A POWERFUL APPLICATION AND TROUBLESHOOTING METHOD FOR MECHANICAL SEALS

by

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ABSTRACT

Understanding the possible thermodynamic paths that a sealed fluid can follow as it goes from stuffingbox conditions, across the seal faces, and then exits to the atmosphere is a basic, but often overlooked, aspect of seal application and troubleshooting. Coupling thermodynamics with hydrostatic stability results in a powerful seal analysis method. Procedures to chart the fluid thermodynamic path and to calculate seal stability enable one to modify pump design, seal flush plans, and seal geometry, to ensure successful seal operation. Possible scenarios and a case history provide working examples.

INTRODUCTION

Pump mechanical shaft seals are the most troublesome general purpose equipment component in refineries and chemical plants from an operational, maintenance and environmental standpoint. They are often treated as a commodity item without the realization that they are the most precision mechanical device in the entire plant, having tolerances on the order of millionths of an inch. A mechanical seal is also a fluid machine in its own right, as its purpose is to transport a fluid from a high pressure region to a lower pressure via an extremely thin film between two sliding surfaces. Thus, fluid thermodynamics play a very important role in the performance of a mechanical seal. The sealed fluid proceeds along a thermodynamic path prescribed by stuffingbox conditions, the seal face heat generation, and exit conditions. Knowing the fluid thermodynamic path will give considerable insight into seal operation and enable one to adjust the seal environment or geometry to improve seal performance. Coupling the thermodynamics with hydrostatic stability criteria results in a powerful analysis method for mechanical seals.

SEAL THERMODYNAMIC STATE POINTS

To perform thermodynamic analyses of mechanical seals, one must first define reference points to use in constructing the fluid thermodynamic path. The point numbers in Figure 1 refer to the thermodynamic state of the pumped fluid at the corresponding physical location for a typical pump and mechanical seal. Point 1 represents the pump suction conditions. Point 2 represents conditions in the pump stuffingbox. Point 3 is located midway across the seal faces. Point 4 is at the seal exit, which is atmospheric pressure for the great majority of single seals. Using these point designations, one can plot the thermodynamic path of a fluid on a pressure vs. temperature (P-T) diagram as it goes from the pump suction to the stuffingbox and then across the seal face (Figure 2).

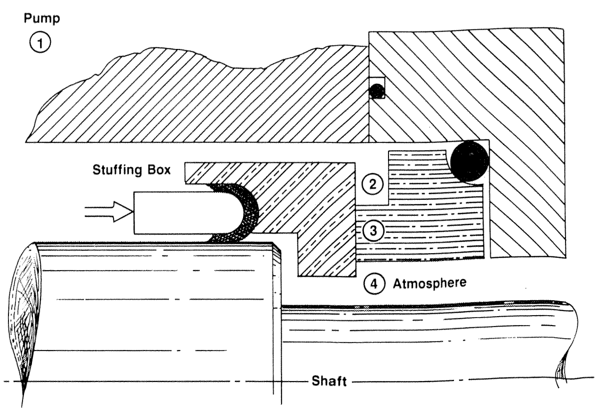

Figure 1. Thermodynamic Reference Points. 1) Pumped Fluid, 2) Stuffingbox, 3) Seal Face, 4) Seal Exit (usually atmospheric).

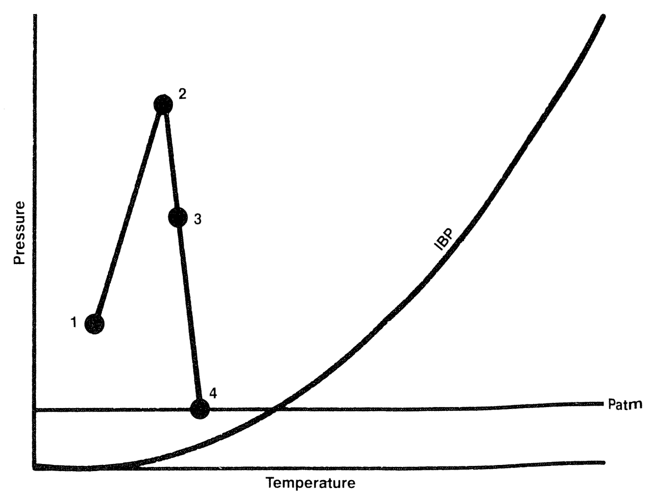

Figure 2. All Liquid Seal. 1) Pumped Fluid, 2) Stuffingbox, 3) Seal Face, 4) Seal Exit.

From a seal success standpoint, the location of Point 2 on the P-T diagram is very important. Point 2 must be in the liquid region on the diagram, i.e., to the left of the IBP (initial boiling point) curve. If Point 2 is in the vapor region, the typical pump mechanical seal will overheat and fail rapidly. For pure fluids, the liquid will flash to vapor as it crosses the IBP curve. For fluid mixtures, which are common in the petroleum industry, not all the liquid will flash to vapor as it crosses the IBP curve. A distillation occurs where the highest vapor pressure components boil off first. Consequently, for mixtures, there are a series of curves representing different fractions of the fluid being vaporized. Experimental work [1,2,3] has shown that the seal will overheat when Point 2 just crosses the IBP curve even though only a small fraction of the liquid (less than five percent), vaporizes. This is due to the large specific volume difference between the liquid and vapor. The volume fraction of vapor will be large, even though the mass fraction of vapor is small. Therefore, when analyzing mixtures, Point 2 must still lie to the left of the IBP curve for successful operation.

The locations of Points 3 and 4 on the P-T diagram are not as crucial as Point 2. They can, and sometimes must, fall in the vapor region to the right of the IBP curve. Experience has shown that seal life improves as Point 2 moves further into the liquid region, i.e., to the left of the IBP curve. It follows then that longer seal life results as Points 3 and 4 move to the left and the point of liquid vaporization moves towards the seal face exit. The exact reason for this is not known, but one can hypothesize that the more liquid there is between the seal faces, the better will be the lubrication and thermodynamic stability.

SOME TYPICAL THERMODYNAMIC PATHS

A typical cool service where the seal is operating in an all liquid state is illustrated in Figure 2. Point 1 shows the pumped fluid to be well below its boiling point. Point 2 indicates stuffingbox conditions resulting from a pressure rise, e.g., due to pump construction or discharge flushing (API Plan II) [4], and a temperature rise due to seal heat generation. Point 3 is midway across the seal face and indicates a pressure drop and more temperature rise from seal heat generation. Point 4 is at the seal exit which is at atmospheric pressure and still within the liquid region for the fluid. Hence, any leakage will be all liquid. This path represents an ideal and relatively easy sealing duty.

A service where the pumped fluid is at a temperature which is close to, but still below, its atmospheric boiling point, i.e., if the pumped fluid is spilled to atmosphere, it will not flash, is shown in Figure 3. This can be determined by following a constant temperature line from Point 1 to the atmospheric pressure. If the line does not cross the IBP curve, the fluid will not vaporize as its pressure is dropped to atmospheric conditions. If the line does cross the IBP curve, then all or part of the fluid must vaporize, depending on whether the fluid is a pure substance or a mixture, e.g., propane or kerosene. A slight pressure rise and a relatively high temperature rise is shown at Point 2. The high temperature rise illustrates what happens if the seal is operated deadended (API Plan 2), and has a wide face (0.25 in) which generates so much heat that the fluid temperature near the seal is significantly increased [3]. Point 3 is across the IBP curve and hence is in the vapor region. Point 4 also lies in the vapor region. Point 2′ would be for a narrow face seal (0.1 in) which generates less heat. Point 3′ is still in the liquid region for the narrow face, with Point 4′ being in the vapor region. The difference in the two paths shows that vaporization occurs closer to the exit of the narrow face seal than for the wide face seal. Vaporization occurs before the face midpoint for the wide face seal and after the midpoint for the narrow face seal. The two paths show that both types of seals

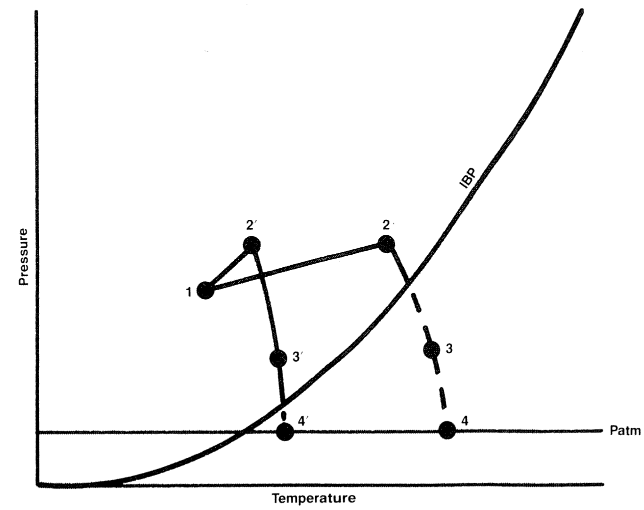

Figure 3. Seal with Vaporization Potential. 1) Pumped Fluid, 2) Stuffingbox, wide face seal, 3) Seal Face, wide face seal, 4) Seal Exit, wide face seal. 2′) Stuffingbox, narrow face seal, 3′) Seal Face, narrow face seal, 4′) Seal Exit, narrow face seal.

will work for this service, but the narrow face will operate cooler and have a longer life.

An example of a seal operating with the fluid at suction conditions above its atmospheric boiling point is presented in Figure 4. Using a wide face seal would result in Point 2 being located in the vapor region, which would cause seal damage from dry operation and overheating. Using a narrow face seal, Point 2′, keeps the stuffingbox fluid in a liquid state and makes the difference between success and failure.

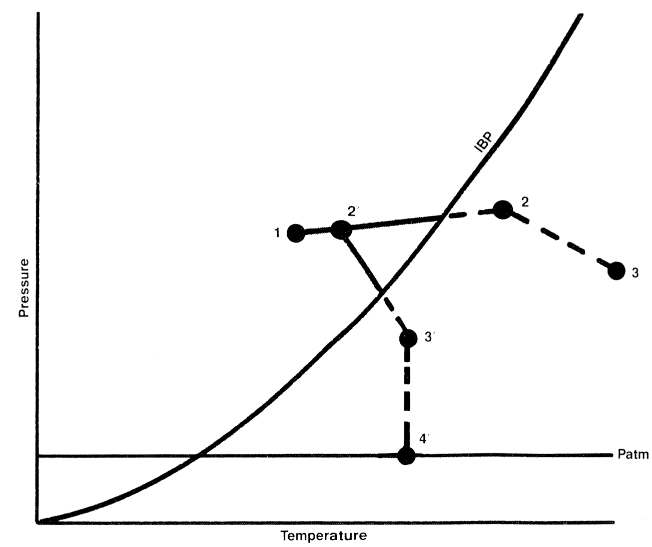

Figure 4. Sealed Fluid Above its Atmospheric Boiling Point. 1) Pumped Fluid, 2) Stuffingbox, wide face seal, vaporization and failure, 3) Seal Face, wide face seal, high temperature, 2′) Stuffingbox, narrow face seal, 3′) Seal Face, narrow face seal, 4′) Seal Exit, narrow face seal.

An elevated temperature fluid is again present on the P-T diagram in Figure 5. The difference from Figure 4 is that the wide face seal vaporization problem is solved by increasing the stuffingbox pressure, which results in Point 2 returning to the liquid region to the left of the IBP curve. This can be done by plugging impeller balance holes, or by using a discharge flush with a close clearance throat bushing in the stuffingbox. Such action has apparently solved the problem, but one must also make certain that the increased pressure has not exceeded the stuffingbox pressure capability or seal stability pressure limit [5]. The narrow face seal, as shown in Figure 4, is always the better solution.

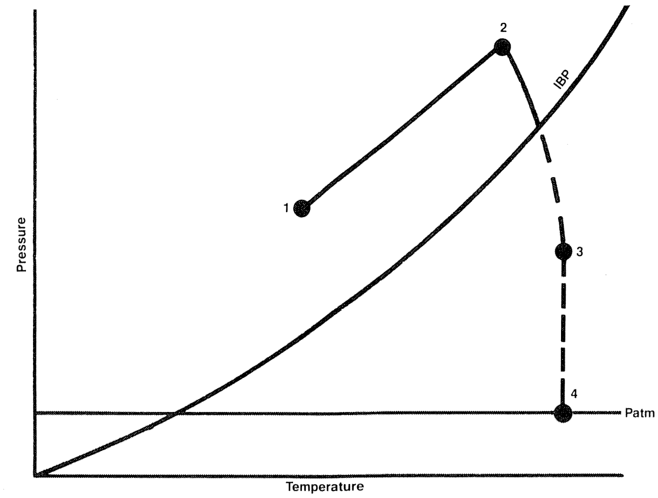

Figure 5. Wide Face Vaporization Solved by Stuffingbox Pressure Increase. 1) Pumped Fluid, 2) Stuffingbox, 3) Seal Face, 4) Seal Exit.

THERMODYNAMIC AND HYDROSTATIC STABILITY ANALYSIS

The following procedure can be used to determine the thermodynamic path (Figure 6) and check the hydrostatic stability for any sealing duty. The resulting information can then be used to determine what options are available, including flush plans, cooling, or pump modifications, to correct the seal environment as needed. One may find that in some cases the problem cannot be solved within the existing pump! In pumping services that are known to be difficult sealing duties, it is wise to put the proverbial cart before the horse by designing the seal installation first and then buying a pump to suit both the seal and the process.

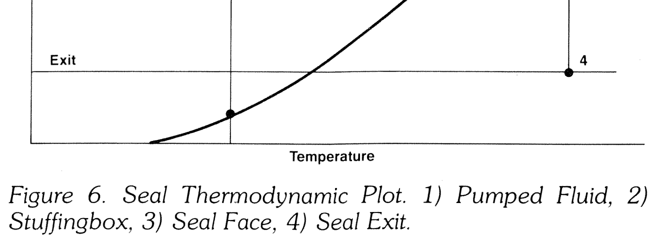

Figure 6. Seal Thermodynamic Plot. 1) Pumped Fluid, 2) Stuffingbox, 3) Seal Face, 4) Seal Exit.

The first step in analyzing a seal is to determine the operating conditions that the seal must withstand. These conditions include the pump suction, discharge and seal exit pressures, pump temperature and pumpage IBP curve. For most single seals and the outboard seal of tandem and double seals, the seal exit is at atmospheric pressure. For seals using an external flushing fluid, the external fluid temperature and fluid properties for the IBP curve must be known. For a seal in tandem or a double seal arrangement that is sealing barrier fluid, rather than process fluid, the barrier fluid temperature and IBP curve properties must be determined. If the needed IBP curve properties are unknown, laboratory data or published references [6,7,8] should be used.

The intersection of the suction pressure and the pump temperature is Point 1, the starting point for the analysis. If the pumping temperature/exit pressure intersection is to the right of the IBP curve, then the seal leakage must be a gas, and the chance exists for cooling the seal due to auto-refrigeration, if there is excessive seal leakage. If the boiling point of the leakage at atmospheric pressure is low enough, frost may accumulate on the gland plate.

To locate the Point 2 pressure, use a measured value, if at all possible. The lantern ring connection on the stuffingbox or the flush port on the gland plate makes good access points for pressure measurement. In refineries and chemical plants, however, the lantern ring connection is often plugged and backwelded, and the flush port is already utilized for a flushing plan. For such cases, especially on problem pumps, specify that two flush ports be provided on the gland plate, so that one can be used for accurate stuffingbox pressure measurement. For a seal in a tandem or double seal arrangement that is sealing the barrier fluid, use the barrier fluid pressure in place of the stuffingbox pressure. If a measured value is not available, then estimate the stuffingbox pressure by examining the pump cross sectional drawing and flush plan being used. Look for features such as internal or external stuffingbox pressure balance lines or ports, impeller balance holes, close clearance throat bushings, and unusual impeller arrangements which will affect the stuffingbox pressure. Remember, the analysis will be only as good as the input data.

The determination of the temperature at Point 2 depends on the seal utilized. For a narrow face seal (0.1 in) add 2°F to the pumping temperature. Wide face seals (0.25 in) generate more heat, adding as much as 50°F to the pumping temperature. Use linear interpolation or extrapolation for other face widths. If Point 2 is to the right of the IBP line, the seal will not be successful. Change the flush or the face width to get Point 2 to the left of the IBP line. Point 2 is the most critical point.

The pressure at Point 3 is the average of the pressure at Point 2 and the seal exit pressure. The seal exit pressure for most seals is atmospheric pressure. The temperature at Point 3 depends on the seal, with narrow face seals adding 2°F to the Point 2 temperature, while wide face seals increase the temperature 40°F. Point 3 may fall to the right of the IBP curve, which is not necessarily a problem. It means that some vaporization will occur across the seal face.

If Points 2 and 3 are to the left of the IBP curve, continue with the slope from Point 2 to Point 3, to the seal exit pressure. If Point 3 is to the right of the IBP curve, Point 4 will be located at the Point 3 temperature and the exit pressure. Point 4 will determine if the leakage is liquid or gas.

To complete the analysis, the seal hydrostatic stability must be checked using Equation 16 by Buck [5]:

$$\Delta P_{max} = \frac{P_{sp}}{1-B} \tag{1}$$

The equation gives the maximum pressure rating for a stable seal. If the difference between the stuffingbox pressure and the seal exit pressure is greater than ΔP_{max}, the seal has the potential for popping open and leaking, or exhibiting "puffing" behavior. Seal face width, spring force, and balance ratio can be adjusted to correct the situation, if necessary, using the following equations:

$$B = \frac{D_o^2 - D_b^2}{D_o^2 - D_i^2} \tag{2}$$

$$D_o = 2WB \left[1 + \sqrt{1 - \frac{1}{B} + \left(\frac{D_b}{2WB}\right)^2} \right] \tag{3}$$

$$D_i = D_o - 2W \tag{4}$$

$$D_b = \sqrt{D_o^2\ (1-B) + (B)\ D_i^2} \tag{5}$$

CASE HISTORY

The following case history illustrates how to construct and use the thermodynamic path and hydrostatic stability to troubleshoot an actual seal problem.

Service	Depropanizer Reflux
Fluid	Light Alkylate
Suction Pressure	275 psig
Suction Temperature	110°F
Discharge Pressure	330 psig
Specific Gravity	0.47
IBP Data	−43° @ 0 psig
	62°F @ 90 psig
	110°F @ 185 psig
	135°F @ 255 psig
	158°F @ 330 psig
Flush Plan	API 610/Plan 11 with close clearance throat bushing
D_o	2.438 in
D_i	1.895 in
D_b	2.000 in
B	0.83
P_{sp}	16.2 psi
Stuffingbox Pressure	Estimated at 310 psig
Pump Type	Single stage overhung, no impeller balance holes
Pump Speed	3600 rpm
Problem	Puffing, frosting of gland plate, seal life one to three weeks

Referring to Figure 7, the thermodynamic path was constructed using the following step-by-step procedure.

Step 1

Draw convenient axes from 0 psia to 350 psia and from 0° to 250°F.

Step 2

Using the given data, plot the IBP curve which defines the gas and liquid regions.

Step 3

Draw horizontal lines at atmospheric, suction, discharge and seal exit pressures. In this case, seal exit and atmospheric pressure are identical. Draw a vertical line at the pumping temperature of 110°F. The intersection of the 110°F line and the 275 psig line is Point 1, the starting point. Because the pumping temperature line intersects the atmospheric pressure line to the right of the IBP curve, this pump has a potential for frost buildup. This potential has been verified by field observations.

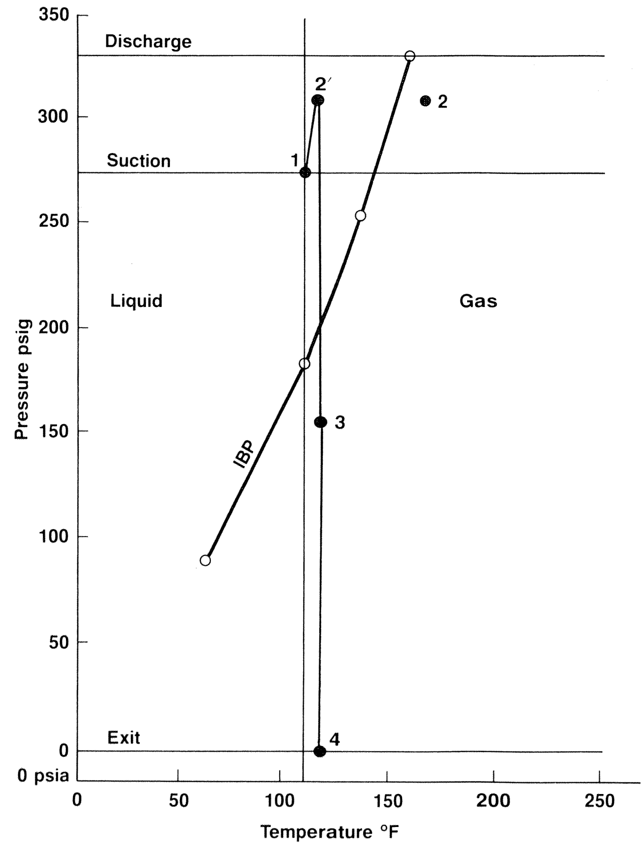

Figure 7. Case History. 1) Pumped Fluid, 2) Stuffingbox, vaporization and failure, 2') Stuffingbox, seal face width reduced, 3) Seal Face, 4) Seal Exit.

Step 4

To locate Point 2, use the estimated stuffingbox pressure of 310 psig. Using linear extrapolation, find the temperature increase for an 0.27 in wide seal, which for this case yields 56°F. This puts the temperature coordinate for Point 2 at 110°F + 56°F or 166°F. Plotting Point 2 shows that it is to the right of the IBP curve (in the gas region) and has little chance for success. Field experience verifies a high failure rate.

Proceeding further at this point is fruitless without changing the seal environment or design. Checking to see if the stuffingbox pressure can be raised to move Point 2 to the left of the IBP curve, one finds that even if the full discharge pressure of 330 psig were somehow available in the stuffingbox, possibly via flush plan modifications, Point 2 would still be in the gaseous region by 8°F (166°-158°F). Cooling of the flush would not be feasible, because the pumping temperature is too close to the ambient and cooling tower water temperatures. The seal as supplied simply will not work in this pump. Therefore, the seal design must be changed. Cutting the face width down to 0.100 in yields Point 2' at 310 psig and 112°F, which is satisfactory. Face widths somewhat larger than 0.100 in would also be satisfactory. However, when going through the trouble of making a modification, it is best to gain as much advantage as possible.

Step 5

To locate Point 3, take the average of stuffingbox and exit pressures, which in this case is 155 psig, and for the narrow seal face add 2°F to the temperature at Point 2, which gives 114°F. Point 3 is in the gaseous region and is acceptable, because it occurs between the seal faces.

Step 6

To locate Point 4, keep the same temperature as Point 3, because Point 3 is already in the gaseous region, and drop down to the exit pressure. Point 4 is therefore at 114°F and 0 psig. The leakage from the seal will be a gas. If excess leakage occurs, the leakage temperature will approach the atmospheric boiling point of -43°F and frost will accumulate on the gland plate.

Step 7

Checking the seal stability using Equation (1), it is found that the maximum stable pressure for the original seal is 95.3 psi. Because this is much less than the 310 psi seal differential pressure, the seal is unstable. The instability has been confirmed by field observations. The results from Steps 1 through 6 show that the face width should be cut to 0.100 in to generate less heat. Because of the seal face width reduction, P_{sp} will increase by a factor of almost 3. Keeping the original balance ratio of 0.83, and using Equations 3 and 4, the new seal outside diameter is 2.165 in and inside diameter is 1.965 in, yielding a new P_{sp} of 46.2 psi. This gives a seal maximum pressure of 272 psi, which is still too low. Increasing the balance ratio to 0.88 as the next step, and maintaining a 0.100 in face width, yields a new outside diameter of 2.175 in, inside diameter of 1.975 in, and a maximum pressure of 383 psi, which is acceptable.

The seal was modified per the above analysis and the life of the seal was increased to greater than a year.

DISCUSSION

From the case history, one can see the value of the thermodynamic and hydrostatic stability analysis procedure. Each pump and seal situation is different and requires individual attention. Generalizations about the best seal design for a class of services are useless. All the details must be known and analyzed before a definitive seal design can be recommended and installed with any assurance of success. Often, the seal design required for a particular duty may not be available among standard seal vendor offerings and, thus, requires special engineering. For known troublesome areas, such as fluids pumped near their boiling points, it is wise to evaluate the seal design first, and then select a pump design that can provide the proper seal environment.

NOMENCLATURE

B	Seal geometric balance ratio
D_b	Seal balance diameter
D_i	Seal face inside diameter
D_o	Seal face outside diameter
ΔP	Pressure differential across the seal face
ΔP_{max}	Maximum pressure rating for a stable seal
P_{sp}	Spring force per unit face area
W	Seal face width

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