THE ROLE OF OFF-DESIGN PUMP OPERATION ON MECHANICAL SEAL PERFORMANCE

by

William V. Adams

Vice President of Technology

Lee A. Waling

Senior Engineering Specialist

Randy R. Dingman

Design Engineer

and

Joseph C. Parker

Design Engineer

Durametallic Corporation

Kalamazoo , Michigan

William V. Adams is Vice President of Technology for the Durametallic Corporation. During Mr. Adams' 26 years with the company, he has held assignments as a Research Engineer, Chief Design Engineer, Assistant Manager of Engineering, Director of Engineering, and Vice President of Engineering, Research, and Quality Assurance. In 1992, he was named Vice President of Technology with supervision responsibilities over Research and Development and Quality Assurance.

Mr. Adams has served on the ASME B73 Chemical Pump Standards Committee. He is a past Chairman of the Seals Technical Committee of STLE and has served as their Educational Committee Chairman and Vice Chairman of their Standards Committee. Currently, he is chairman of their WG-3 Emissions Task Force, a member of the API 682 Seal Standards Committee and a member of the International Pump Users Symposium Advisory Committee. He received a B.S. degree in Mechanical Engineering from Western Michigan University (1967).

Lee A. Waling is a Senior Engineering Specialist in the Research and Development Department at Durametallic Corporation. He has been involved in product development and product improvement work for 20 years as a Design Engineer, Manager of Product Development, and now as a Senior Engineering Specialist. He has made major contributions in areas such as bellows seals, split seals, dry running seals, and other specialty seals. He received a B.S. degree in Mechanical Engineering from Western Michigan University (1971).

Randy R. Dingman is a Design Engineer in the Research and Development Department at Durametallic Corporation. He has been involved in laboratory research and product development work for seven years as a Research Engineer and now as a Design Engineer. He has made major contributions in areas such as materials research, product emissions testing, laboratory equipment development, and dry running seal technology. He received a B.S. degree in Mechanical Engineering from Western Michigan University (1986).

ABSTRACT

Field experience indicates that a high percentage of centrifugal pumps in industrial applications are not operated at their best efficiency point as related to product flow. Dry running, gas ingestion, vibration, cavitation, and internal recirculation are often the consequences of such off-design operation and are regarded as detrimental to the life and performance of end-faced mechanical seals. Little direct evidence of the role of these off-design operating conditions on mechanical seal performance has been documented in literature.

A test program documented temperature rise at the seal faces and damage to seal faces, secondary seals, and seal hardware during various off-design pump operations. Results provide users with a selection guide to aid in identifying mechanical seal designs, seal chamber configurations, and environmental controls that withstand off-design pump operation and improve mean time between planned maintenance.

INTRODUCTION

Historically, seal users and manufacturers have been handicapped by having to adapt mechanical seals to chambers originally designed for use by compression packing. The radial dimensions of the chamber cavities were established based on the needs of compression packing. Pressure in the conventional stuffing box was minimized to reduce loads on the packing and minimize leakage past the packing to atmosphere. These restricting dimensions remained industry's equipment standards until the late 1980s. While these chamber conditions were best for compression packing, they are not necessarily conducive for long life and extending mean time between planned maintenance (MTBPM) for mechanical seals.

Current application guides recommend use of enlarged bore seal chambers to improve MTBPM. However, more than 90 percent of pumps remain equipped with conventional stuffing boxes which limit the cavity space for seal application and provide poor pressure and poor flow circulation in the seal cavity.

As industry continues to demand longer MTBPM of pumps, it is essential that research work continue to evaluate the factors affecting the life of mechanical seals. Earlier studies determined the dissipation of seal generated heat in various seal chamber designs [1] and the effects that seal chamber designs and their application have on mechanical seal performance [2]. While seal chamber studies have advanced the longevity of mechanical seal designs and provided an environment for more creative seal designs, evaluation of pump operation as it relates to mechanical seal MTBPM has not been adequately addressed.

Work done in 1993 that further investigates the effects of off-design pump operations on various types and arrangements of mechanical seals and seal chamber designs is reported.

Previous evaluations [1, 2] were conducted under recommended pumping conditions of 100 percent of best efficiency point (BEP) flow. However, it is very common for pumps to operate under off-design settings which can induce conditions such as dry running, gas ingestion, vibration, cavitation, and internal recirculation all of which can be detrimental to mechanical seal life. To understand in greater depth how industrial pump operations affect MTBPM of seals, it was decided to study how various mechanical seals and seal chamber configurations withstand the off-design or upset conditions most frequently experienced in industry.

Evaluations were conducted by cycling the pump through the following user suggested conditions.

- 100 percent of BEP flow
- 10 percent of BEP flow
- 127 percent of BEP flow at a low NPSH condition
- 100 percent of BEP flow at a low NPSH condition
- Three percent entrained gas by volume at BEP
- Dry running operation

Various mechanical seal designs and arrangements were tested to determine what seal configurations presented specific positive or negative effects.

TEST EQUIPMENT AND METHODS

Testing was conducted on a standard 2 × 1-10 ANSI B73.1 pump, widely used in industry, to obtain measured data for documenting operating effects. The original intent was to conduct this evaluation using a clear acrylic pump having the same internal dimensions as the standard ANSI pump. This would have allowed visualization of all resulting activities. It was decided, however, that the differences in pump materials would alter the thermal characteristics of the entities being measured and that the acrylic pump would not withstand the severe off-design operating conditions planned for the test series. Therefore, a standard ANSI pump, incorporating 316 stainless steel for all wetted parts, was selected for the test series. The acrylic pump was also used, however, to obtain visual records of the fluid flow, gas entrapment, and cavitation effects for selected test conditions.

Three mechanical seal configurations were selected for testing in both a conventional stuffing box and an enlarged tapered bore seal chamber. The seal configurations included: a single pusher seal, a single cartridge metal bellows seal, and a double cartridge metal bellows seal. These six test arrangements are shown in Figure 1.

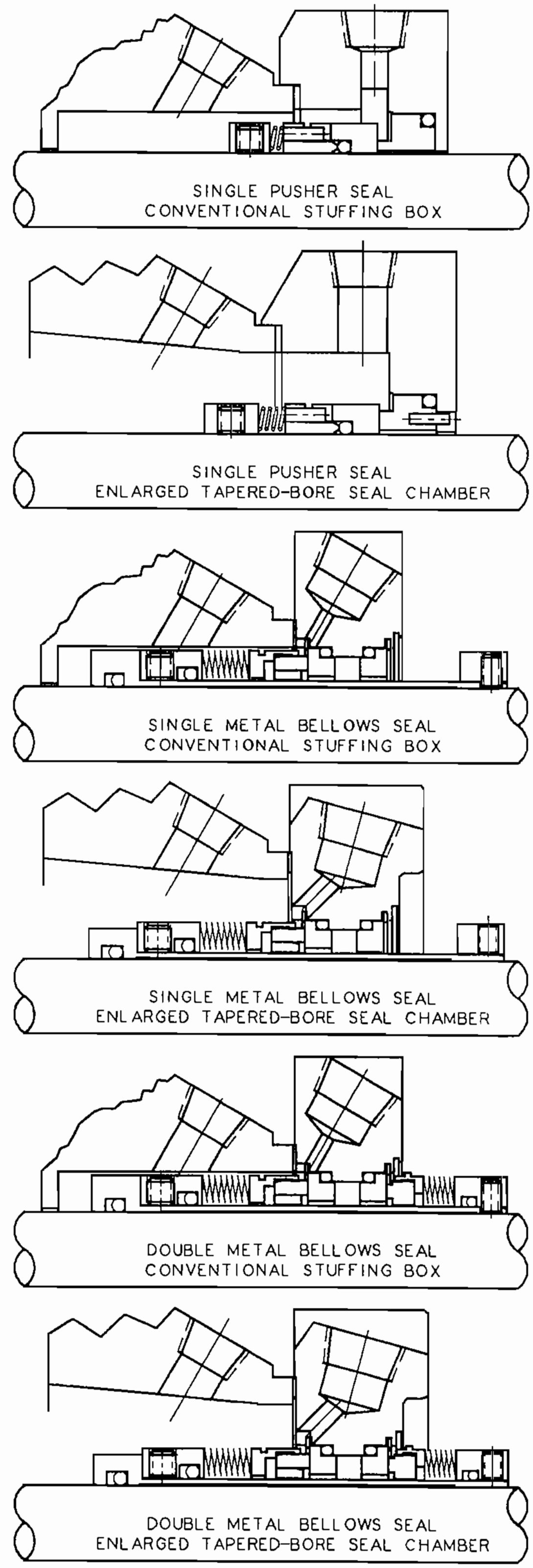


Figure 1. Mechanical Seal and Seal Chamber Arrangements.

The test circuit used during all testing is shown in Figure 2. All tests were conducted with softened municipal water as the pumped fluid. The seal chambers were dead ended per API Plan 02 except where noted otherwise. The suction reservoir had a 400 gallon capacity and a water level of approximately 4.0 ft (1.2 m) above the pump centerline. Suction pressure readings are listed in Table 1. The pumped fluid temperature was maintained at 150°F (66°C) by use of a cooling coil placed in the suction reservoir.

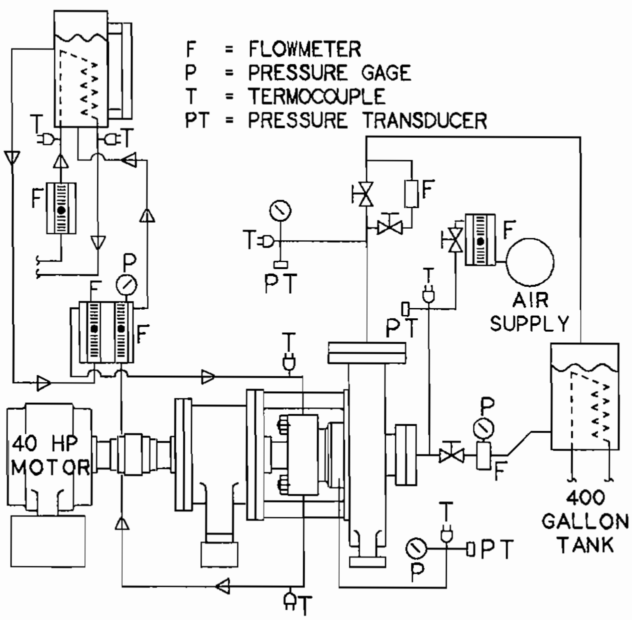


Figure 2. Test Circuit of ANSI B73.1 Pump.

Table 1. Summary of Seal Chamber Pressures and Pump Flowrates.

	100% BEP	10% BEP	LOW NPSH AT 127% BEP	LOW NPSH AT 100% BEP	3% GAS AT 100% BEP	DRY RUN AT ZERO FLOW
FLOW RATE (GPM)	150	15	190	150	150	0
SUCT PRESS (IN HG VAC)	10	3	22	22	10	0
DISCH PRESS (PSIG)	140	215	110	7.6 TO 30	140	0
BOX PRESS (PSIG)	13	18	10	22 IN HG VAC	13	0
AVAIL NPSH (FT)	17.2	22	5.5	3.5	17.2	---
TYP NPSH (FT)	5.5	1.5	8	5.5	5.5	---

The barrier fluid reservoir and associated piping were connected per API Plan 53 for use with the double seal testing only. Softened municipal water was used as the double seal barrier fluid which was maintained at 35 psig (241 kPa), approximately 25 psig (172 kPa) above the seal chamber pressure. The barrier fluid reservoir was located approximately 36 in (0.91 m) above the seal chamber center line. The reservoir was cooled using internal cooling coils. Cooling water was maintained at a constant temperature of 75°F (24°C) with a constant flowrate of 0.75 gpm.

The double seal barrier fluid was circulated through the reservoir, using a circulating feature built into the gland ring. The flow produced by this circulating feature was measured by using a double side-by-side rotameter. One rotameter was piped between the barrier cavity outlet and the reservoir. The other rotameter was piped between the reservoir and the barrier cavity inlet. Flow was measured to be 0.13 gpm. Catastrophic leakage into or out of the barrier cavity could be quickly detected by observing changes in the fluid height of each rotameter sight gage.

The pump was equipped with a fixed speed, 40 hp motor operating at 3600 rpm. All seal designs tested were sized to fit the 1.875 in (47.62 mm) shaft diameter of the pump. All seal configurations contained seal faces of resin-impregnated carbon graphite versus nickel-bound tungsten carbide and fluorocarbon compound O-rings.

Volumetric flow through the pump was recorded manually using a differential pressure analog flow meter. The pump was equipped with appropriate transducers to record pressures, temperatures, and vibration levels, using a computer based data acquisition system, as shown in Figure 3.

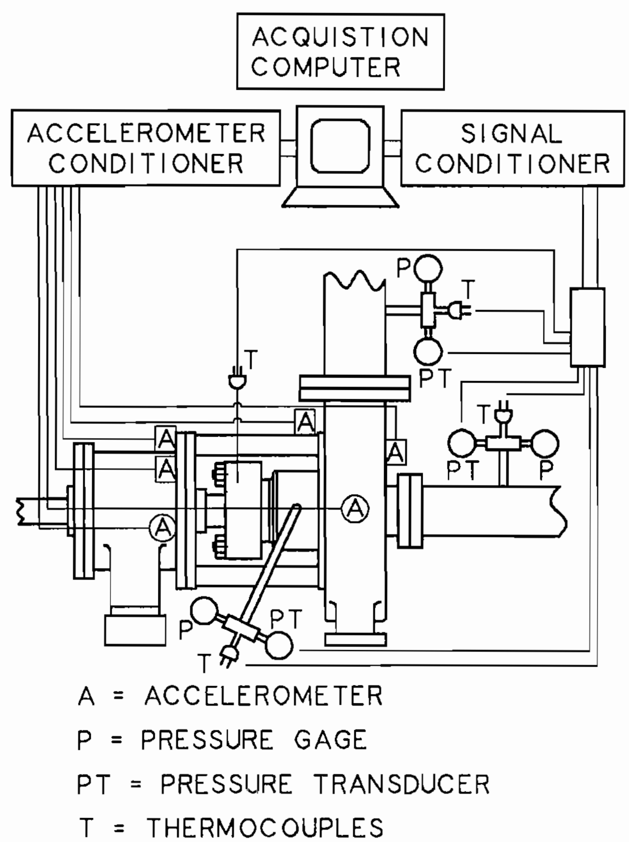


Figure 3. Schematic of Data Monitoring System.

Seal face temperatures were monitored with thermocouples mounted 0.06 in (1.52 mm) back from the stationary seal face. Seal chamber temperatures were monitored with thermocouples located approximately 0.25 in (6.35 mm) back from the face of the gland ring. The barrier fluid inlet and outlet temperatures and the barrier reservoir cooling fluid inlet and outlet temperatures were measured for double seal operations. Seal face temperatures during double seal operations were measured for the inner seals only.

Vibration was monitored by mounting accelerometers in the vertical, horizontal, and axial locations on the bearing housing near the gland ring and on the pump casing, as shown in Figure 3. Seal face surface profiles, waviness, and wear measurement data were evaluated before and after each test cycle.

In order to document mechanical seal performance *vs* pump operating conditions, two test cycles were established to evaluate various combinations of seal design, seal arrangement, and seal chamber configuration. The first test cycle encompassed a 48 hr time period, as shown in Figure 4. This test cycle was applied to each test setup. The pump was operated for 12 hr at the pump BEP; 11 hr at a low flow, high head condition (10 percent of BEP flow); one hr returning to the pump BEP; 11 hr at a high flow, low head, low NPSH condition (127 percent of BEP flow); one hr returning to the pump BEP; 11 hr at 100 percent of BEP flow with low NPSH; and one hr returning to the pump BEP.

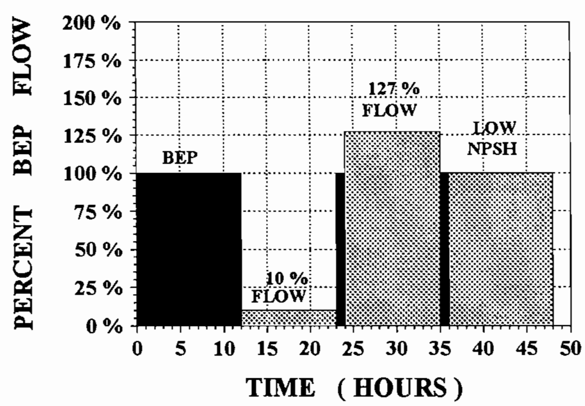


Figure 4. Off-Design Operation Test Cycle 1.

The second test cycle encompassed a 36 hr time period, as shown in Figure 5. This test cycle was applied to selected setup combinations. Pump operation was conducted for 12 hr at the pump BEP; 11 hr at 100 percent of BEP flow with three percent entrained gas by volume; one hour returning to the pump BEP; 11 hr at a dry running zero flow condition; and one hr returning to the pump BEP.

Gas entrainment was achieved by injecting compressed air into the pump suction line through a flow meter that could be adjusted to vary the percent of entrained air. Dry running operation was achieved by fully closing a valve in the suction line to stop water flow from the suction reservoir and opening the suction line to atmosphere through a small bleed line.

Pump operation through the two test cycles was performed using a pump impeller containing balance holes. Additional testing was conducted to observe differences in the performance of mechanical seals when using pump impellers containing balance holes versus those containing back pump-out vanes without balance holes. These tests were operated through the test cycle shown in Figure 4, using a single pusher seal design in an enlarged tapered-bore seal chamber.

TEST OPERATION AND RESULTS

The following discussion describes the mode of operation and results of the test cycles conducted for each pump operating condition. Comparative data are presented in several tables and

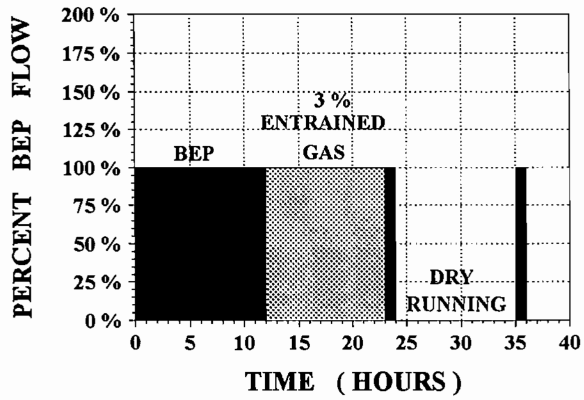


Figure 5. Off-Design Operation Test Cycle 2.

figures to support this discussion. Tabulated results for the seal chamber pressures and pump flowrates through these test cycles are shown in Table 1.

Pump BEP Operation

Testing during the initial stage of the first test cycle, as shown in Figure 4, at the pump BEP condition was achieved by fully opening the suction valve and throttling the discharge valve to obtain a flow of 150 gpm. This stage was important as it established baseline operating data to which succeeding off-design conditions could be compared. This testing produced an average seal chamber pressure of 13 psig (90 kPa) for both the conventional stuffing box and the enlarged tapered-bore seal chamber. Vibration behavior associated with all testing is discussed later.

Both the single pusher and metal bellows seals operated with lower differential temperatures between the seal faces and the pump suction in the enlarged tapered-bore seal chamber than in the conventional stuffing box. This is shown in Figure 6. The single pusher seal ran 40.8°F (22.7°C) cooler while the single metal bellows seal ran 24°F (13.3°C) cooler. If the pump had been operating with a process fluid near its boiling point, this could have had significant effects on seal performance.

For example, both of these single seal designs would be running dry if the difference between the temperature at the seal faces and the pumped fluid boiling point had been less than 25°F (13.9°C).

The double metal bellows seal performed with similar temperature characteristics when operated in both seal chamber designs. The inner and outer seal faces of the double metal bellows seal ran cooler than the pump suction temperature. The flow of the fluid, aided by the circulation feature, through the barrier fluid reservoir removed seal-generated heat at a rate greater than the seal faces could generate it. Operation of the double side-by-side rotameter indicated that there was no measurable leakage of the barrier fluid. This was confirmed by a visual check of the fluid level in the barrier fluid reservoir.

A separate test cycle using the single pusher seal in the enlarged tapered-bore seal chamber with a properly set impeller containing back pump-out vanes and no balance holes generated seal chamber pressures and seal face temperatures equivalent to those produced when using an impeller containing balance holes. However, when the same back pump-out vane impeller was improperly positioned, off the seal chamber back plate instead of the pump casing, the pump pulled a vacuum of 26 in (0.66 m) of Hg in the seal chamber. Seal face temperatures shot up to 550°F (288°C) during these vacuum conditions.

Ten Percent of BEP Flow Operation

Testing at 10 percent of BEP flow condition, the first off-design condition in the test cycle shown in Figure 4, was achieved by

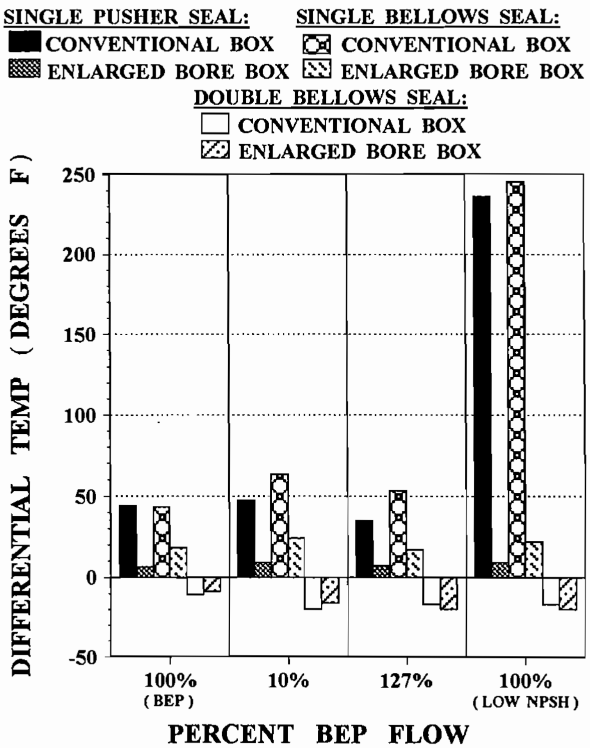


Figure 6. Seal Face to Suction Differential Temperatures.

throttling the discharge valve to obtain a 15 gpm flow. This produced an average pressure of 18 psig (124 kPa) in both the conventional stuffing box and the enlarged tapered-bore seal chamber. The pump ran with higher levels of vibration during this cycle when compared to operation at BEP.

Both the single pusher seal and the single metal bellows seal operated with differential temperatures between the seal faces and the pump suction which were 37.2°F (20.7°C) cooler when using the enlarged tapered-bore seal chamber *vs* the conventional stuffing box. This is shown in Figure 6. The overall differential temperatures between the seal faces and the pump suction for these single seals were higher than those observed during the BEP operation.

Operation of the double metal bellows seal provided seal face temperature and barrier fluid leakage performance equivalent to that of operation at BEP.

Separate cyclic testing under this condition using a single pusher seal and a properly set impeller with back pump-out vanes generated a seal chamber pressure of 44 psig (303 kPa) and seal face temperatures equivalent to those produced when using an impeller containing balance holes.

127 Percent of BEP Flow and Low NPSH Operation

Testing at 127 percent of BEP flow and a low NPSH, the second off-design condition in the test cycle shown in Figure 4, was achieved by fully opening the suction valve and throttling the discharge valve to obtain a 190 gpm flow. This produced a pressure of 10 psig (69 kPa) in both the conventional stuffing box and the enlarged tapered-bore seal chamber. As with operation at the 10 percent of rated flow condition, the pump ran considerably rougher during this cycle when compared to operation at BEP.

The single pusher seal and the single metal bellows seal both operated with lower differential temperatures between the seal faces and the pump suction in the enlarged tapered-bore seal chamber than in the conventional stuffing box, as shown in Figure 6. The single pusher seal ran 27.9°F (15.5°C) cooler while the single metal bellows seal ran 34.1°F (18.9°C) cooler. The overall differential temperatures between the seal faces and the pump suction for these single seals were slightly higher than those observed during the BEP operation but were lower than those observed during the 10 percent of rated flow operation. This is probably due to the varied fluid exchange rate between the pump casing and the seal chamber under these operating conditions.

Operation of the double metal bellows seal through this portion of the test cycle provided seal face temperature and barrier fluid leakage performance equivalent to that of BEP operation.

Separate cyclic testing under this condition using a single pusher seal and a properly set impeller with back pump-out vanes resulted in the pump pulling a vacuum of 26 in (0.66 m) of Hg in the seal chamber and the seal faces generating temperatures up to 670°F (354°C). This caused the rotor O-ring to melt and square up within one hour. When the pump operation was returned to BEP, the seal chamber pressure returned to a positive pressure and the seal leaked excessively.

Low NPSH at 100 Percent of BEP Flow Operation

Testing at the low NPSH at 100 percent of BEP flow condition, the last off-design condition in the test cycle shown in Figure 4, was achieved by fully opening the discharge valve and throttling the suction valve to obtain a 150 gpm flow. This produced cavitation at the pump suction valve and a vacuum of 22 in (0.56 m) of Hg in both the conventional stuffing box and the enlarged tapered-bore seal chamber.

When the single pusher seal and the single metal bellows seal were operated in the enlarged tapered-bore seal chamber, their seal face temperatures were similar to those obtained during the previous three operating cycles, even though the seal chamber was under a vacuum condition. This is shown in Figure 6. When similar testing was performed using the acrylic pump, air bubbles which were generated by pump cavitation and air bubbles which were being pulled across the seal faces were evacuated out of the seal chamber through the balance holes in the impeller.

When the single pusher seal was tested in the conventional stuffing box, excessive seal face squealing occurred along with seal face temperatures as high as 675°F (357°C). This led to melting and squaring up of the rotor O-ring. The carbon seal face contained one large pit beginning at the seal face outside diameter and extending radially across 40 percent of the seal face width. The carbon seal face roughness was three times greater than that obtained during operation with the enlarged tapered-bore seal chamber. Comparisons of the single pusher seal operation in the enlarged tapered-bore seal chamber versus the conventional stuffing box are shown in Figure 7.

The single metal bellows seal tested in the conventional stuffing box also created excessive seal face squealing and seal face temperatures as high as 500°F (260°C). There was no evidence of melting on the rotor O-ring. This is partly due to the seal design which locates the rotor O-ring away from direct contact with the heat-generating seal faces. The carbon seal face roughness was 13 times greater when operated in the conventional stuffing box *vs* the enlarged tapered-bore seal chamber. Comparisons of the single metal bellows seal operating in the enlarged tapered-bore seal chamber *vs* the conventional stuffing box are shown in Figure 8.

Operation of the double metal bellows seal through this portion of the test cycle continued to provide seal face temperature and barrier fluid leakage performance equivalent to that of operation at BEP.

When separate testing of the single metal bellows seal was performed using the acrylic pump, vapor bubbles which were generated by pump cavitation and air bubbles which were being pulled across the seal faces were observed to be collecting around

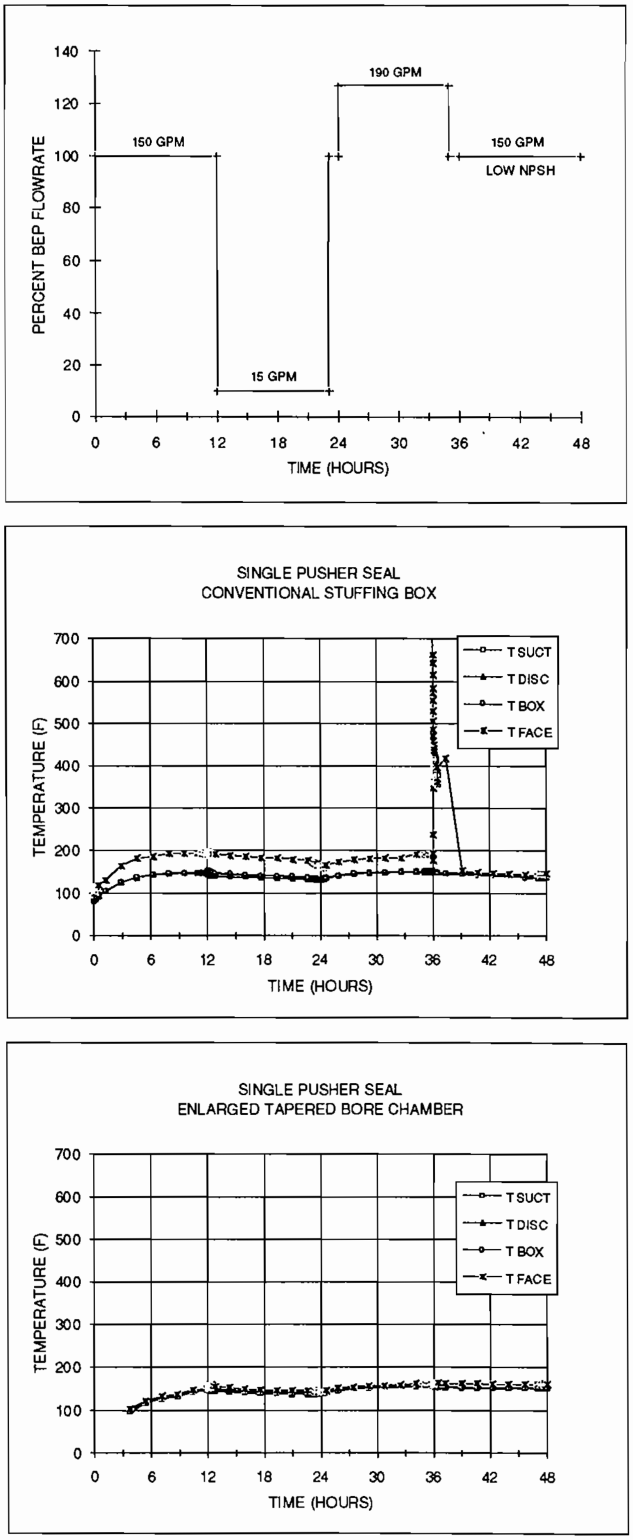


Figure 7. Single Pusher Seal Temperature Measurements.

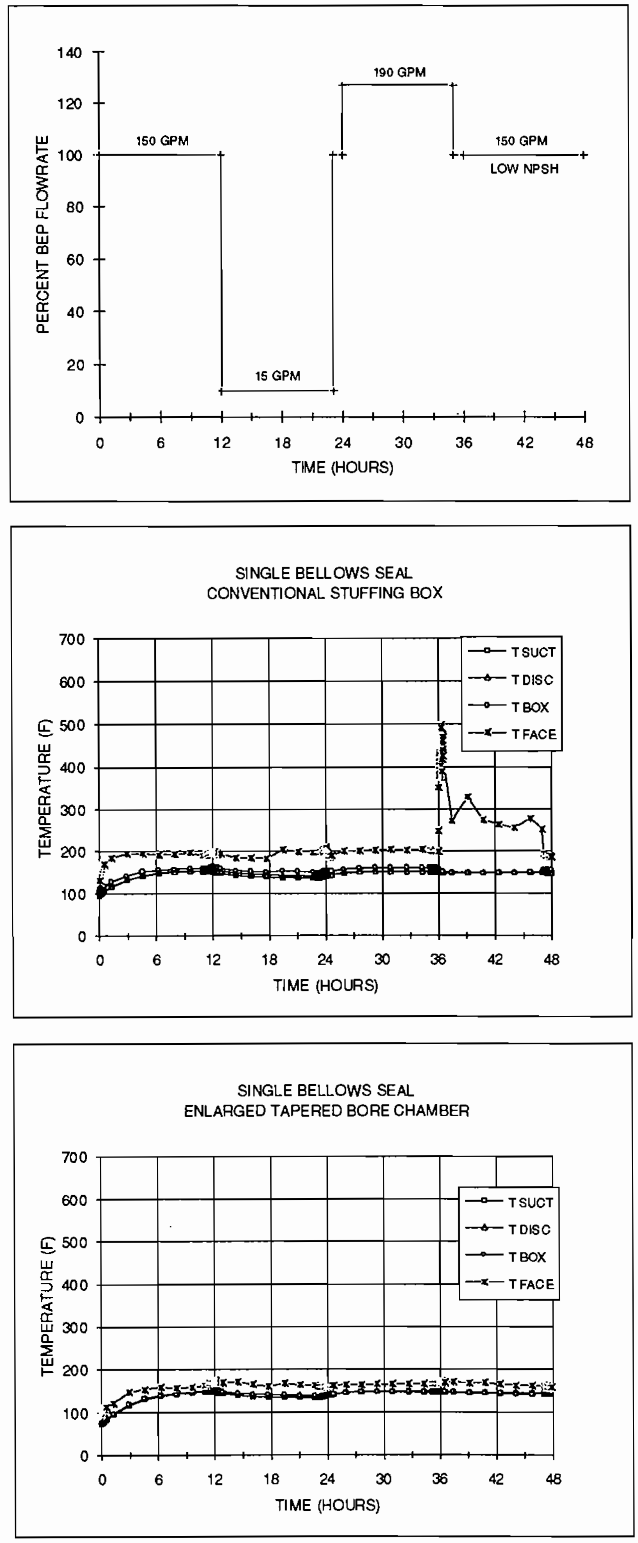


Figure 8. Single Metal Bellows Seal Temperature Measurements.

the seal in both the conventional stuffing box and the enlarged tapered-bore seal chamber.

Due to seal failure during testing at the 127 percent of BEP flow and low NPSH condition, testing was not conducted at the low NPSH at 100 percent of BEP flow condition with an impeller containing back pump-out vanes.

Low NPSH at 100 Percent of BEP Flow with Bypass Flush Operation

A separate off-design test was performed at the low NPSH at 100 percent of BEP flow condition with the addition of a bypass flush per API Plan 11 to see what effect it might have on mechanical seal performance. A single pusher seal with the conventional stuffing box was used. A bypass flush line was connected to the gland ring flush port. The pump was operated for five hours at BEP with the bypass flush line closed until the fluid temperature in the suction reservoir reached equilibrium at 150°F (66°C). The pump discharge valve was then opened fully and the suction valve was throttled to obtain a flowrate of 150 gpm.

This operating mode produced cavitation and a vacuum in the seal chamber of 20 in (0.51 m) of Hg. Excessive seal face squealing and a seal face temperature rise to 500°F (260°C) occurred within the first 60 sec. After operating at this condition for one minute, the bypass flush line was activated for a flow of 1.0 gpm. This had no effect on the seal chamber pressure. However, the seal face temperature dropped significantly to the same temperature as the pump suction inlet, which was 55°F (31°C) lower than the seal face temperatures encountered at BEP. This occurred within 30 sec. The change in seal face temperature is shown in Figure 9. It is apparent from this test that a bypass flush provides substantial protection for single mechanical seals during low NPSH modes of pump operation.

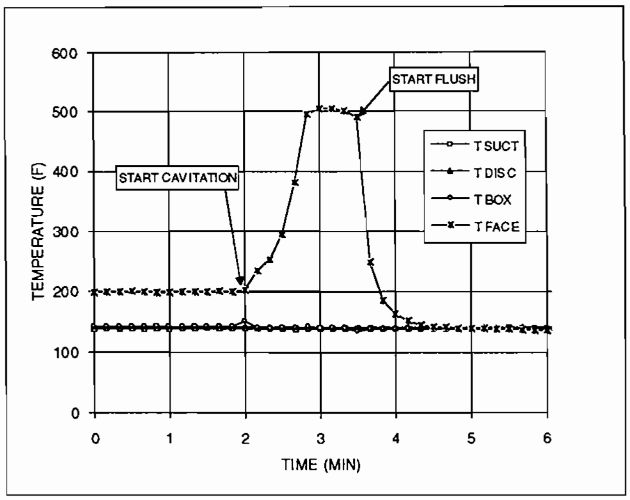


Figure 9. Effects of Bypass Flush on Seal Face Temperatures.

Three Percent Entrained Gas at 100 Percent of BEP Flow Operation

Air and gas may be present in centrifugal pumps due to vortices, poor sealing at inlet pipe connections, leaky valves on the suction side of the pump, or processes, which themselves contain gases. Most centrifugal pumps can handle liquids with gas entrainment only to a limited extent.

Testing during the first off-design condition in the test cycle shown in Figure 5 concerned pump operation with water containing three percent entrained gas by volume. The pump was operated at 100 percent of BEP flow while injecting compressed air at the rate of three percent by volume (38 SCFH) into the suction line of the pump. When the pump was fitted with a conventional stuffing box, operation was conducted using both a single pusher seal and a single metal bellows seal. When the pump was fitted with an enlarged tapered-bore seal chamber, operation was conducted using only the single metal bellows seal.

The injection of air into the pumped fluid did not alter seal chamber pressures and did not produce any harmful effects on the mechanical seal performance in terms of seal face temperature rise or excessive seal face wear rates. Pump operation appeared to be smoother than standard operation at BEP. The seal face temperatures for the single pusher seal and the single metal bellows seal operating in the conventional stuffing box rose only 3°F (2°C) and 15°F (8°C), respectively, above that at BEP. The seal face temperatures for the single metal bellows seal operating in the enlarged tapered-bore seal chamber rose only 5°F (3°C).

Because operation of the single seals in the conventional stuffing box showed little effect from entrained gas at three percent by volume, double seal operation was not tested under this condition in the standard ANSI pump.

Separate flow visualization tests for entrained gas were conducted using the acrylic pump fitted with the conventional stuffing box, the single metal bellows seal, and an impeller containing balance holes. As the injection of air into the process stream increased from one percent to three percent by volume, gas bubbles separated from the liquid and accumulated around the mechanical seal. As the volume of injected air was increased above three percent, centrifugal force and the accumulation of gas around the impeller hub eventually caused flow at the volute of the pump to collapse. Flow typically collapsed when the air injection reached the range of five percent to seven percent by volume. With the collapse of the flow at the volute of the pump, the seal chamber became void of liquid and the mechanical seal ran dry, exhibiting a rapid seal face temperature rise.

When this test in the acrylic pump was rerun using the enlarged tapered-bore seal chamber, the gas bubbles were evacuated from the seal chamber through the impeller balance holes. When this same testing was conducted with an impeller containing back pump-out vanes, a greater amount of gas bubbles accumulated in the seal chamber for the same level of gas entrainment.

Dry Running Operation

The second off-design condition in the test cycle shown in Figure 5 involved a dry-running segment, which was achieved by fully closing a suction valve in the suction line to stop water flow from the suction reservoir and opening the suction line to the atmosphere through a small bleed line. Temperature measurements and maximum seal face temperatures resulting from this testing are shown in Figure 10 and Figure 11, respectively.

After 100 minutes of dry running, the single metal bellows seal operating in the conventional stuffing box started to squeal and exhibited excessive seal face temperatures up to 650°F (343°C). The single metal bellows seal operating in the enlarged tapered-bore seal chamber started to squeal and developed high seal face temperatures of up to 450°F (232°C) after running dry for 10 minutes.

The single pusher seal operating in the conventional stuffing box started to squeal and exhibited excessive seal face temperatures up to 700°F (371°C) after 60 min of dry running. The single pusher seal was not operated in the enlarged tapered-bore seal chamber under this operating condition.

Operations under dry-running conditions were also viewed using the acrylic pump. When operating a single metal bellows seal in a conventional stuffing box, the liquid was evacuated from the pump casing once the dry-running condition was attained. However, some liquid remained trapped in the conventional stuffing box. When this test was repeated using the enlarged tapered-bore seal chamber, the liquid was totally evacuated from the seal chamber at the same time that it was evacuated from the pump casing. This would explain the more rapid increase in seal face temperature when using the enlarged tapered-bore seal chamber.

Operation of the double metal bellows seal under this dry-running condition had little effect on the seal face temperatures, producing results similar to those of the other test cycles. The heat removal rate of the double seal barrier system increased 28 percent from 2041 Btu/hr to 2626 Btu/hr when going from operation at BEP to a dry-running condition in the conventional stuffing box. The heat removal rate of the double seal barrier system remained

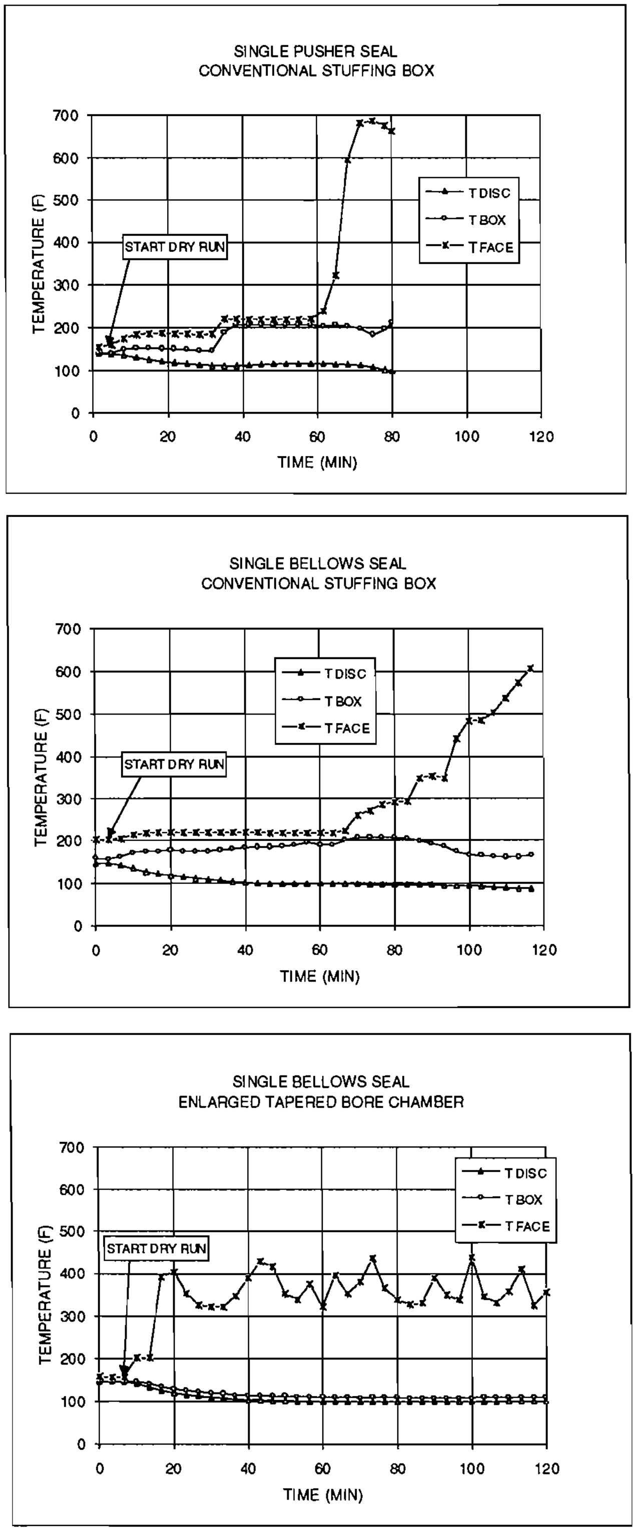


Figure 10. Dry-Running Operation Temperature Measurements.

constant at 2100 Btu/hr when going from operation at BEP to a dry-running condition in an enlarged tapered-bore seal chamber.

Pump Vibration in Off-Design Operation

During each test cycle, a mechanical vibration analyzer measured the axial, vertical, and horizontal vibrations on the bearing

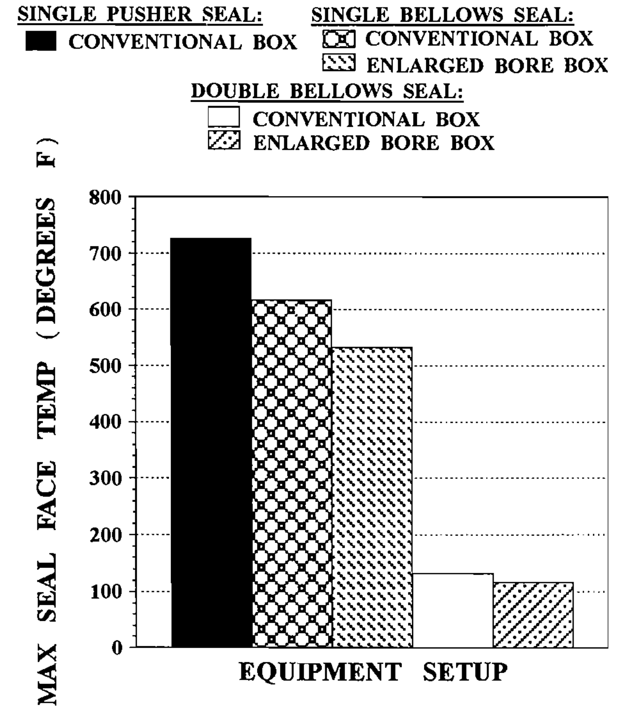
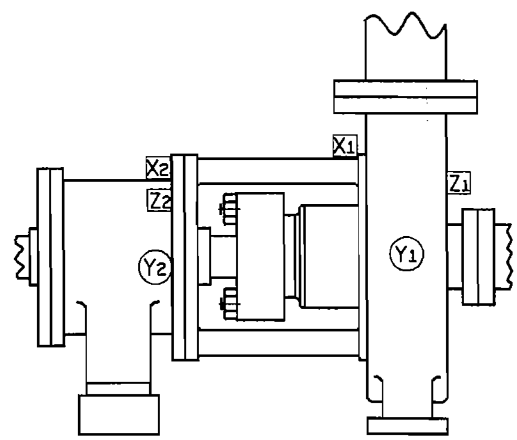


Figure 11. Dry-Running Operation Maximum Seal Face Temperatures.

housing and the pump casing. Vibration probes were located as shown in Figure 12. All tests were operated at 3600 rpm. As previously stated, the impeller used during the off-design testing contained balance holes. The impeller was mechanically balanced prior to installation (to within 0.011 oz-in/lb of impeller).



	100%BEP	10%	127%	LOW NPSH	DRY RUN
	IN/SEC	IN/SEC	IN/SEC	IN/SEC	IN/SEC
X1	0.14	0.25	0.15	0.27	0.07
Y1	0.14	0.15	0.22	0.34	0.07
Z1	0.23	0.23	0.29	0.37	0.07
X2	0.20	0.19	0.19	0.23	0.10
Y2	0.21	0.22	0.22	0.34	0.13
Z2	0.19	0.27	0.20	0.28	0.07

Figure 12. Vibration Probe Locations and Measurements.

The vibration velocities encountered during the various cycles of pump operation are shown in Figure 12 as filtered out readings. The lowest magnitudes of vibration were measured during operation at BEP and during dry running; vibration velocity values were 0.23 in/sec (5.8 mm/sec) or less at all measured locations. The greatest vibrations were encountered axially on the pump casing during the low NPSH cavitation testing where vibration velocities reached 0.37 in/sec (9.4 mm/sec).

Seal and Shaft Vibration in Off-Design Operation

A separate test series was conducted to evaluate more closely the adverse effects of vibration on mechanical seals during the off-design operating conditions which generated the greatest pump vibrations. For this test series, another ANSI pump was set up, as shown in Figure 13, so that displacements could be measured on metal bellows seal designs readily available in industry, four with vibration dampeners and four without vibration dampeners.

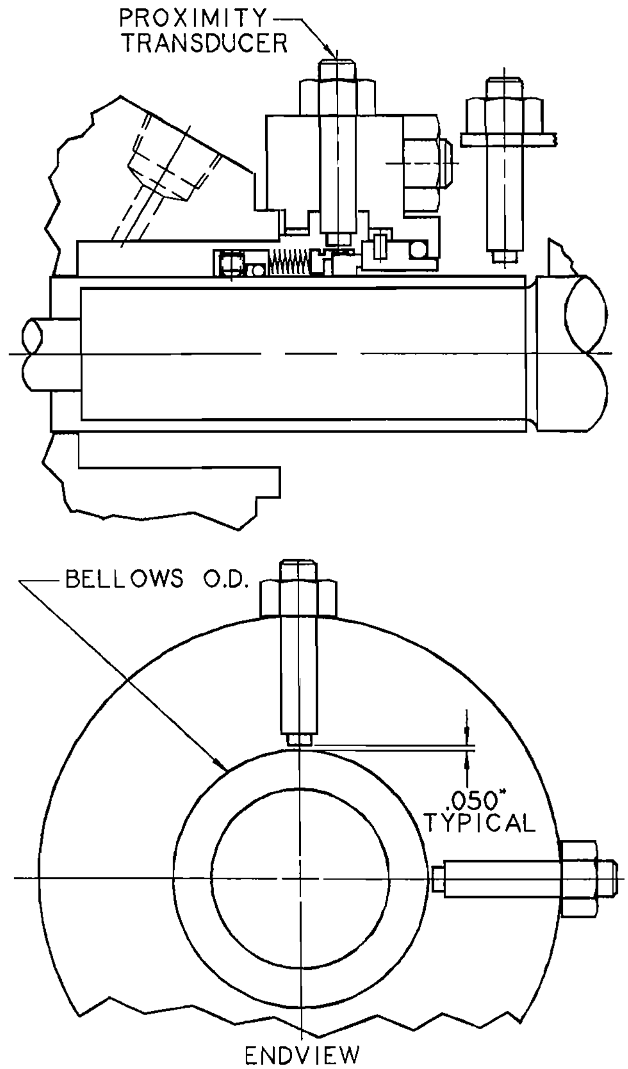


Figure 13. Metal Bellows Seal Vibration Test Arrangement.

Two noncontacting 8.0 mm proximity probes were mounted 90 degrees apart in the seal chamber and spaced 0.050 in (1.27 mm) from the face end of the metal bellows seal. Two 5.0 mm proximity probes were mounted next to the gland ring to measure shaft deflection on the atmospheric side of the seal. One 5.0 mm proximity probe was mounted over the coupling keyway to measure shaft rpm. The orbits of both the metal bellows seal face end and the shaft were monitored on an oscilloscope. The seal face temperatures were monitored with a thermocouple located 0.060 in (1.52 mm) from the stationary seal face.

The pump was operated under normal conditions and then forced to cavitate by throttling the suction valve. Pitting, due to cavitation, occurred on the impeller housing and the seal chamber back plate due to cavitation and is shown in Figure 14. Tabulated results are shown in Table 2. As in previous cavitation testing, all the metal bellows seals, with and without vibration dampeners, exhibited excessively high seal face temperatures, from 600 to 800°F (316 to 427°C).

Table 2. Cavitation Measurements.

	NORMAL RUNNING	PUMP CAVITATION
SEAL CHAMBER TEMPERATURE °F (°C)	150 (65)	150 (65)
SEAL FACE TEMPERATURE °F (°C)	160 (71)	600 - 800 (315 - 427)
SHAFT SLEEVE DISPLACEMENT (IN PEAK - PEAK)	0. 002	0. 0037
BELLOWS END PIECE DISPLACEMENT W/O VIBR. DAMPENERS (IN PEAK - PEAK)	< 0. 005	0. 04
BELLOWS END PIECE DISPLACEMENT W/ VIBR. DAMPENERS (IN PEAK - PEAK)	< 0. 005	0. 015
ESTIMATED SEAL LIFE W/O VIBR. DAMPENERS (HOURS)	1100 +	< 1
ESTIMATED SEAL LIFE W/ VIBR. DAMPENERS (HOURS)	1100 +	31 +

During normal operation, all seal designs exhibited vibration displacements of less than 0.005 in (0.13 mm) peak-to-peak. During cavitation operation, the metal bellows seals without vibration dampeners experienced vibration displacements of 0.040 in (1.02 mm) peak-to-peak which caused all seals to fail in less than one hour.

The seal faces were pitted and the metal bellows leaflets were torn, partially on some designs and completely on others. There was no damage to the rotor O-rings. This is again partly due to the seal design which locates the rotor O-ring away from direct contact with the heat source.

During cavitation operation, the metal bellows seals with vibration dampeners had vibration displacements of 0.015 in (0.38 mm) peak-to-peak. This caused two seals to fail after 31 and 36 hr of operation. The remaining two seals survived the entire 48 hr test period.

Influences of Pump Impeller Design

The running clearance for pump impellers vary with the type of impeller installed. Observations of seal performance indicate that mis-setting of clearances on impellers containing back pump-out vanes can cause excessively low or high pressures in the seal chamber. This problem emphasizes the importance of properly setting pump impeller clearances.

Tests were run with a standard open impeller containing back pump-out vanes with no balance holes. In one test, the impeller

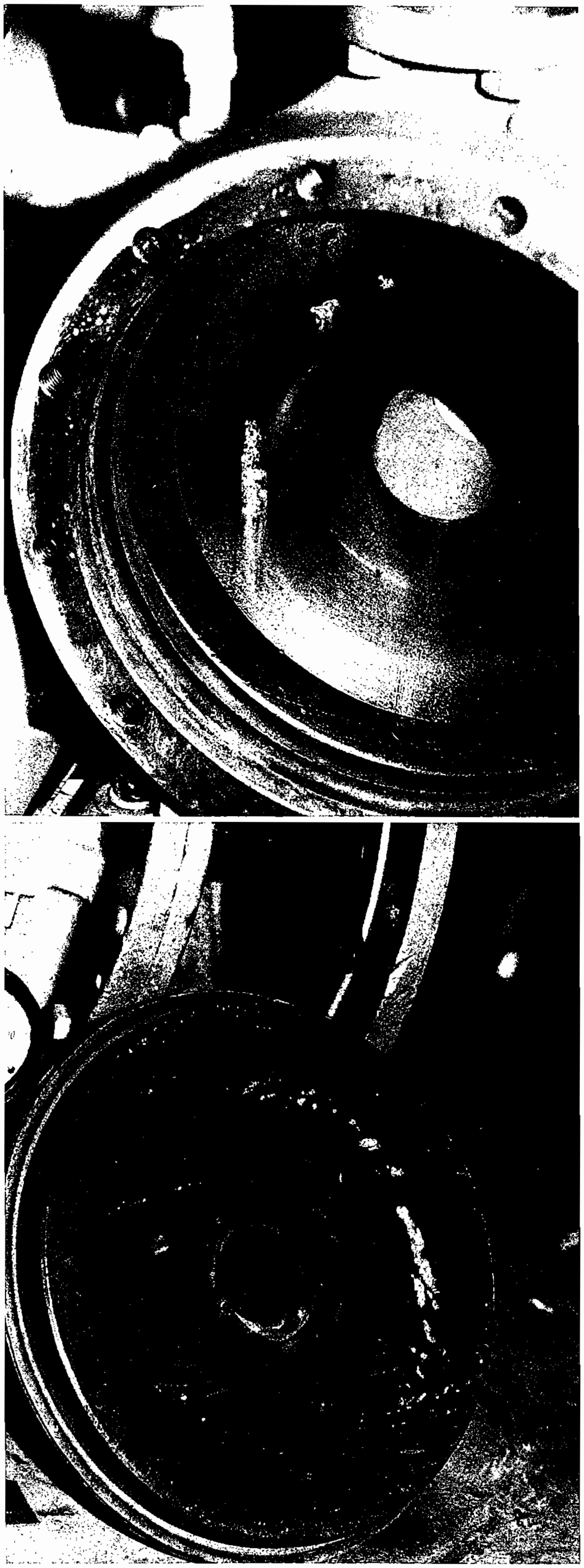


Figure 14. Pump Cavitation Damage.

was accidentally set 0.015 in (0.38 mm) away from the seal chamber wall instead of the casing wall. Operation of this test setup caused excessive vacuums (negative pressures) to occur in the seal chamber, even while operating the pump at BEP. This resulted in air being drawn across the seal faces, accumulating in the seal chamber, and resulting in seal face temperature excursions greater than 600°F (316°C). Secondary seals were squared up and melted from the excessive temperatures.

At the other extreme, long term operation of the pump by resetting the impeller to casing clearance to make up for wear at the face of the impeller and the casing resulted in continuing higher pressures against the seal chamber. This higher pressure against the inner seal of a double mechanical seal could result in a pressure reversal on the mechanical seal and leakage of product into the double seal barrier system.

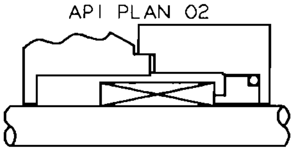
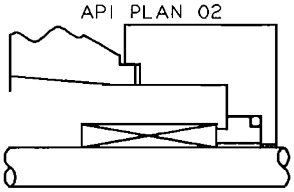
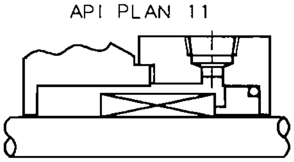
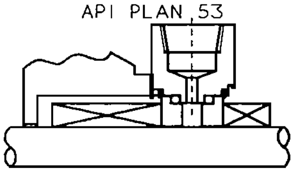
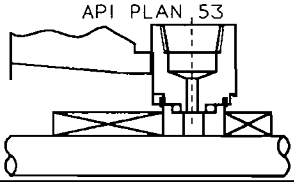
The characteristics of impellers with back pump-out vanes can be inconsistent because of the manufacturing processes used and/or modifications made by users. These inconsistencies, in turn, can affect the operating environment for mechanical seals. Significant height variations in the back pump-out vanes have been noted by users. Some users state that the axial length of an impeller is controlled by taking a final clean-up cut on the back of the impeller. In some instances, back pump-out vanes have been measured to be less than 0.030 in (0.76 mm) and have resulted in high seal chamber pressures. The manufacture of impellers containing back pump-out vanes must be carefully controlled to ensure proper seal chamber pressures. Users should be careful in purchasing new pumps and repair parts to ensure the purchase of high quality parts from experienced and reputable manufacturers.

SEAL SELECTION GUIDELINES

The preceding observations and analysis confirm the proposition that selecting the proper seal chamber and mechanical seal design for various pump operating conditions can significantly extend MTBPM of mechanical seals. Previous publications [1] and [2] have discussed the effects of seal chamber design on mechanical seal performance. Table 3 was developed to further assist in this selection process. The "Seal Selection Guide" provides recommendations for each off-design operating condition. These recommendations are specified for applications using various combinations of seal and seal chamber designs. They are not meant to illustrate a comparison between impeller designs and/or seal types (i.e. pusher versus welded metal bellows). They are the best recommendations based on the combination of equipment installed and, even though some of these recommendations appear to be equivalent, performance levels will vary accordingly.

For example, if a pump is known to run dry after its feed tank has been drained, then the best seal design for the application is a double seal with a barrier fluid circulating feature in either an enlarged tapered-bore seal chamber or a conventional stuffing box. If a single seal is required for this same application, then the best configuration would be a single seal in a conventional stuffing box. If a pump operates at low NPSH due to high flow or a restriction in the suction line where there is a potential for pulling a vacuum in the seal chamber, then the best configuration for the application is a double seal with a barrier fluid circulating feature (Note that the asterisk on the selection guide indicates that the barrier system needs to be monitored more often at this condition since higher vibrations may cause higher barrier fluid leakage rates). If a single seal is required for this application, then the best configuration would be a single seal with a bypass flush (API Plan 11). If a bypass flush is not applicable, then a single seal with an enlarged tapered-bore seal chamber would be the next best selection. A bypass flush per API Plan 11, where the flush is directed over the seal faces, will always result in improved single seal performance as long as the flush is not excessive enough to cause

Table 3. Mechanical Seal Selection Guide.

CONFIGURATION \ OPERATING CONDITION	100% BEP FLOW	10% BEP FLOW	LOW NPSH, 127% BEP	LOW NPSH, 100% BEP	3% GAS, 100% BEP	DRY RUN, NO FLOW	LOW BOIL PT MARGIN
SINGLE SEAL CONVENTIONAL BOX API PLAN 02 	3	4	5	5	3	4	4
SINGLE SEAL ENLARGED BORE CHAMBER API PLAN 02 	2	3	3	3	2	5	3
SINGLE SEAL API PLAN 11 	1	2	2	2	2	4	2
DOUBLE SEAL CONVENTIONAL BOX API PLAN 53 	1	1 (*)	1 (*)	1 (*)	1	1	1
DOUBLE SEAL ENLARGED BORE CHAMBER API PLAN 53 	1	1 (*)	1 (*)	1 (*)	1	1	1

1 = OPTIMUM CHOICE, 5 = LESS OPTIMUM CHOICE
* VIBRATION MAY CAUSE HIGHER BARRIER LEAKAGES

seal face erosion. If an operator is not sure of the conditions within a seal chamber, a bypass flush is recommended for all single seal applications.

CONCLUSIONS

The purpose for presenting this information is to point out more clearly the need for a good marriage between the pump, the pump impeller, and the seal chamber in order to provide an adequate environment in the seal chamber for mechanical seals and to achieve maximum MTBPM for the total pumping system. Opportunities for substantial improvements appear feasible with pump manufacturers and seal manufacturers working more closely in this area.

The uptime of centrifugal pumps, the longevity of mechanical seals, and the minimization of fugitive emissions can be further improved. These studies have revealed significant information on the operation and outfitting of centrifugal pumps and the application of mechanical seals. These studies support the following conclusions:

- It is imperative that careful consideration be given to the pump fluid characteristics and the heat rates of mechanical seals to ensure that the fluid in the seal chamber is always in a liquid state and capable of providing good lubrication to the mechanical seal. Past guidelines have always stressed operating 25°F (14°C) away from the flash temperature of the fluid in the seal chamber. This recommendation appears to be frequently overlooked or ignored, but is still applicable.

- Open enlarged-bore seal chambers are more forgiving to the survival of single mechanical seals than conventional stuffing boxes during both normal pump operation and off-design pump operating conditions. Seal face-to-product boiling point margins can be increased by as much as 40°F (22°C) without the aid of a bypass flush. Double mechanical seals with barrier fluid circulating features achieve adequate protection to survive off-design pump conditions including low NPSH cavitation and dry-running operation independent of the seal chamber design.

- Low NPSH cavitation conditions can cause significant damage to single mechanical seals when there is a vacuum in the seal chamber. Applying a bypass flush can reduce the seal face temperatures considerably during this condition. However, the use of a close clearance throat bushing in combination with a bypass flush is the best arrangement to assure maintaining a positive pressure in the seal chamber, if the pump must run at this condition.

- Dry-running conditions can cause significant damage to single mechanical seals. Operating with a conventional stuffing box is less severe during dry running than with an enlarged tapered-bore seal chamber. A good indicator of dry-running operation is a squealing sound emanating from the seal gland ring. There is no known "fix" for keeping a single mechanical seal cool during dry-running operation; the only alternatives are the use of an external flush with a close clearance throat bushing or use of a double mechanical seal.

- Off-design operation of typical ANSI pumps at 10 percent of BEP flow has no direct harmful effects from short term testing on mechanical seal performance. However, the accompanying vibration levels may be damaging to the seal and reduce overall long-term reliability of the equipment.

- Gas entrainment, at volumes up to three percent, in the pumping stream have no apparent damaging effects on the performance of single mechanical seals, even though flow visualization studies show the presence of gas bubbles in the liquid around the mechanical seal. Flow visualization studies also show that strakes added to the seal chamber bore may have some beneficial operating effects with various impeller designs.

- Pump impeller design can have significant effects on the seal chamber environment and the operating performance of mechanical seals. For pumps equipped with enlarged tapered-bore seal chambers, impellers containing balance holes tend to bleed off gases from the seal chamber during cavitation operation or operation with gas entrainment. Improper setting of pump impellers containing back pump-out vanes can cause high vacuums in the seal chamber even when operating at 100 percent of BEP flow which would result in significant damage to mechanical seals.

- Metal bellows seals equipped with close-clearance vibration dampeners can tolerate vibrations from off-design pump operation many times longer than metal bellows seals which are not equipped with vibration dampeners.

- Dual mechanical seals are capable of tolerating low NPSH and dry-running off-design conditions with less damage than single mechanical seals. In addition, dual mechanical seals equipped with circulating features will achieve adequate protection to survive operation during low NPSH and dry-running, off-design conditions.

- Even though mechanical seals can tolerate these off-design conditions for short periods of time, the post-test carbon surface profiles indicate that the mechanical seal could have appreciably higher leakage when and if the pump is returned to normal operating conditions. In the development of future pump and seal chamber standards, every effort should be made to assure a net positive pressure in the seal chamber during low NPSH and off-design pump operation.

- Maintaining a positive pressure in the seal chamber per API Plan 11 has been found to have beneficial effects on the performance of single mechanical seals during off-design pump operations.

These studies, evaluating the effects of off-design pump operations on the performance of mechanical seal designs, have resulted in a seal selection guide which can be used to optimize mechanical seal life.

REFERENCES

1. Davison, M. P., "The Effects of Seal Chamber Design on Seal Performance," *Proceedings of the Sixth International Pump Users Symposium,* The Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 3-8 (1989).
2. Adams, W. V., Robinson, R. H., and Budrow, J. S., "Enhanced Mechanical Seal Performance Through Proper Selection and Application of Enlarged-Bore Seal Chambers," *Proceedings of the Tenth International Pump Users Symposium*, The Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 15-23 (1993).