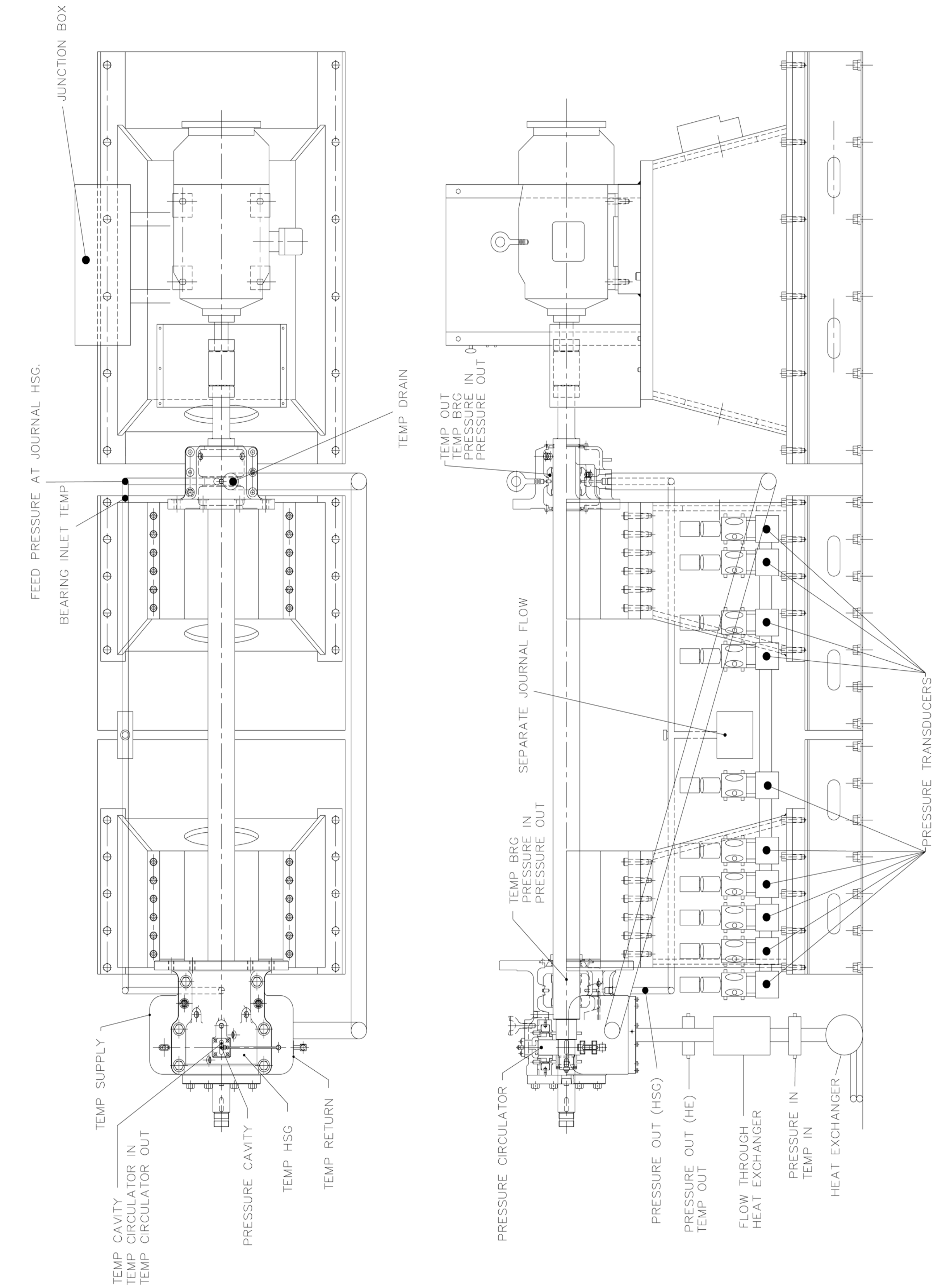

Figure A-6. Test Rig Instrumentation.

REFERENCES

"Cameron Hydraulic Data," 1984, Sixteenth Edition, Ingersoll Rand, Woodcliff Lake, New Jersey, pp. 2-14.

ACKNOWLEDGEMENT

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