

George Ponton
**MAINTENANCE REDUCTION for LUBE OIL
PUMPS**

ABSTRACT

The fleet is plagued with leaking lube oil pumps due to mechanical shaft seal failures. Correction programs have resulted in improved shaft sealing techniques however, there continues to be a leakage problem on most classes of ships.

The ultimate solution is to eliminate the mechanical shaft seal. This has been proven to be a very successful solution and has additional benefits of eliminating the motor to pump coupling, which further reduces maintenance.

The newly developed Canned Lube Oil Pump is the solution. It combines the proven screw pump design with the proven Oil-Through motor design. The resulting compact pumping unit is substantially smaller and lighter than the legacy unit. The motor is cooled by a portion of the oil, that is being pumped to the system. This eliminates the motor cooling fan and eliminates grease packed motor bearings, further reducing maintenance.

A comprehensive factory and shipboard testing program has demonstrated that this new unit will eliminate oil leaking from lube oil pumps, thus will reduce shipboard maintenance substantially.

INTRODUCTION

Lubricating Oil Pumps for Naval Ships are designed in accordance with the requisite Military Specification [Mil-P-18547,1963]. That specification has always been invoked in Naval Shipbuilding Programs Top Level Specifications.

Thus all USN War Ships constructed in the last 40 years have pumps installed in accordance with that specification, as modified for the particular program. The pumps are positive displacement rotary screw pumps of two types: Timed (type II) and untimed (type III).

The function of the pumps is to provide lubricating oil to the propulsion machinery for gearbox and turbine bearing lubrication and cooling.

The fluid being pumped is turbine/reduction gear lubricating oil [Mil-L-17331, 1992], commonly called 2190 TEP. The fluid is used for Main Reduction Gear Lubrication on all Steam and Gas Turbine Powered Ships, and in some hydraulic systems.

The requirement for mechanical shaft seals is stated in paragraph 3.6.14.7 of the pump Military Specification [Mil-P-18547] as follows:

“Mechanical shaft seals shall be used in all pumps intended for installation on ships having nuclear powered propulsion plants and for installation on other ships when specified”

There are no other requirements stated for the seals, such as materials, types, installation, etc.

A mechanical shaft seal is defined as: “A device designed to prevent the movement of fluid from one area to another, or to exclude contaminants” [Fluid Sealing Association, 1990]. In a Lubricating Oil Pump its function is to prevent the lube oil being pumped from leaking along the power rotor shaft and into the Machinery Room. The mechanical shaft seal is mounted on the pump shaft and consists of a stationary

sealing element and a rotating sealing element. These elements are also referred to as rings, faces and seats. The faces of the sealing elements have a very fine finish, measured in helium light bands. The faces are held in contact with a spring(s), which maintains pressure between the faces. Figure 1 shows a typical Mechanical Shaft Seal as used in Lube Oil Pumps to comply with the specification requirements.

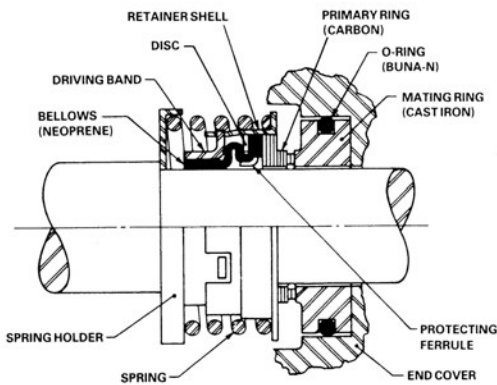


Fig. 1 Mechanical Shaft Seal Assembly

SEAL LEAKS AND FAILURES

Leaking Seals cause several problems including:

- Increased maintenance cleaning up leaked oil
- Hazardous conditions due to leaked oil
- Housekeeping in disposal of oil soaked rags
- Pier side costs involved with frequent seal replacement

The first formal documentation of problems with Lube Oil Pumps mechanical seals was a 1986 report [Failure Analysis of CG 47 Class Propulsion Lube Oil Pumps]. The report was basically a study of Lube Oil Pumps Casualty reports (CASREPTS) and 3M data on several classes of surface combatants. One significant statement in that report was: "Mechanical Shaft Seal failures are the single largest cause of pump failure reported".

Another significant report of the problems with Oil Pump seals was a 1987 study

[Analysis of IMO Pump mechanical shaft seal removed from Trident Class submarines] certainly implied problems with seals. That report documented 42 seal failures over four years. The Trident Hydraulic Pumps handle 2190 TEP the same fluid as Lube Oil Pumps. Unfortunately, the recommendations of that report were erroneous and suggested materials that are unsatisfactory with respect to seal face materials.

During this same time frame Naval Sea Systems Command [NAVSEA] established requirements for mechanical seals for Navy Pumps. In a memo that was issued in 1986 [SER 56Y21/267] several mechanical seal requirements were addressed including: Alternate Manufacturers, Backup Packing, Materials, Positioning, Liquid pressure and Circulation. This was a step in the right direction, the real goal being reduction in the number of different seals with which the Navy had to contend. The downside of this directive was it addressed centrifugal pump seals (water service) as if they were interchangeable with positive displacement pump seals (lube oil). This memo also excluded the face materials that have the best results with 2190TEP.

Serious Seal leaks occurred when the Main Lube Oil Pumps for a new support ship were operated at the building yard. These were the first Lube Oil Pumps that utilized seals in accordance with the NAVSEA guidance memo [SER 56Y21/267]. So by the end of the 80's it was well documented that the Mechanical Seal leakage/failure problem was getting worse, not better. Discussions involving allowable seal leaks ranged from 5 drops per hour to 5 drops per minute. What needed to be addressed was how to eliminate seal leaks so the criteria could become ZERO leakage.

Vertical Lube Oil Pump Seal Leak Investigation

In 1994 when the first ship of a new class of Combat Support Ships was nearing delivery,

the Main Lube Oil Pumps were a source of serious oil leakage. The mechanical shaft seals were failing during operational testing of the Lube Oil System.

The factory investigated the pumps still at the manufacturer, that were slated for delivery to a later ship of the class. These were the first pumps designed with the mechanical shaft seals per the new directive [SER 56Y21/267] which required tungsten carbide or silicon carbide stationary seats and F91 Carbon rotating seats.

The Mechanical seal installed in the Vertical Lube Oil Pump was a Type 21 Seal with a Tungsten Carbide stationary seat and a F91 carbon rotating seat. The stationary seat utilized an O-ring. The design was reviewed to verify that seal chamber design was proper for a 2 3/4 inch size seal including working length, PV limit, cooling flow, pressure, and venting. The seal arrangement/installation was also reviewed to verify proper procedures were being followed for installation and operation. The viscosity characteristics of 2190 TEP oil were also reviewed. This oil has a viscosity range from 100 SSU to 4000 SSU, as shown in Figure 2.

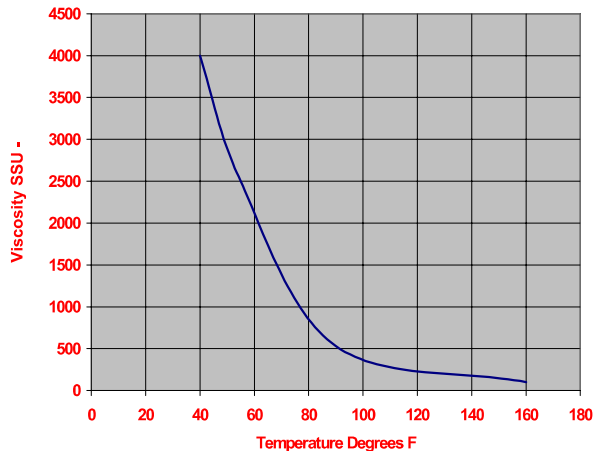


Fig. 2 Viscosity of 2190 TEP

Since the design and installation were determined to be proper, a test program was initiated to obtain technical information as to the cause(s) of seal failures. This included a

thermal couple installed in the stationary seat to measure the seal face temperature. The normal operating temperature is in the range of 120 to 160 degrees F, however the cold oil start up was more of a concern. So several tests were run with oil at temperatures from 55 degrees to 130 degrees, at low and high speed. The seals started leaking with in one to five hours of start up. Inspection of the seal faces showed blistering on the carbon faces, a sign of high heat. Flow rates through the seal chamber were 0.22 GPM cold to 0.85 GPM hot, temperature rises of the seal faces were very rapid. Figure 3 is a graph of temperature rise with respect to time.

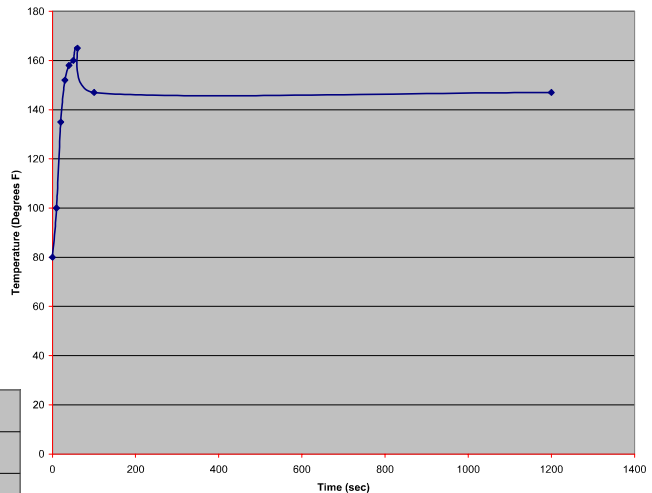


Fig. 3 Seal Face Temperature Rise

As a result of the tests it was determined that there were several principle causes of seal failure:

1. Inadequate oil flow through the seal chamber to cool and lubricate the seal faces.
2. The hard tungsten carbide stationary seat not retaining an oil film and not conducting heat away from the seal faces.
3. Air was accumulating in the seal chamber, the highest point in a vertical pump.
4. Cold start was causing the Carbon Seat to blister.

5. Cold oil (high viscosity) was slow to reach seal faces.

As a result of testing various combinations of materials, the best results were obtained by replacing the tungsten carbide seat with Ni-Resist and increasing the flow rate to 1.5 GPM. When fitted with this design, the pumps operated for 100 hours with no visible sign of leakage.

Aboard ship two changes were made:

1. A vent line was run from the seal chamber to the oil sump to allow the seal chamber to be continuously vented
2. A line was run from the purifier to the seal chamber to supply oil to the seal during pump start up.

Lessons learned from this program have been applied to other Vertical Oil Pumps on other ship classes with good success in reducing and eliminating seal leaks.

Vertical Hydraulic Pumps Seal Leaks Investigation/Correction

Hydraulic pumps also use 2190 TEP oil; the major difference is the high pressure developed by hydraulic pumps. While Lube Oil Pumps operate in the range of 50 psig to 130 psig discharge pressure, Hydraulic pumps operate in the range of 200 psig to 3000 psig discharge pressure. This causes the pump designer to handle the seal chamber design differently. In a screw pump the oil flow to the seal chamber is controlled by the annular area between the balance piston and the balance piston bushing. The pressure in the seal chamber is limited by the differential pressure across the balance piston and the seal chamber return line to pump suction. Therefore the pump designer sizes the flow area to provide the Seal Manufacturer's required flow rate and limit the maximum seal chamber pressure. Of course the designer bases his calculations at the highest pressure and normal operating temperature.

When the screw pump manufacturer started the investigation into the excessively high numbers of seal leaks and seal failures in hydraulic pumps, the design criteria was reviewed and verified. Since the pumps were designed for normal operating conditions, the investigation moved into the area of alternate seat materials.

Over 15 different combinations of seat materials and seal types were tested from several different seal manufacturers. Test results were poor with seal leaks occurring within 30 minutes to 3 hours after starting each combination of seat materials.

Inspection of the carbon stationary seats consistently showed blistering which was attributed to high heat.

This led the investigating Engineers to challenge whether the seal manufacturers recommended flow rate to the seal faces was adequate. As it turned out the seal industry flows were based on seals used in water service. Oil is a poorer conductor of heat than water, in fact the coefficient of heat transfer for oil is 0.47 (water is 1.0). So the next step was to double the flow rate to the seal chamber and run additional tests.

At this point the investigation took two courses: one for high pressure-low flow hydraulic pumps used on Submarines and the other for moderate pressure-medium flow hydraulic pumps used on Surface Ships.

The Surface Ship seal leakage turned out to be a simple one to solve, basically the pumps were being operated at conditions much different than the pump design conditions. The pumps were being operated at very low pressure most of the time and the attached (driven from the main reduction gear) were operated at very slow speeds. Calculations and actual tests showed that when the pumps were operated at low pressure and low speed there was no flow to the seal chamber. Flow is caused by the differential pressure across the balance piston. When the pressure is low (300 vs. 1000 psig) there was little or no flow. Also, when the pump is running slowly (200 vs. 1800 rpm) there was no flow. The solution was to provide additional oil flow to the seal

chamber (install an oil passage in parallel to the balance piston) and to reduce the backpressure in the seal chamber return line to pump suction. These two simple modifications were proven at the pump manufacturers factory and were retrofitted in a DDG, for underway testing. During the test period there was no observed oil leakage from any of the modified pumps.

These pumps originally used ductile Ni-Resist (ductile nickel cast iron) J seats rather than O ring stationary seats. The J seat has superior heat exchange transfer properties and the NI-Resist material has good oil retention characteristics.

The Submarine hydraulic pump leakage proved to be much more difficult to solve. The immediate problem was that substantial additional oil to cool the seal was not available. The solution required some creative engineering and design by the investigating engineers. It actually was two complementary solutions in one fix. Increase the oil flow during cold conditions and low pressure operation, and eliminate the laminar flow in the seal chamber to increase heat exchange away from the seal faces. Again when this new design was proven in the pump factory, it was installed in an SSN. During the underway evaluation period the pump did not exhibit any leakage.

SEAL LEAK CORRECTION

It is now known what needs to be done to correct leaking shaft seals.

The problem then becomes the availability of resources to correct the many leaking pumps in the fleet. Installing modification kits, adding seal chamber supply and vent lines and of course correcting documentation that goes with the hardware changes so the modified pumps are supportable, is expensive and time consuming.

For new programs a Marine Pump Seal Standard [ASTM F1511] has been developed and updated to include current materials for Lube Oil Pump Seals, and this

Standard has been required by most recent Shipbuilding Specifications. Still, shaft seals will wear out, the statistical design life of the seal carbon face of 16,000 hours is really not that long. A pump running continuously on a six-month deployment will accumulate 4,368 hours. This means that in theory a seal will last for just over 3 deployments. And, in spite of the operators best efforts it will probably start leaking sooner. Then the seal will need to be replaced involving system tag outs, interfering equipment removal/replacement and the Electric motor removal/replacement. These are all maintenance hours better spent on other equipment.

This situation begs the question: how can the seal be eliminated completely?

SEAL LESS INDUSTRIAL PUMPS

Seal less centrifugal pumps have existed for years and are used to pump toxic liquids, radioactive wastewater, and other liquids that pose serious risks if the pumped fluid escapes from the system.

Centrifugal pumps are not suitable for pumping viscous fluids when the viscosity exceeds 500 SSU, so for those applications rotary positive displacement pumps are utilized. Screw pumps are preferred due to their constant flow and high suction capability when handling viscous fluids like Lube Oil.

In 1995 a screw pump manufacturer completed development of the worlds first “canned rotary/screw pump” package. [Pumps and Systems January 1997]. Figure 4 is a cross section of this package.

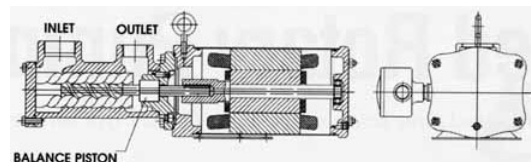


Fig. 4 Commercial Canned Screw Pump

Three screw pumps were a natural choice since they are versatile and can be directly

driven by electric motors. This eliminates speed reducers between the pump and motor. A small portion of the fluid being pumped passes from the discharge end of the pump to the motor, circulates through the motor and then returns to the pumps suction end through the motor shaft and power rotor shaft which are drilled.

All joints exposed to the liquid being pumped are O ring sealed, so the unit is sealed, hence the name: canned.

Most importantly, there is no mechanical shaft seal!

The motor rotor is support by an open ball bearing at the none-drive end and by the balance piston bushing at the drive end. The pumped fluid cools the motor and lubricates the ball bearing. The motor casing contains the pumped liquid at pump inlet pressure. Since the stator winding insulation is exposed to the liquid it must be chemically compatible with the pumped fluid. This is called a wet stator design.

These initial production units were available in three motor sizes with flows to 50 gpm and pressures to 150 psig and viscosity range from 32 to 1500 ssu. The original industrial applications were for refrigeration compressor lubrication and power cable cooling, both applications were closed loop systems with high inlet pressure where unacceptable leakage occurred on the original pumps with conventional shaft seals.

SEAL LESS SCREW PUMPS FOR NAVAL APPLICATIONS

Based on the success of the industrial seal less pumps installed for lubricating oil service in industrial refrigeration compressors the development of a unit for a Navy Service Refrigeration Compressor was undertaken. The basic performance requirements were:

Capacity	15 GPM
Discharge Pressure	125 PSIG
Inlet Pressure	50 PSIG
Viscosity Range	55 to 230 SSU
Fluid	ICI 32

In order to meet these conditions a screw pump rotor size of 1.43 was selected operating at 1750 RPM and requiring a two HP motor.

The basic industrial design (commercial off the shelf) was reviewed and it became obvious that it would not meet the Navy high shock requirements. A redesign was undertaken with the major change being a new mounting foot constructed of steel plate with appropriate “beefed up” hardware. Other changes included the materials of principle components and hardware items as well as the new piping connections (e.g., tapered pipe threads are not allowed). The new design consisted of components manufactured by the pump manufacturer and components manufactured and furnished by the electric motor manufacturer. The motor components were the rotor, stator and housing (can) with conduit box. The unit was assembled and tested by the pump manufacturer. The unit was subjected to a shock test with some minor discrepancies occurring which were corrected by redesign. Upon entering into production some leakage problems occurred at the conduit box where the motor leads pass through the Can. The original design utilized a soft pass through and due to the fact the unit was pressurized the pass through seal was very difficult to control.

The solution was a mechanical pass through which when properly installed solved the leakage at the conduit box.

Over one hundred of these small units have been installed in Refrigerant Compressors utilized on several different classes of ships.

PROTOTYPE CANNED LUBRICATING OIL PUMP UNIT

Interest by the Atlantic Fleet maintenance engineers in reducing maintenance on Main Lube Oil Pumps, mainly frequent mechanical seal replacements, lead to the design of a large Canned Lube Pump (CLP). The basic design performance requirements

were 1000 GPM at 70 PSIG Discharge pressure and 10 inches of mercury inlet. Most importantly the CLP unit had to be interchangeable with the existing units with respect to form, fit and function.

The basic unit design was established as a 4.625 rotor size, double end configuration with a 1.6 D screw lead and of course three idler rotors to handle air entrained oil.

The decision was made to utilize a two speed motor to supplement the attached pumps, as is done on several classes of surface ships that have one attached pump and two motor driven pumps per reduction gear. A single speed motor would create conditions of excessive flow at some ship operating speeds. A complete discussion of lubricating oil pumps and systems is presented in a SNAME Paper "Marine Lubrication Systems" [Ponton 1985].

Due to the many challenges of the small CLP discussed earlier, when utilizing several components and assembling them into a unit, it was decided to purchase the motor as a complete package. The pump and motor would then be assembled to form a unit. This proved to be a very desirable arrangement.

Cooling of the motor utilizing the oil being pumped, instead of air, required some very thoughtful design effort. The oil being pumped could reach 160 degrees F. The motor would be a 75/37.5 Horsepower machine. If a Totally Enclosed Fan Cooled (TEFC) machine was selected per the Electric Motor military specification [MIL-M-17060], it would be a frame size 445. If the oil being pumped was used to cool the motor with a temperature range of 120 to 130 degrees F, the motor frame size could be reduced to 365. A supply of oil could be provided by connecting into the ships main lube oil supply header into the reduction gear, downstream of the filter and cooler. It was determined that 10 GPM of this lower temperature (120 to 130 degree) oil was available and would provide ample cooling flow.

Connecting the motor output shaft to the pump power rotor shaft with out a coupling was accomplished with a spline. The power

rotor shaft has a male spline and the motor rotor shaft has a female spline. This makes for a very simple assembly since the ends of the shafts are not visible when the unit is being assembled. The motor to pump alignment is controlled by accurately controlled rabbit fits so proper alignment is assured.

The motor was provided with a small o-ring sealed removable cover over the shaft so direction of rotation could be verified and the motor rotor could be hand rotated for ease of assembly to the pump.

Since the motor stator would be immersed in oil the insulation sealing system was critical. The motor military specification [Mil-M-17060] requires a vacuum pressure impregnated (VPI) epoxy insulation system.

This requirement is to protect the electric motor windings in case of a Machinery Space flooding occurred and the motors became immersed in seawater. This same system was considered suitable for oil and utilized for the "Oil-Through" motor. A Motorette (section of the stator) was immersed in 2190 TEP oil for years and the VPI system proved satisfactory.

Upon working out the many design details between the pump manufacturer and motor manufacturer the prototype entered production in late 2000. The arrangement of this CLP unit is shown in Figure 5.

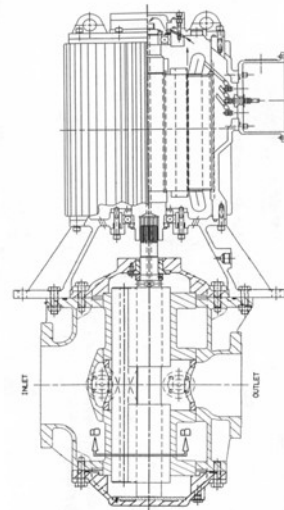


Fig. 5 1000 GPM Prototype CLP

In 2001 the prototype motor was tested at the motor manufacturer in air. The number of tests were limited since there was no way to load the motor. The motor was then installed on the pump and the test program was begun. The first article tests in the applicable Military Specifications [MIL-P-18547] and [MIL-M-17060] were performed, except for those having to do with air cooling of the motor or water content of the oil. Many other tests were performed at the factory, an interesting test was to drain the oil from the motor and determine the time for the motor to fill. This would be in the unusual situation where the unit was started with out first filling it with oil. The motor bearings were instrumented for this test to monitor temperature and vibration which did not vary during the test. After the unit had accumulated approximately 1000 hours, including the 500 hour endurance test, the pump was disassembled and inspected. There was not any wear on the rotors or in the housing bores indicating an expected 50 year life when pumping clean filtered oil.

The ship class this unit was destined for utilizes a lubricating oil system with three pumps per Plant. The Attached Lube Oil Pump (ALOP) is driven directly from the Main Reduction Gear. At low speeds it provides insufficient flow to the Gear, so the Standby Lube Oil Pump (SLOP) which is steam turbine driven, supplements the ALOP. The SLOP runs at high speed during plant start up, and then slows as the ALOP speed increases, providing a constant flow of oil to the Gear. This is controlled by a pressure sensor at the oil inlet to the Gear. At high shaft speeds the ALOP provides all the flow necessary to the Gear. The Emergency Lube Oil Pump (ELOP) is only used when the SLOP is unavailable. A two speed motor driven pump can not duplicate the performance of a turbine driven pump, however it has proven satisfactory on many ship classes. During the testing program a variable frequency drive controller became available. The CLP was tested with the drive and using a small pump that could simulate a pressure change,

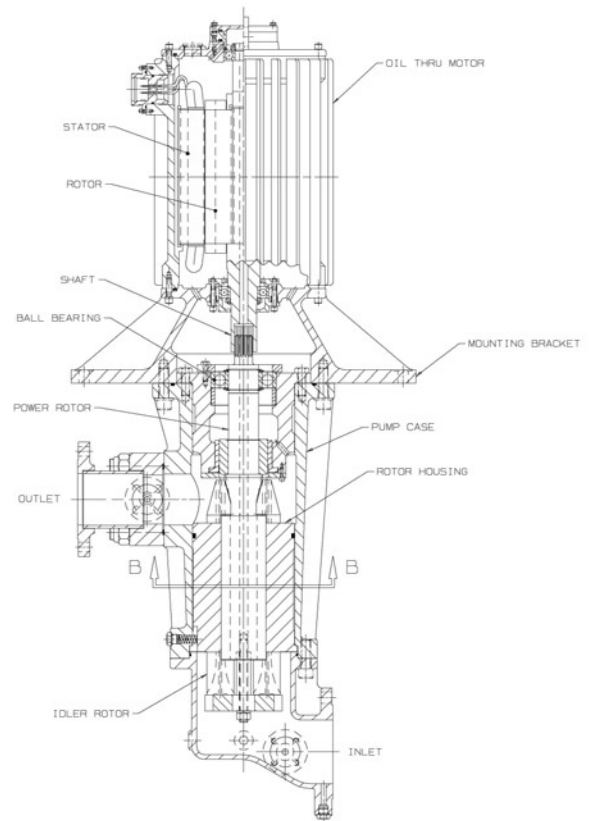


Fig. 6 300 GPM CLP

the speed of the CLP was varied based on a pressure input signal to the control system. This testing demonstrated that the CLP could be controlled the same as a variable speed turbine driver, which it is destined to replace.

In summary, the CLP unit performed all factory first article tests with out incident or problems.

Testing performed includes the Grade A shock test [Mil-S-901] and the environmental vibration test [Mil-Std-167]. A major advantage of this CLP is weight reduction, the legacy ELOP weighs 3,900 pounds and the CLP weighs 2,400 pounds, a difference of 1,500 pounds each.

PRODUCTION CLP FOR RETROFIT

Modernization plans of an older class of Amphibious ships included replacing the Turbine driven Main Lube Oil Pumps with motor driven units. Based on the successful testing on the 1,000 GPM CLP it was decided to utilize a CLP for this application. The requirements were 300 GPM at 50 PSIG which required a 20 HP motor. The CLP had to be identical to the existing pumps with respect to inlet and outlet flange sizes and locations. The pump selected was a single end screw pump per MIL-P-18547 with a 3.5 inch rotor size, a standard 2D lead and three idler rotors for handling entrained air. A two speed Oil-Through motor was selected so the pump could supplement the attached pump with minimum unloading of excessive oil to the sump tank at higher speeds. In this design a small flow of oil, for motor cooling and bearing lubrication, is fed from the pump discharge to the motor and then returned to the suction side of the pump through the rifle drilled motor and pump shafts. The pump and motor shafts are splined for ease of connection and the bracket between the pump and motor is machined with rabbit fits to assure proper alignment during assembly and operation. A mechanical pass through is used for the motor lead seals through the motor case. Figure 6 is a cross section assembly of the 300 GPM CLP. Upon completion of the factory performance, shock and vibration Type I tests, the first shipset of units will be installed in early 2004.

CONCLUSION

Maintenance reduction of screw pumps handling Lubricating Oil and Hydraulic Oil can be achieved by eliminating oil leakage from mechanical shaft seals. Seal leaks are the major cause of maintenance on oil pumps. Seal conversion kits have been developed and proven that will eliminate seal leaks, allowing the existing seals to operate for their design life. Seal less

pumps have also been developed and proven that completely eliminate the seal, and as an added benefit eliminate the grease packed coupling between pump and motor.

RECOMMENDATIONS

Existing Fleet Assets

Pumps that are in good condition can be retrofitted with a seal conversion kit that includes all wearing parts and provides increased cooling and lubrication to the seal faces. The price of the kit is approximately 10% of the pump replacement value, installation takes 2 to 8 hours depending on the unit.

Assets undergoing overhaul pumps that are “in the shop” for overhaul or repair should have the kit installed during the overhaul process. Since the pump has all ready been removed from the ship, the additional labor expense involved is minimal. If the motor needs overhaul a CLP should be considered.

Conversion Programs

Assets that are going to be changed from turbine drive to motor drive should be replaced with a CLP. There is not any price difference between a new pump (with a mechanical seal) equipped with an air cooled motor and a CLP of the same size with identical form, fit and function. There is also the advantage of weight reduction with the CLP.

New Construction Programs

New Motor Driven Screw Pumps should be Specified as Seal Less, without shaft couplings. The maintenance reduction over 50 years would approach 1,000 hours per installed CLP. The weight savings on the larger size units would be 2,000 pounds per unit.

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BIOGRAPHY

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