THE EFFECTS OF SEAL CHAMBER DESIGN ON SEAL PERFORMANCE

by

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ABSTRACT

Growing attention has focused on the need to increase mean-time between failure (MTBF) for pumps used in the chemical process and refinery industries. Increasingly, discussion has centered on mechanical seals and the benefits of optimizing seal chamber design to improve the seal's operating environment and hence, its service life. A test program was recently completed which examined typical heat transfer phenomena within seal chambers of various selected designs. The test results confirmed previous research in this area, identified specific causes of deficient seal performance, and indicated definitive areas of improvement for seal chamber design.

INTRODUCTION

Over the past several years, industry has recognized the generally high frequency of pump failure as a global problem. Moreover, concern is being concentrated on mechanical seals and their operating environments; i.e., stuffing boxes or seal chambers. A major area of effort has focused on increasing stuffing box/seal chamber size to improve seal life and to extend pump MTBF. One key objective is to increase radial clearance between shaft and housing; the advantages being: increased design flexibility and better heat transfer away from mechanical seal faces. An extensive research program was conducted by Durametallic to determine the advantages and disadvantages of the enlarged cylindrical and tapered bore seal chambers.

Several seal chamber designs were evaluated: two standard stuffing boxes (minimum radial clearance), three different enlarged cylindrical bore seal chambers, and two enlarged tapered seal chambers. These seal chambers were installed in two standard ANSI pumps. Some were tested with a balanced, metal bellows seal, and some were tested with an unbalanced, pusher-type seal. Various temperatures and pressures were monitored to determine the effects of design features such as taper, bore size, throat configuration, and impeller characteristics.

The results of the testing showed that certain seal chambers significantly improved seal performance. They also showed how specific seal chamber features affected seal performance. Recommendations are developed concerning several seal chamber design features.

Background

The typical stuffing box in a pump built to ANSI or API standards is intended to accept either packing or a mechanical seal. This combination of functions has often limited the performance and design of mechanical seals. Specifically, the radial clearance from shaft to housing bore is better suited to packing than mechanical seals. However, in the last few years, several new seal covers have been designed solely for use with mechanical seals. The seal area in these new covers features increased radial clearances and, in some cases, a tapered bore. This change in approach from a common stuffing box to separate housings for packing and mechanical seals has two main objectives. One is to increase radial clearance, providing greater design flexibility. The other is to increase the area around the seal, providing a better environment that will improve seal life.

The benefits of design flexibility are easily recognizable. New and improved seals can be designed for these pumps. But, these benefits will not be realized until widespread use of enlarged seal chambers allows seal manufacturers to develop seals that take full advantage of the increased area.

Improved seal environment, however, can be utilized today. Mechanical seals operate best with a stable, thin fluid film between the running seal faces. Formation of this lubricating film is critical to seal performance. An incomplete film or a too thin film will cause face damage. A too thick film will cause unacceptable leakage. Many of the factors that determine the film thickness are built into the seal. But, some factors are influenced by the seal environment. For example, product temperature, product pressure and the presence of vapors are very important factors. High temperatures can cause heat checking or other seal face damage. Operation near the vapor point of the product fluid can cause vaporization at the seal faces. Dry running of the seal faces can occur if the point of phase transformation is too near the product side of the seal faces. Air present in an improperly vented seal chamber can be centrifuged to the seal faces and cause dry running. Therefore, a seal chamber that offers protection against these factors will improve seal performance.

On the basis of some promising case histories, industry has begun to embrace the concept of separate housings. In fact, the new editions of ANSI B-73 and API 610 will include specifications for these new covers. Minimum recommended radial clearances will be specified. They will also require that the housing be self-venting. But, what real effect will these new seal chambers have on seal performance? Which, if any, of the new seal chamber designs is the best? What makes the enlarged seal chambers better? These questions led to a research program aimed at determining the effects of seal chamber design on seal performance.

TESTING

To gain a better understanding of the interaction between seal chamber and seal, it was decided that testing in a real environment would be the best approach. The seal chamber is an integral part of the pump fluid end. Testing in a real pump allowed the interactions between fluid end, seal chamber, and seal to be considered. Water at moderate temperature and pressure was used as the process fluid. Although this duty was not demanding of the pump or seal, the test results indicated significant improvement in seal performance. Seal face temperature was used as the indicator of seal performance because of its sensitivity to the conditions at the seal faces. Any changes in the lubricating film were immediately evident in the seal face temperature.

The Testing and Data

Two test circuits were constructed, as shown in Figure 1. Each consisted of a pump, isolation valves, leakage make up tank, heat exchanger, and instrumentation. The make up tank was pressurized by an air pressure regulator. This allowed control of the product suction pressure. In order to maintain consistency, the suction pressure was set such that the seal chamber pressure was always 50 psig. The heat exchanger provided control of the product suction temperature. This was held at 75°F for all of the tests. The test loop was constructed with 1.50 in schedule 40 galvanized steel pipe.

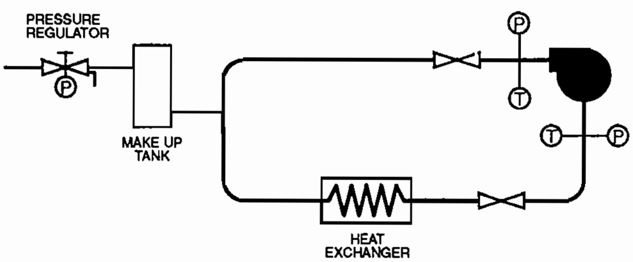

Figure 1. Test Circuit Diagram.

One test loop used a standard 1 × 2-10 ANSI pump with a 3500 rpm drive motor. It was equipped with a minimum diameter, open impeller with back pump out vanes. A 1.750 in balanced, metal bellows seal was used. The rotating seal ring was carbon and the stationary seal ring was nickel-bound tungsten carbide (Pump A).

The other test loop also used a standard 1 × 2-10 ANSI pump with a 3500 rpm drive motor. This pump was equipped with a minimum diameter, semi-open impeller with balance holes. A 1.875 in unbalanced, multiple-spring, pusher-type seal was used in this pump. The rotating seal ring was nickel-bound tungsten carbide and the stationary seal ring was carbon (Pump B).

Each pump/seal combination was run with several different seal chamber designs. Each seal chamber was run in the pump for approximately one week. The data shown in Table 1 was collected at least twice a day. The seal chamber temperature was measured with a thermocouple probe located 0.25 in from the rotating parts as shown in the seal chamber diagrams. The seal face temperature was measured by a thermocouple located in the stationary seal ring 0.060 in from the sealing face.

The Seal Chambers

Four seal chambers were tested in Pump A. They are shown in Figures 2, 3, 4, and 5. Three seal chambers were tested in Pump B. They are shown in Figures 6, 7, and 8. As the figures illustrate, both pumps were tested with an ANSI standard stuffing box, an enlarged cylindrical bore seal chamber and an enlarged tapered bore seal chamber. Additionally, Pump A was tested with an alternate design cylindrical bore seal chamber. All of these seal chambers are commercially available and were not altered.

Table 1. Data Collected.

PRESSURE, PSI	TEMPERATURE, °F
SUCTION	SUCTION
DISCHARGE	DISCHARGE
SEAL CHAMBER	SEAL CHAMBER
	BEARING HOUSING
	AMBIENT

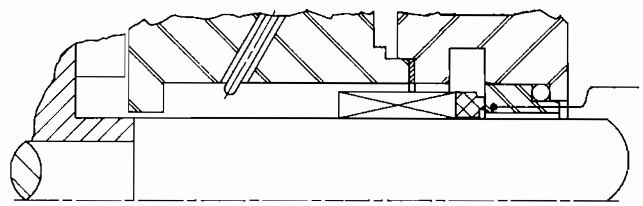

Figure 2. Standard Stuffing Box, Pump A.

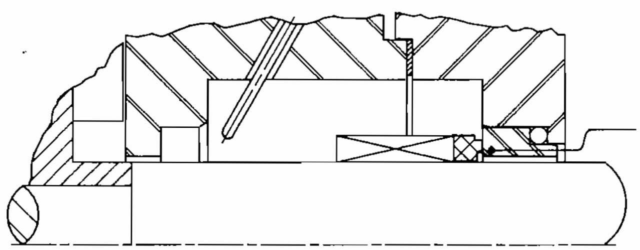

Figure 3. Enlarged Cylindrical I Seal Chamber, Pump A.

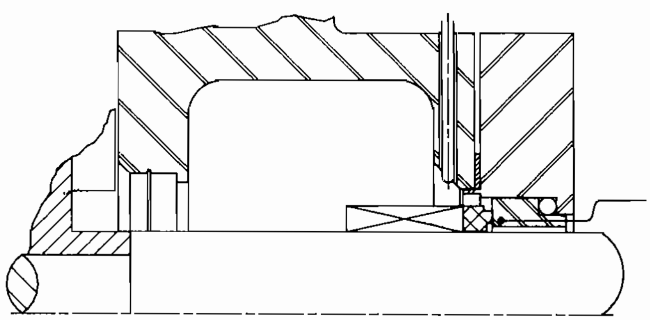

Figure 4. Enlarged Cylindrical II Seal Chamber, Pump A.

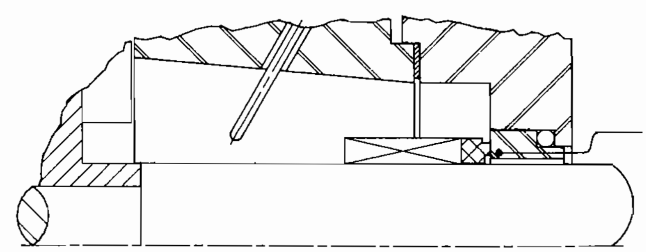

Figure 5. Enlarged Tapered Seal Chamber, Pump A.

RESULTS

Seal performance in an ANSI standard stuffing box versus enlarged bore seal chambers has been investigated previously [1]. The results of these current tests confirm those reported earlier. The new tests also allow us to determine the mechanism of the improvement offered by enlarged bore seal chambers. The results for the tests on Pump A are given in Table 2. The results for the tests on Pump B are given in Table 3. The values in these tables represent steady state operation.

Table 2. Test Results for Pump A Seal Chambers.

SEAL CHAMBER	SEAL CHAMBER TEMP RISE (°F)	SEAL FACE TEMP RISE (°F)	SEAL CHAMBER DIFF PRESS (PSI)
STANDARD	8	15	6
ENLARGED I	10	9	6
ENLARGED II	1	14	11
TAPERED	1	5	18

***Note:** Pump differential pressure was 51 psi for all tests.*

Table 3. Test Results for Pump B Seal Chambers.

SEAL CHAMBER	SEAL CHAMBER TEMP RISE (°F)	SEAL FACE TEMP RISE (°F)	SEAL CHAMBER DIFF PRESS (PSI)
STANDARD	18	52	14
ENLARGED	1	5	5
TAPERED	1	5	12

***Note:** Pump differential pressure was 40 psi for all tests.*

Seal Chamber Temperature

The seal chamber temperature rise is the difference between the product suction and seal chamber temperatures in degrees Fahrenheit. For Pump A, the standard stuffing box and enlarged cylindrical bore seal chamber I had high temperature rises of 8°F and 10°F, respectively. Whereas, both the enlarged cylindrical bore II and tapered seal chambers had only a 1°F rise. The reason for this can be seen in Figures 2 through 5. The first two seal chambers had a restricted throat. This prevented mixing of the product and seal chamber fluid. Therefore, the seal generated heat could not be carried away. When the throat clearance increased, mixing occurred and the seal chamber temperature dropped approaching the product temperature.

The results were similar for the tests on Pump B. The standard stuffing box, with a restricted throat, had an 18°F rise. Both the enlarged cylindrical and tapered seal chambers for this pump had an open throat. Both of them had only a 1°F rise (Figures 6, 7, and 8).

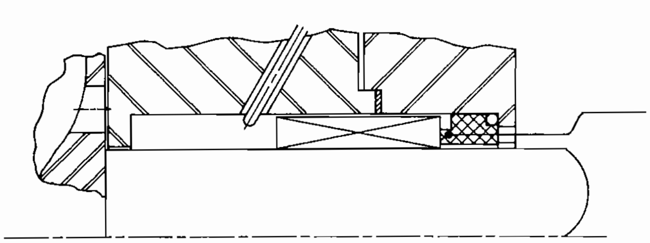

Figure 6. Standard Stuffing Box, Pump B.

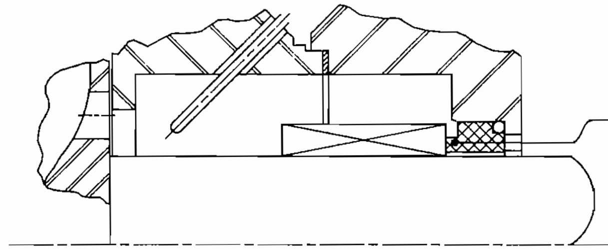

Figure 7. Enlarged Cylindrical Seal Chamber, Pump B.

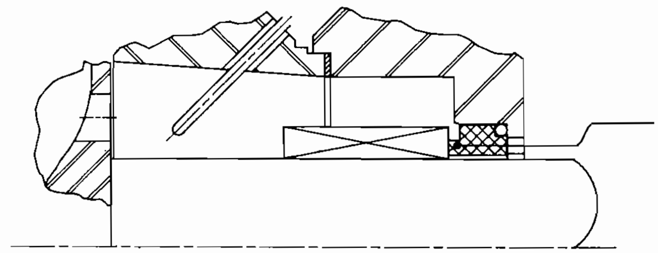

Figure 8. Enlarged Tapered Seal Chamber, Pump B.

Seal Face Temperature

The seal face temperature rise is the difference between the seal chamber and seal face temperatures in degrees Fahrenheit. The results for Pump A show that this was 9°F and 5°F for the enlarged cylindrical I and tapered seal chambers, respectively. The standard stuffing box and enlarged cylindrical II seal chamber were basically the same at 15°F and 14°F, respectively. Again, the reason for this can be seen in Figures 2 through 5. When the seal faces were located in a restricted bore area, the temperature rise was higher than when they were in an open area. This showed that the enlarged seal chambers were effective only when the seal faces were located in the open area. The same phenomenon occurred in Pump B. In the standard stuffing box, the area around the seal was restricted and the temperature rise was 52°F. This temperature rise is greater than those in Pump A because Pump B used an unbalanced seal. In the enlarged cylindrical and tapered seal chambers the seal faces were in an open area and the temperature rise was only 5°F.

Pressure Data

The chamber differential pressure is the difference between the product suction and seal chamber pressures. In Pump A, the standard stuffing box and enlarged cylindrical seal chamber I each had a 6 psi differential. The differential was 11 psi in the enlarged cylindrical II seal chamber and was 18 psi in the enlarged tapered seal chamber. This increase was expected. The impeller in this pump had back pump out vanes. Pump out vanes produce flow and a pressure differential between the seal chamber and pump discharge. The restricted clearance between the vanes and cover affects their performance. In the enlarged cylindrical II seal chamber, part of the vanes were exposed by the throat opening. Therefore, they did not produce as great a differential as the standard stuffing box and enlarged cylindrical I seal chamber. This raised the seal chamber pressure. More of the vanes were exposed in the tapered seal chamber; therefore, the further increased seal chamber pressure.

The tests on Pump B did not follow this trend. Here, the seal chamber differential pressure was 14 psi for the standard stuffing box, 5 psi for the enlarged cylindrical seal chamber and 12 psi for the enlarged tapered seal chamber. The impeller in this pump had balance holes. Balance holes allow more or less unrestricted fluid transfer between the back of the impeller and the pump suction. This tends to equalize the pressure at these two points as long as there is a large enough pressure drop down the back of the impeller. The standard stuffing box had a restricted throat. This limited flow between the stuffing box and back of the impeller and allowed pressure to build up. The enlarged cylindrical seal chamber had an open throat. This allowed the balance holes to keep the seal chamber pressure low. As the throat opened further in the tapered seal chamber, the pressure drop down the back of the impeller decreased. This raised the seal chamber pressure (Figures 6, 7, and 8).

Temperature Excursions

Erratic seal face temperatures were observed occasionally during the testing. To get a better idea of what was happening at the seal faces, several tests were run using a strip chart to record the seal chamber and seal face temperatures. These tests showed that, in some cases, the seals were running very erratically. The results for Pump A are shown in Figures 9 through 12. In all of the figures, the top line represents the difference between the product suction and seal face temperatures. The bottom line represents the difference between the product suction and seal chamber temperatures. As would be expected, repetition of these tests did not yield exactly the same results. However, the pattern was consistent for each seal chamber. There are two things to note in this data—first, the difference in behavior of the seal in the various seal chambers, and second, the seal chamber temperatures.

Data presented in Figures 9 and 11 correspond to tests on the standard stuffing box and enlarged cylindrical II seal chamber.

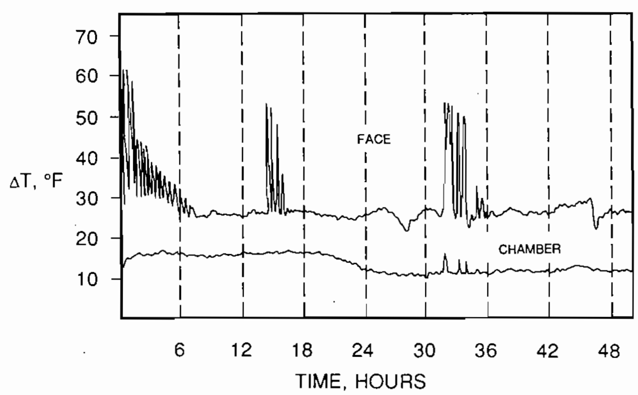

Figure 9. Seal Face and Seal Chamber Temperatures for the Standard Stuffing Box in Pump A.

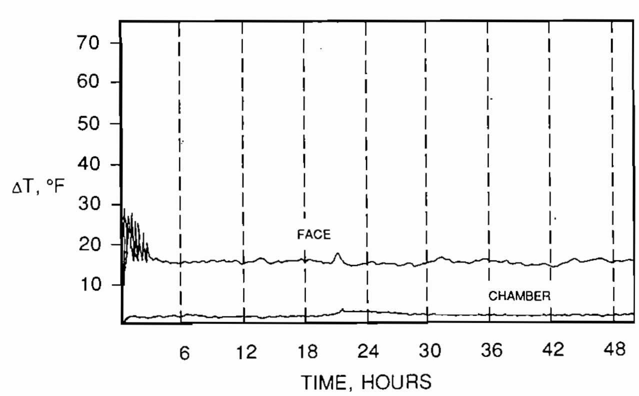

Figure 10. Seal Face and Seal Chamber Temperatures for the Enlarged Cylindrical I Seal Chamber in Pump A.

In both of these seal chambers the faces were located in a restricted area. This meant inconsistent and insufficient cooling. Therefore, the seal faces ran erratically. Part of the instability experienced by the enlarged cylindrical II seal chamber was probably caused by air. The shape of this chamber traps air which can be centrifuged to the seal faces when the pump is started. A special provision for venting was made and resulted in somewhat better seal performance.

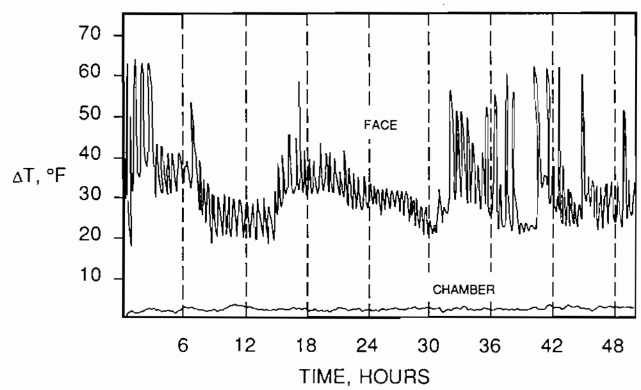

Figure 11. Seal Face and Seal Chamber Temperatures for the Enlarged Cylindrical II Seal Chamber in Pump A.

Results featured in Figures 10 and 12 correspond to tests on the enlarged cylindrical seal chamber I and enlarged tapered seal chamber. The faces were exposed in these seal chambers. This allowed ample cooling of the seal faces. Therefore, the seal face temperatures were very stable. Again, the data shows that seal faces must be located in the open area to benefit from enlarged bore seal chambers.

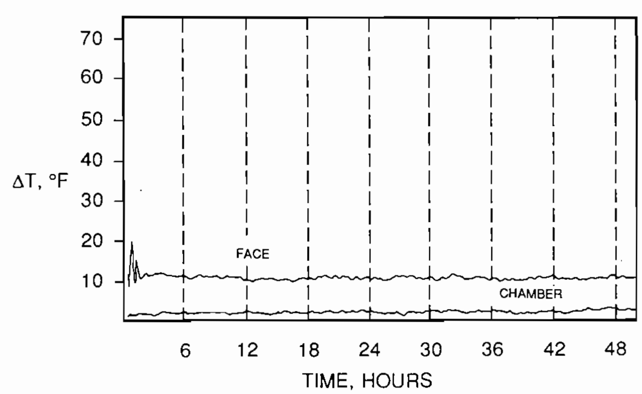

Figure 12. Seal Face and Seal Chamber Temperatures for the Enlarged Tapered Seal Chamber in Pump A.

Note the seal chamber temperatures in Figures 9 through 12. In every case the temperature was very stable. The seal chamber temperature did not indicate the seal face instability.

Although only the data for Pump A is shown, the results for Pump B were similar. The seal face temperature in the standard stuffing box was high and unstable for more than 24 hours. Whereas, the enlarged seal chambers experienced only moderate temperature rises and became stable after the first few hours. In some testing of the tapered seal chambers, no initial spike in face temperature was observed. The seal simply came up to temperature and stayed there.

Supplemental Data

Other data points collected were the bearing housing and ambient temperatures. The bearing housing temperature was affected by the ambient temperature. This was due to the cooling effect provided by the drive motor fan. For example, the ambient temperature ranged from 60°F to 98°F and the bearing housing from 87°F to 116°F on Pump A. The seal faces and seal chambers were not affected by these variations. Also, the product discharge temperature was recorded. In all of the tests this temperature was only 1°F or 2°F above the product suction temperature.

ADDITIONAL TESTING

The initial testing was successful and answered some questions about the advantages of enlarged bore seal chambers. However, the results also created some questions. This led to a second round of testing intended to answer specific questions.

Throat Bushing Clearance

Two additional tests were run on Pump A with the enlarged cylindrical I seal chamber. First, the throat was opened to a 2.000 in diameter. This provided 0.125 in of radial clearance. Next, the throat was opened to a 2.688 in diameter, providing 0.469 in of radial clearance (Figure 13). The results from these tests are given in Table 4. In both modified seal chambers the chamber temperature rise was 1°F. This showed that it was not necessary to open the throat very much in order to cool off the seal chamber. A 2.000 in throat did not significantly change the seal chamber pressure because it did not expose the pump out vanes. The 2.688 in throat began to expose the vanes and the seal chamber pressure increased.

Table 4. Test Results for Alternate Throats on Enlarged Cylindrical I Seal Chamber.

THROAT	RADIAL CLEARANCE	SEAL CHAMBER TEMP RISE (°F)	SEAL FACE TEMP RISE (°F)	SEAL CHAMBER DIFF PRESS (PSI)
STANDARD	.015	10	9	6
2.000"	.125"	1	14	7
2.688"	.469"	1	12	12

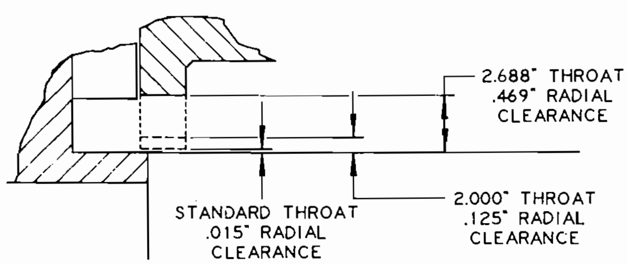

Figure 13. Modifications to Throat of Enlarged Cylindrical I Seal Chamber for Pump A.

In some applications, it may be necessary to isolate the seal chamber from the product. In those cases, a restricted throat is still recommended.

Product Flow Effects

Additional tests were run to determine the effects of product flow on seal performance. In a given centrifugal pump, increasing the impeller diameter increases the differential head and capacity of the pump. For a given impeller diameter, decreasing the differential pressure increases the product flow. Up to this point, all testing had been done on pumps with minimum diameter impellers and on a test loop requiring relatively high differential pressure. Therefore, the product flow was approximately 5.0 gpm (near shutoff conditions). A test circuit was constructed to allow operation of Pump B at its best efficiency point (BEP), with a maximum diameter impeller. A recirculation loop with a large storage tank was used. For these tests, the product flow was approximately 155 gpm. The two operating points are shown on the pump curve in Figure 14.

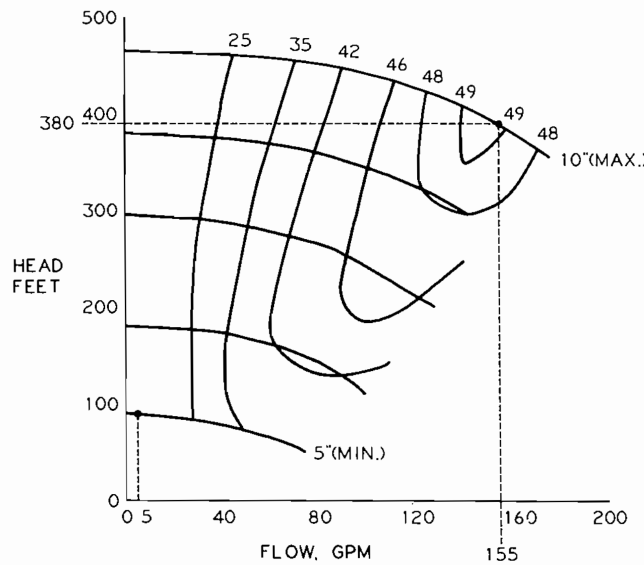

Figure 14. Operating Conditions for Tests Near Shutoff and at BEP for Pump B.

All three seal chambers were tested. The results are given in Table 5 along with the results from the initial testing. The first value shown in the table is from the tests at BEP. The second value is from the tests near shutoff. The results show that increasing the product flow affected only the standard stuffing box temperature rise and the seal chamber pressure differentials.

Table 5. Test Results for Pump B at Different Operating Conditions.

SEAL CHAMBER	SEAL CHAMBER TEMP RISE (°F)	SEAL FACE TEMP RISE (°F)	SEAL CHAMBER DIFF PRESS (PSI)
STANDARD	5/18	50/52	20/14
ENLARGED	1/1	4/5	5/5
TAPERED	1/1	5/5	18/12

Note: *Pump differential pressure was 127 psi for all tests. The first table entry is from tests at BEP, the second from near shutoff conditions.*

The increased product flow and pump differential pressure caused better flow through the restricted throat. This allowed the standard stuffing box to run cooler than in the low flow test. Note, however, that the seal chamber temperature rise did not change in the seal chambers with an open throat. Also, the seal face temperature rise was not affected in any of the seal chambers.

The standard stuffing box and tapered seal chamber pressures also increased. As previously described, the pump differential pressure affects the pressure in these two seal chambers. Therefore, the seal chamber pressure increase was caused by the increase in pump differential pressure.

Product Temperature Effects

All of the tests reported above were run with a product suction temperature of approximately 75°F. Therefore, additional tests were run on the tapered seal chambers to determine whether the product temperature had any effect on temperature rise. For these tests, the product temperature was increased and the temperature rise data was recorded. Pump A was run with a product temperature of 190°F. Pump B was run with a product temperature of 160°F. The results of these tests are given in Table 6. In this table, the increase from suction to seal face temperature is given. This temperature rise was not affected in either pump by increased product temperature. The seal chamber temperature rise data did change, however. In Pump A the seal chamber was actually cooler than the product. This phenomenon has been observed and described by Will [2]. This did not happen in Pump B because of the lower product temperature. In both pumps, the seal chamber and seal face temperatures were more unstable at the increased product temperatures.

Table 6. Test Results for Tapered Seal Chambers at Elevated Product Temperatures.

PUMP	SEAL CHAMBER TEMP RISE (°F)	SUCTION TO SEAL FACE TEMP RISE (°F)
A	1/-15	6/5
B	1/3	6/7

Note: The first table entry is from low temperature tests, the second is from high temperature tests.

Seal Flush Effects

None of the previous tests utilized a seal flush. To determine the difference between using seal flush *vs* an enlarged seal chamber, an API plan 11 flush was installed for several tests. A very liberal flush line was used (0.500 in tubing). The results (Table 7 for Pump A and Table 8 for Pump B) showed very little difference between the flushed standard boxes and the enlarged tapered seal chambers. They also showed that flushing a tapered seal chamber does not offer any real additional cooling. Finally, the chamber pressure was higher in the flushed standard box than in the tapered seal chambers on both pumps. All of this means that a tapered seal chamber is just as effective at cooling the seal faces as using a flush on a standard box.

Table 7. Test Results for Seal Chambers with Seal Flush on Pump A.

SEAL CHAMBER	SEAL CHAMBER TEMP RISE (°F)	SEAL FACE TEMP RISE (°F)	SEAL CHAMBER DIFF PRESS (PSI)
STANDARD (FLUSHED)	2	7	16
TAPERED (NOT FLUSHED)	1	5	12
TAPERED (FLUSHED)	1	4	11

Note: Pump differential pressure was 40 psi for all tests.

Table 8. Test Results for Seal Chambers with Seal Flush on Pump B.

SEAL CHAMBER	SEAL CHAMBER TEMP RISE (°F)	SEAL FACE TEMP RISE (°F)	SEAL CHAMBER DIFF PRESS (PSI)
STANDARD (FLUSHED)	2	8	24
TAPERED (NOT FLUSHED)	1	9	18
TAPERED (FLUSHED)	!	5	20

Note: Pump differential pressure was 51 psi for all tests.

The difference, then, between using a tapered seal chamber or flushing a standard box is the inherent disadvantages of flush piping. For example; the cooling tends to be concentrated around the flush inlet, solids in the flush fluid cause the flush to act as an abrasive jet, flush piping can be a safety hazard, the piping often requires seal welding, insulation, or heat tracing and the piping often requires high point vents. Therefore, in many cases, it would be advantageous to use a tapered seal chamber instead of using flush piping.

CONCLUSIONS

Results of the test program support the assumption that definite benefits can be gained from using the enlarged bore seal chambers. Properly designed seal chambers allow seals to operate cooler and more stable than conventional stuffing boxes. The tests also revealed that seal face operating temperature, not seal chamber operating temperature, is the best indication of seal face performance. In contrast, seal chamber temperature is primarily a function of the product temperature. Thus, the test data suggest that the seal chamber temperature alone obscures rather than clarifies the picture of seal performance or life.

The major goal of seal chamber design is to provide an environment that allows seal faces to run cooler and with greater stability. Based on the test findings, specific mechanisms have been identified and criteria established which directly satisfy that goal. An open throat should be used whenever possible. The seal faces must not operate in a restricted bore. The seal chamber should be self-venting. Finally, in many cases, a tapered bore may be used in lieu of flush piping.

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