TWO-PHASE FLOW PERFORMANCE OF A HEAT TRANSPORT PUMP

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ABSTRACT

Heat transport pumps in a Canadian deuterium uranium reactor circulate coolant to remove fuel decay heat from the reactor core during normal and accident conditions. A loss-of-coolant-accident is one condition where heat transport pumps play a major role in the safe shut down of the reactor. They must operate under adverse conditions of low suction pressures and two-phase flows for at least 17 minutes. Analytical methods and model testing are no longer acceptable to show this capability. The heat transport pumps for a new reactor have been qualified by full-scale testing under simulated field conditions. The test's scope covered both vibration and pressure pulsation behavior of the pump under low suction pressures and two-phase flow conditions. Despite high vibrations, shaft runouts, and high pressure pulsations during the loss-of-coolant accident tests, the pump bearing, mechanical seal, and motor bearings performed without any significant damage. The test loop geometry and the $NSPH_{avl}$ were found to significantly influence the vibration and pressure pulsation behavior of the heat transport pump.

INTRODUCTION

A CANadian Deuterium Uranium (CANDU) 6 reactor has four heat transfer (HT) pumps circulating heavy water (D_2O) coolant at 511°F which remove decay heat from the reactor core during normal and accident conditions. A loss-of-coolant accident (LOCA) is one condition under which HT pumps are required to operate. The LOCA may be caused by a pipe break or the failure of a valve in the main coolant system. HT pumps are required to operate and remove the decay heat from the reactor core for 17 min before safe reactor shutdown is initiated.

Past analytical methods used to demonstrate the operational integrity of HT pumps under LOCA conditions are not accurate nor reliable because they are based on erroneous assumptions. Details of current analytical methods used for qualification of pump-motor sets are discussed in the literature [1].

Some success has been achieved with tests using scaled down models of the pump-motor set. These models must accurately simulate pump specific speed and other hydraulic conditions of the full-scale pump-motor set.

The most accurate data are obtained from tests using full-scale models of the pump-motor set. Reliability of data is dependent on simulation of actual operating conditions and on accuracy of measurement. Results are also affected by test loop piping geometry, water level ($NPSH_{avl}$), and volume of water above the pump impeller center line.

DESIGN FEATURES OF HT PUMPS

The pumps comply with the ASME Code Section III Class 1 and Canadian Standards Association Code Z299.1 Quality Standards. HT pumps are vertical, single stage, single suction, double discharge, twin volute, centrifugal pumps (Figure 1). Discharge nozzles are located diametrically opposite each other, connected to 90 degree long radius elbows. The stuffing box, located above the pump volute, contains the pump bearing, the mechanical seal package, and an auxiliary impeller for circulation of seal water. At the rated operating point, which corresponds to the best efficiency flow, the HT pump delivers 35300 usgpm, with a head of 705 ft, at a nominal speed of 1800 rpm. The $NPSH_{req}$ under normal operating conditions is 250 ft. The pump specific speed is 2450 (US units) and the suction specific speed corresponds to 5334 (US units). The design diameter of the impeller is 31 in and the impeller eye diameter is 20.75 in. The impeller eye peripheral velocity U_{eye} is 163 ft/s.

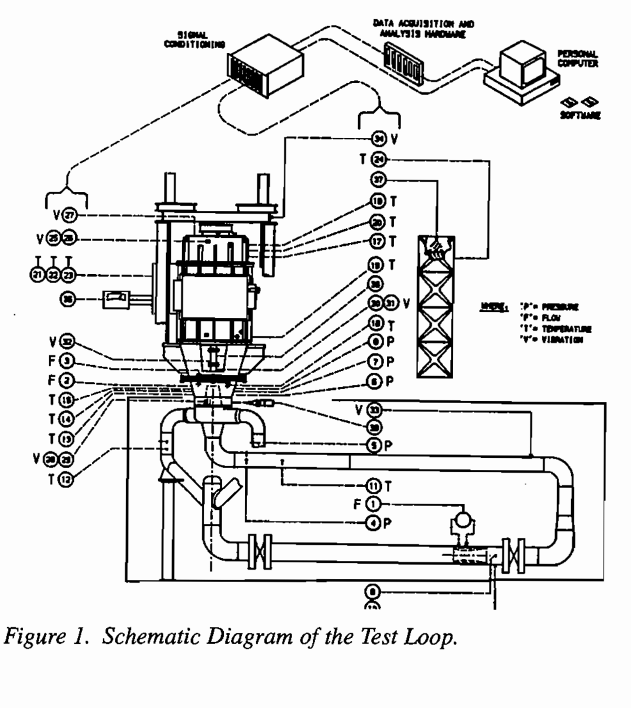

Figure 1. Schematic Diagram of the Test Loop.

Located above the pump impeller is a hydrodynamic carbon-graphite bearing (Figure 2), contained within a stainless steel housing. The housing is free to rotate about a vertical axis. The bearing is lubricated by D_2O at 170°F with a specific gravity of 1.001.

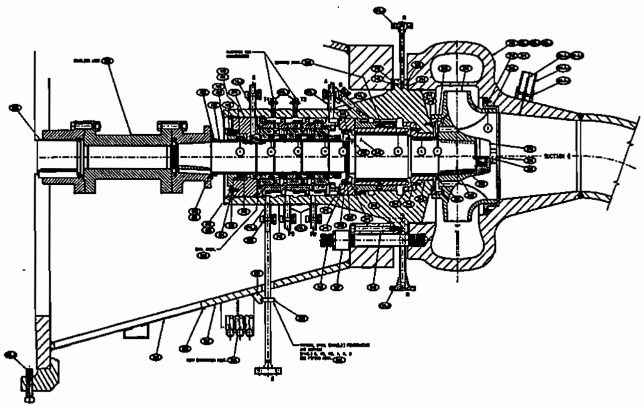

Figure 2. PHT Pump Cross Section.

The mechanical seal package (Figure 2) consists of three stages in series. Under normal operating conditions, each stage carries one-third of the system pressure. The seals are supplied with external seal injection at 170°F and 50 psi above pump suction pressure. This excludes hot water at 511°F from entering the seal package from the pump. The external seal injection is connected directly to the pump's internal recirculation loop. When seal injection is lost during a LOCA, the pump bearing and seals will be adequately lubricated by the residual liquid from this loop and can operate for at least 30 min satisfactorily ensuring that the seals and bearing do not run dry. A tube-in-tube type heat exchanger provides cooling to dissipate heat generated by the pump bearing and seals.

The HT pump is driven by a 9000 hp electric motor, directly coupled to the pump via a rigid, spacer-type coupling. Located at the top and bottom ends of the motor are tilting pad guide bearings that carry radial loads from the pump and the motor. A tilting pad bearing carries the axial pump-motor thrust loads.

The motor is mounted on a driver stand (Figure 3) which is bolted onto the lower end of the pump volute. The motor is also supported by two hanger rods connected to spring supports. Three seismic supports, located at 120 degree intervals around the motor frame, restrict lateral movement of the motor frame.

THE TEST LOOP

Mechanical Simulation

To obtain reliable test data, it is critical to closely simulate reactor site conditions. Exact simulation was not possible because testing was performed in an existing test loop, not in a test loop built specifically for LOCA testing.

The pump volute was identical to the volute supplied to the reactor site. The only exception was the orientation of the discharge elbows. The angle between the plane passing through the discharge elbows and the plane passing through the suction elbow was 35.5 degrees at the test site, compared to 90 degrees at the reactor site.

The structural stiffnesses were verified using a finite element analysis method. Natural frequencies of the pump-motor set, the piping, and the support system of the test model were similar to field conditions. The results of the test model were compared with actual field measurements and the results were very favorable.

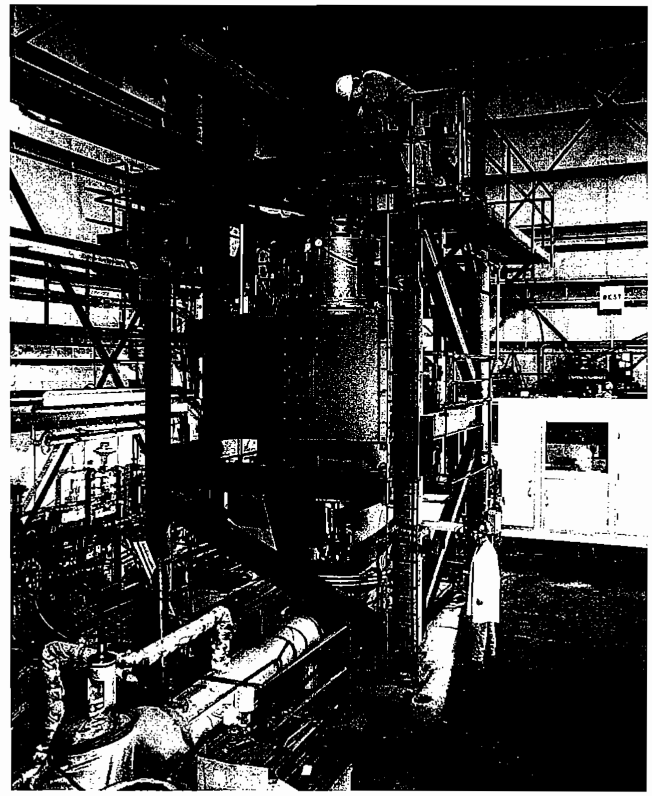

Figure 3. PHT Pump and Motor in the Test Loop at SBPI.

Hydraulic Simulation

The pressure pulsations generated by the pump is modulated by the system hydraulic characteristics. The amplitude of the pulsations could be amplified if the acoustic natural frequency corresponds to the excitation frequency. Therefore, in order to obtain meaningful results in the test loop, it requires simulating identical acoustic resonances in both field and test systems. Because attainment of this state is not realistic, only the simulation of the excitation source, i.e., the pump ("near field" [2]) was duplicated. This ensures accurate results in terms of acoustic source strength assessment. Acoustic magnification effects caused by system resonances, which are loop dependent, could not be compared or simulated.

To accurately simulate pump suction conditions, the head of water and volume above the impeller center line, in other words the $NPSH_{avl}$, should be similar to that found under field conditions. Because testing was performed on an existing test loop, such simulation was not possible. However, since the $NPSH_{avl}$ during the LOCA test was much less than what is normally available at site, the test conditions are much more severe than actual field conditions.

Instrumentation

A high speed, data-acquisition system was used to gather data during the tests. A total of 38 channels of data were monitored. Dynamic signals: vibrations, shaft runouts, pressure pulsations, and airborne noise, were gathered at 2000 Hz. The balance of the signals: pressures, temperatures, flows, horsepower, and LAV (loop average void) fractions, were gathered at one Hz. Placement of the transducers and detectors is depicted in Figure 1.

Loop Average Void (LAV) Fraction

The LAV fraction is defined as the total volume of voids present in the entire loop divided by the total volume of the loop. The liquid removed from the loop is collected in the vertical tank shown in Figure 1, and the volume in the tank at any given time corresponds to the volume of voids in the loop. The total volume of liquid in the loop is predetermined by calculations.

LOCA SIMULATION

It is not practical to test HT pumps under all possible LOCA conditions (void fractions and temperatures). Therefore, the pump was tested for the 17 min under the worst LOCA condition (the point at which the highest vibrations in the pump-motor frame and shaft are produced). The pump flow and head at the start of the tests was set at the normal single-phase rated conditions.

When the system is depressurized during test simulation, pump suction conditions go through two phases. During Phase 1, suction pressure decreases from the normal value to the sum of vapor pressure and $NPSH_{avl}$. The loop average void (LAV) fraction remains at zero percent. During this phase, voids are created locally at the inlet of the impeller and disappear at the discharge of the pump. Both loop temperature and suction pressure affect pump behavior.

During Phase 2, two-phase flow conditions at the pump suction and discharge occurs as liquid is removed from the loop. The LAV fraction range is zero percent to 60 percent. Pump behavior is affected by temperature and LAV fraction.

Under normal single-phase operation, the temperature of coolant is 511°F and the LAV fraction zero percent. Immediately following a LOCA, there is a broad spectrum of temperatures (212°F to 511°F) and void fractions (zero percent to 100 percent) under which the worst LOCA conditions can occur. To reduce the number of tests, the range of temperatures and void fractions were divided into convenient intervals.

A matrix of temperature *vs* void fraction covering the entire LOCA operating range was created. Temperatures of 212°F, 302°F, 392°F, 428°F, and 511°F, and void fractions ranging from zero percent to 100 percent, in one percent intervals, were utilized. Matrix tests were conducted at established matrix points. Based on the results of the matrix tests, the worst LOCA condition was identified for the 17 min test.

TEST PROCEDURE

Matrix Tests

The pump was stabilized at 212°F, normal operating conditions of 35,300 usgpm, and a suction pressure of 1380 psia. The suction pressure was then slowly reduced to 14.7 psia, which corresponds, approximately, to the vapor pressure of the liquid at 212°F. The pressure was reduced at a rate permitting stabilization of the loop. Thus far, no liquid had been removed from the loop (zero percent LAV). This part of the test corresponds to the Phase 1 described before.

At this point, the loop drain valve was opened and water slowly bled, permitting stabilization of the loop, until the discharge head of the pump was totally degraded. The LAV fraction increased from zero percent to approximately 10 percent, corresponding to the second phase described above. Upon completion of the first part of the matrix test, the pump-motor set was turned off and data acquisition stopped. Note that during the entire test, loop temperature was maintained at 212°F.

The above procedure was repeated at 302°F, 392°F, 428°F, and 511°F.

It is important to note that although the tests were started at the rated flow and head conditions, and the loop resistance was not altered during the entire test (throttling valve at the same setting), the flow and head through the pump automatically changed due to low suction pressures and two-phase conditions.

Based on matrix test analysis, the worst LOCA condition was identified. At 392°F, at a LAV fraction of 25 percent, the pump shaft run-out peaked at 11.4 mils peak-to-peak and the pump-motor frame vibrations reached 0.209 in/s pk. Under normal operating conditions, these values are 2.56 mils peak-to-peak and 0.05 in/s pk, respectively.

17 Min Worst-LOCA Test

The pump was stabilized at 392°F under normal operating conditions of 35,300 usgpm, and a suction pressure of 1380 psia. Suction pressure was then slowly reduced to approximately 226 psia. At this point, the drain valve was opened to bleed water from the loop. When the LAV fraction reached 25 percent, the drain valve was closed, and the seal injection to the pump stopped to simulate the worst condition, as seal injection water will not be available during a LOCA. This was the beginning of the worst LOCA test. The pump was run under these conditions for 17 min and then turned off.

Following completion of the LOCA tests, the pump was restarted and a hydraulic performance test conducted at normal operating conditions of 511°F. Results indicated that there was no deterioration of pump internals due to operation at two-phase flow conditions and high vibration levels.

TEST RESULTS

Two peaks of shaft vibration levels, measured at the pump coupling hub, were observed at each test temperature. The highest peak during Phase 1, considering all the temperatures, occurred at 302°F and a suction pressure of 421 psia (Figure 4). The highest peak during Phase 2 occurred at 392°F and a LAV fraction of 25 percent (Figure 5). The amplitude of the second peak (11.3 mils peak-to-peak) is much greater than the first (4.7 mils peak-to-peak).

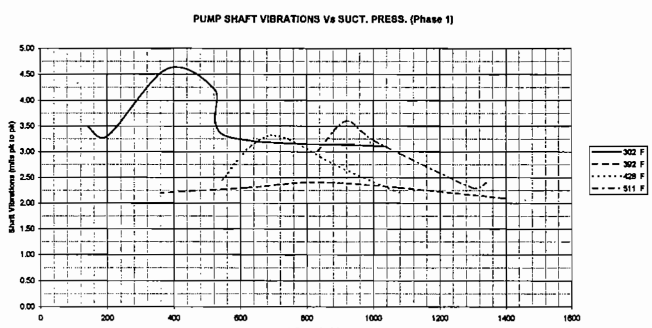

Figure 4. Pump Shaft Vibrations vs *Suction Pressure (Phase 1).*

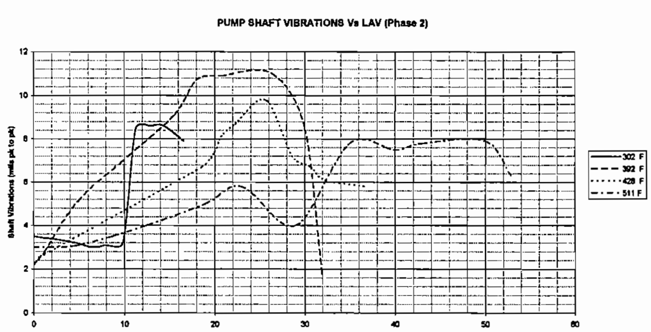

Figure 5. Pump Shaft Vibrations vs *LAV (Phase 2).*

Pressure pulsations in the pump discharge behaved in a manner similar to the shaft vibrations. The peak during Phase 1 (42 psi peak-to-peak, Figure 6) occurred at 302°F and 478 psia suction pressure. During Phase 2, pressure pulsations maximized at 392°F

(44 psi peak-to-peak, Figure 7), similar to the shaft vibrations, but at a lower LAV fraction of 18 percent. Note in Figure 5, that at 392°F, there is a secondary peak of shaft vibrations at an LAV of 18 percent.

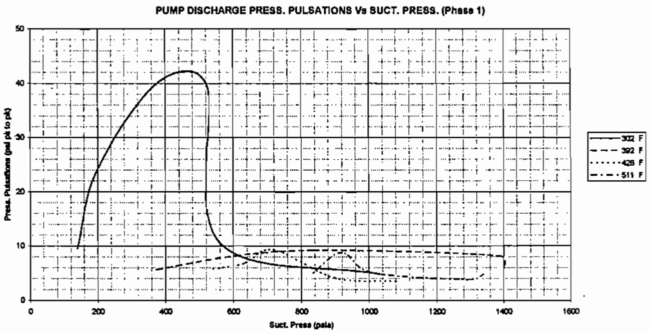

Figure 6. Pump Discharge Pressure Pulsations vs *Suction Pressure (Phase 1).*

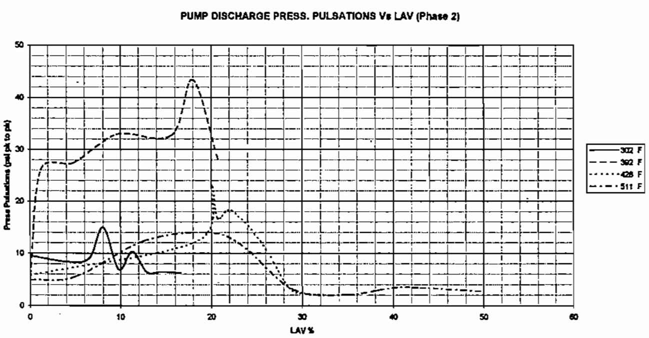

Figure 7. Pump Discharge Pressure Pulsations vs *LAV (Phase 2).*

The vibration readings shown in Figures 8 and 9 were taken at the upper end of the motor in the horizontal plane for Phase 1 and Phase 2, respectively. Unlike the shaft vibrations and pressure pulsations where two peaks are observed, motor vibrations peak only during Phase 2. At 302°F, vibrations peak at a LAV fraction of eight percent (0.26 in/s pk), and the second peak (0.22 in/s pk) occurred at 392°F and LAV of 27 percent.

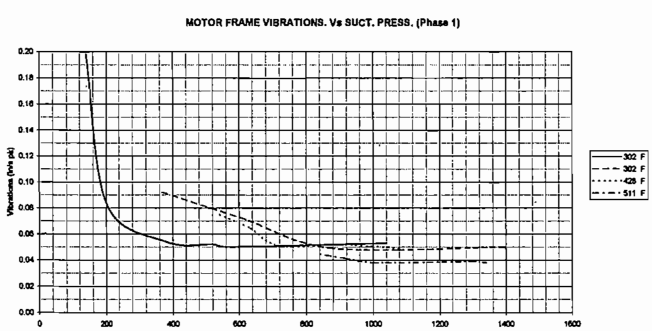

Figure 8. Motor Frame Vibrations vs *Suction Pressure (Phase 1).*

Based on the test results at different temperatures, the worst LOCA conditions were identified. At 392°F, at a LAV fraction of 25 percent, the combination of the shaft vibrations and motor frame vibrations were considered the most severe. The pump was run under these conditions for 17 min. During this test, the seal injection to the pump was also stopped to simulate the worst field conditions, as seal injection may not be available during a LOCA.

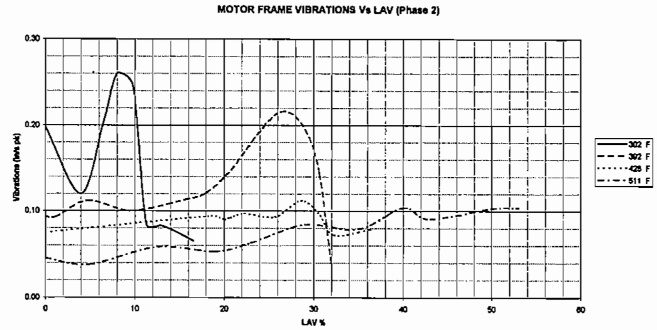

Figure 9. Motor Frame Vibrations vs *LAV (Phase 2).*

Post-Test Inspection

Following completion of all the LOCA tests, the pump and motor were completely stripped down and all parts inspected. There was no change in the pump and motor bearing clearances after the LOCA test. The mechanical seals showed no signs of damage or heat checking. All the structural components were visually inspected and the pressure boundary components were inspected by the liquid penetrant method. There were no signs of any damage to any of these components.

DISCUSSION

Void Distribution

The local void fraction distribution through the loop is not uniform and changes depending on the local temperature and pressure. Typically, the local void fraction at the discharge would be less than that at the pump suction, as the pump discharge pressure is much higher than the suction pressure.

The local void fraction distribution at the pump suction does not remain stable or homogenous at values less than 90 percent of $NSPH_{req}$. Suction flow distribution could change instantaneously from a uniform, bubbly flow to a stratified, slug, or annular flow. Maintaining a stable suction void at values less than 90 percent of $NSPH_{req}$ is impractical. The 90 percent $NPSH_{req}$ point occurs during the first phase of each matrix test, at a LAV fraction of zero percent. During the second phase, at LAV fractions greater than zero percent, the instability at the suction is more severe causing higher levels of pressure pulsations and vibrations.

Effects Of Loop Geometry

Tests conducted in a different test loop [3], on a twin volute single discharge HT pump of slightly higher specific speed (2800 US units), indicated that the worst LOCA conditions occurred at a much lower temperature (212°F) and at a void fraction (approximately 10 percent). This indicates that test loop geometry and pump design influence the worst LOCA condition. A critical difference in the two test loops was the head of liquid and volume above the impeller center line, resulting in a different $NPSH_{avl}$ in each case.

Loop geometry plays a major role in determining the void fraction distribution within a two-phase medium approaching the impeller. At low LAV fractions, the entire volume of void could remain stagnant at the highest point in the loop. A loop with a simple piping arrangement is the best means of uniformly distributing the void by maintaining uniform fluid momentum from the pump discharge to suction.

Deviation from results of previous tests can also be explained by variances in the natural frequencies of the test systems and also pump designs. Meaningful results can be attained by maintaining the highest level of simulation.

Shaft Vibrations

Phase 1, with a LAV fraction of zero percent, is similar to a standard NPSH test. As the suction pressure is lowered, when the resulting $NPSH_{avl}$ is less than required, cavitation at the impeller inlet occurs. Local pressure at the impeller blade leading edge drops below the vapor pressure of the liquid, resulting in the formation of local voids. The void then develops inside the impeller and due to repressurization, implodes, causing shock loads whose intensity can be very high. This generates high pressure pulsations and eventually vibrations in the pump. When suction pressure is further reduced, the number and volume of bubbles at the impeller inlet increases. Some of these bubbles fail to collapse within the impeller. These bubbles then dampen and cushion shock loads generated by the collapsing bubbles. Thus, shaft vibration levels and pressure pulsations reach peak values, decreasing as suction pressure is further lowered.

As liquid is removed from the loop during the second phase, voids that leave the impeller remain without collapsing and reenter the impeller. During this phase, $NPSH_{avl}$ remains unchanged. High levels of shaft vibrations can be caused by acoustic and convective wave propagation, pressure drop oscillations, and thermal oscillations [4].

Motor Frame Vibrations

Results indicate that pump-shaft vibrations, pump-motor frame vibrations, and pressure pulsations are not totally synchronized. This is expected as system response is dependent upon the relationship between exciting frequencies and resonance frequencies (acoustic and structural) of the systems.

Key Design Features of the Pump

The pump bearing plays a major role in the ability of the pump-motor set to survive a LOCA. The hydrodynamic bearing made of graphite impregnated carbons can operate under boundary lubrication with very high loading without a failure. The maximum amplitudes of shaft vibrations reached during the test (11.3 mils peak-to-peak) is nearly equal to the internal clearance of the pump bearing (12 mils diametrical) indicating contact between the bearing and journal surfaces. The self lubricating properties of the bearing (graphite) make it possible to operate with the minimum of external lubrication and cooling.

Another remarkable feature of the hydrodynamic bearing is its ability to reduce the impact of impulse loading by squeeze film damping. Also, the nonlinear bearing stiffness effect, where the stiffness increases as the inverse of the clearance, helps reduce the impact loading on the bearing.

The seal recirculation loop contains sufficient liquid to cool and lubricate the mechanical seals and the bearing for at least 30 min under two-phase flow conditions. This enables the mechanical seals and the bearing to operate without sustaining any distortion or damage, even without external seal injection.

The mechanical seals have the ability to operate under very low pressures, two-phase flow, and marginal cooling conditions while also having to operate at high pressures under single-phase conditions. This feature of being able to operate satisfactorily over a wide spectrum of conditions is quite remarkable.

The pump shaft and coupling stiffness is quite significant. This enables the uniform distribution of the radial loading between the motor and pump radial bearings. As a result, all of the loading generated at the pump impeller during two-phase flow conditions is not carried by the pump bearing alone. The stiff shaft (8.5 in) and coupling design also reduces the shaft deflection under high upthrust conditions which occur under normal operation.

CONCLUSIONS

The PHT pumps for a CANDU 6 reactor can satisfactorily operate under two-phase flow conditions, which may result from a loss of coolant accident. This has been demonstrated by full scale testing of the pumps under the worst possible conditions they may ever encounter.

The design features of the pump bearing (hydrodynamic) and the combination of bearing materials play a major role in the satisfactory operation of the pump under severe load conditions.

The pump mechanical seals are capable of operating without external seal injection for over 17 min. The design features of the seal recirculation loop facilitate this capability.

Although tests can be conducted at the worst case conditions (conservative), to obtain exact field simulation, loop geometry, and the head and volume of liquid above the impeller center line has to be duplicated.

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