FACE SEAL BALANCE RATIO SELECTION FOR TWO PHASE SINGLE AND MULTICOMPONENT MIXTURES

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From 1971 to 1987, he served on the faculty of the Mechanical Engineering Department at the University of New Mexico, both as Professor and Chairman of the Department. During this time, he started a mechanical seals research program under the sponsorship of NSF and the U.S. Navy. This work served as a basis for numerous papers, reports, and inventions. A seal test program was started in 1978, which continues to the present.

In 1987, he started his own company and continues to pursue research, development, writing, and consulting in the seals area.

ABSTRACT

It is well known that the fluid pressure distribution between the faces of a mechanical seal greatly affects seal performance. In cases where a sealed fluid changes phase from liquid to vapor as it leaks across the seal faces, the load support derived from fluid pressure can be considerably greater than for the all liquid case. Furthermore, unlike for all liquid seals, this load support changes with seal temperature. Thus, using seals with balance ratios selected based on liquid applications can result in excessive leakage. On the other hand, the selection of balance ratio at too high a level can cause excessive wear of the seal faces by forcing all vapor operation. Thus, balance ratio selection is critical to the successful operation of a two phase seal. The question of how to select balance ratio for two-phase applications has not been clearly resolved.

A simplified method of balance ratio selection for two phase applications is presented herein. By assuming isothermal faces, it is shown how the equilibrium face temperature can be calculated by iterative methods. Equations are presented which are used to calculate critical balance ratio. There is a minimum critical balance ratio and maximum critical balance ratio corresponding to the two undesirable operating conditions above. The method is applied first for the case of water where it is shown how a balance ratio selection graph can be constructed which relates critical balance ratio to pressure and temperature. One graph covers applications over all temperatures and several pressure values. The technique is then extended to a more typical refinery product stream, a two component solution of propane and methane. The behavior of this solution in a two phase seal application is somewhat different than the single

component. Studies are made which show how critical balance ratio is affected by changes in design, specifically face width, temperature coefficient, secondary seal type, and radial taper. The results illustrate the sensitivity of critical balance ratio to both design parameters and to operating conditions.

In conclusion, it is shown that knowledge of the product stream, coupled with calculations of critical balance ratio, as outlined herein, can lead to a rational choice for seal balance ratio.

INTRODUCTION

Background

Mechanical face seals (Figure 1) are commonly used in process pumps to provide the necessary seal between the shaft and the pump housing. Fluids sealed by such devices range from hot crude oil to water to light end hydrocarbons. The life and reliability of mechanical seals is an important factor in the economical maintenance of pumps in process plants of all types. There is a strong economic incentive to find ways to improve seal reliability and increase average life. This paper addresses this issue by proposing a methodology to better select seal balance ratios for two phase applications so that the life and reliability of such seals can be improved.
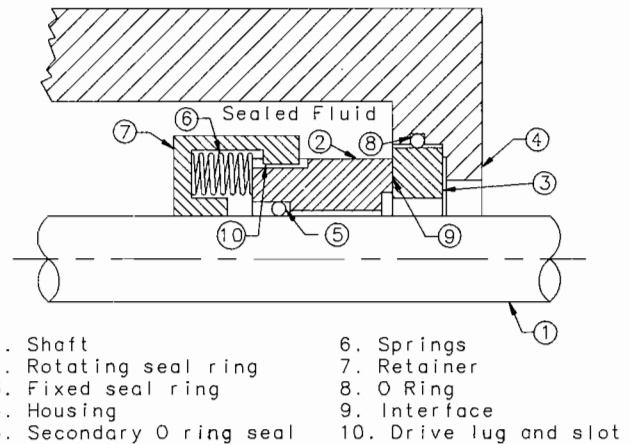

1. Shaft
2. Rotating seal ring
3. Fixed seal ring
4. Housing
5. Secondary O ring seal
6. Springs
7. Retainer
8. O Ring
9. Interface
10. Drive lug and slot

Figure 1. Mechanical Face Seal.

Seal applications where a phase change occurs between the faces as the fluid pressure decreases across the seal are common. Typical examples are the sealing of hot water and light hydrocarbons. A phase change in the interface of a seal causes the pressure distribution to be substantially different than for a single phase gas or liquid. The effect of the phase change on seal mechanics can be dramatic. Sometimes the seal may blow out entirely and sometimes life can be considerably shortened.

The effect of operating a parallel face mechanical seal in an environment where the phase change is shown in Figure 2. In the normal liquid seal, the pressure drop across the face is approximately linear for parallel faces. For a single component fluid, when a phase change occurs, liquid exists over a large fraction of the seal face. As the pressure drops across the seal, at some point (radius) the liquid reaches saturation conditions. At that point, the liquid must change entirely to a gas (for a pure fluid). Then, superheated vapor flows from this transition point to the inner radius. The consequence of the two phase flow is that the amount of load supported by fluid pressure is generally much larger than that normally supported by an all liquid or all gas interface fluid. In fact, if the balance ratio of the seal is set too low, the phase change will cause the seal to lift off uncontrollably.
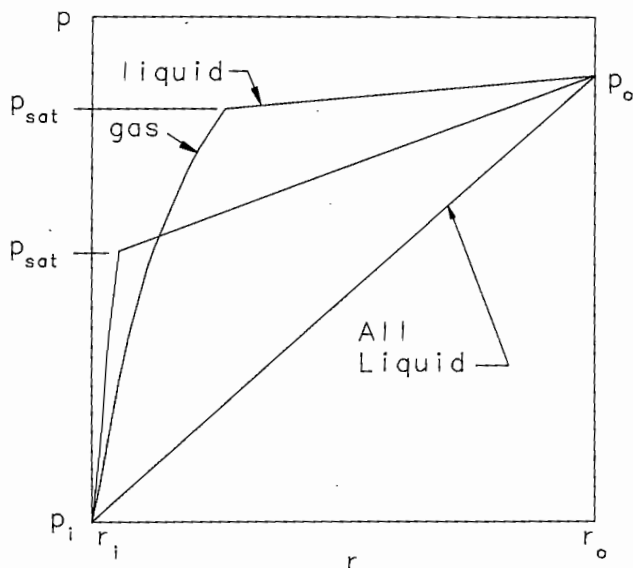

Figure 2. Pressure Distribution Under Two Phase Conditions.

Previous Work

Several different types and levels of models which predict the effects of two phase flow have been developed. The first known model developed for predicting the instability in two phase sealing was provided by Lymer [1]. Lymer estimates the temperature rise of the seal face using a simple heat transfer model and then compares this to the saturation temperature at the sealed pressure. He finds an experimental relationship between this temperature difference and the instability of the seal. Hughes, et al. [2], perform extensive modelling of the seal interface and find that there are in general two values of film thickness corresponding to a given load, only one of which is a stable solution. Hughes, et al., also find that the temperature variation radially across the seal is not large as is confirmed by some recent measurements by Will [3]. In a later work, Hughes and Chao [4] evaluate isothermal and adiabatic models for two phase flow using real fluid properties. They find, for small film thicknesses, that the isothermal model is more accurate than the adiabatic model. Lebeck [5] proposes a two phase seal system model based on a detailed solution for film thickness and face temperature distribution, which predicts seal performance and the instability point.

Considering experimental work, Lebeck and Chiou [6] report on the results of experiments in water run by Chiou [7]. They show that friction decreases with increasing temperature up to a point where friction suddenly increases. They show that this behavior can be predicted using the model by Lebeck [5] mentioned previously. More recently, Will [3] reports on the results of seal tests run using kerosene as the working fluid. Will ran tests over a range of pressure and temperature conditions. He shows by experiment that the temperature rise of the face above its surroundings is reduced considerably by using a more narrow faced seal.

Need

In all of the work noted previously, no practical guidelines for the selection of seal balance ratio or guidelines for seal design have been developed (except for the case of Lymer [1] which is of limited use). Furthermore, all of the theoretical and experimental work has been for a single component fluid. No multicomponent fluids have been evaluated. Thus, studies of balance ratio in two phase seal operation and the application of the two phase models to many important process fluids have not been made.

Concerning balance ratio, as matters stand now, the seal manufacturer chooses the balance ratio as a part of the design and may recommend that a seal have a greater or lesser balance ratio for different applications. API 610 [8] influences this decision by requiring that balanced seals be used on process duty pumps, but the amount of balance is not specified. As will be shown later, for a given fluid composition and process conditions, there exists a stable balance ratio range for a particular seal design. A lower balance ratio may lead to excessive leakage; a higher balance ratio may lead to excessive wear. Thus, the precise value of the balance ratio is very important.

Given that the stable balance ratio range is specific to a seal design and application, there is a need for a relatively simple method by which such optimum balance ratios can be determined. This tool can be used by either the seal user or the seal manufacturer to obtain the information needed to specify the optimum balance ratio for each application. The information can also be used to evaluate seal operation problems. A method of evaluating the practical limits of balance ratio for a specific application is developed herein.

MODEL

Assumptions

The essential parameters which enter into the model are defined in Figure 3. The seal environment temperature and pressure are assumed to be known. Specifics of the materials and geometry of the seal are also assumed to be known.

In order to develop the simplest possible model which still is meaningful, some strong but well supported assumptions are made.

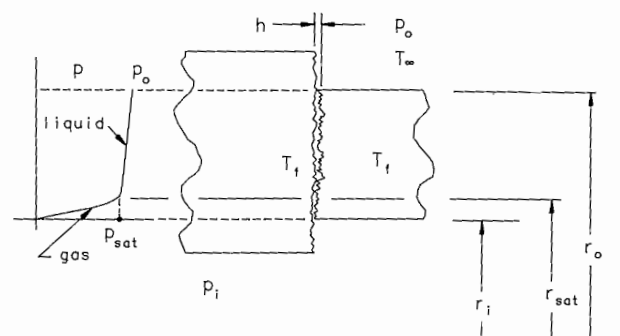*Assumption one.* The faces are parallel.

Figure 3. Parallel Face Model Definitions.

Assumption two. The faces are rough and mechanical contact supports whatever fraction of the load which is necessary for equilibrium. A flow gap always remains which is characterized by the surface roughness of the two faces.

Assumption three. The seal faces and fluid between them are isothermal.

Assumption four. For a single component fluid, phase change occurs over a very short radial distance. The phase change is from 100 percent liquid to 100 liquid vapor.

Assumption five. The vapor phase behaves as an ideal gas.

Assumption six. Axisymmetry applies.

Assumption one is justified on the basis that after some period of operation and wear at steady state conditions, a condition which can be represented by a parallel face model exists, at least for a contacting type of seal of interest here. Of course in the event of transient operation causing thermal radial taper, this assumption would be inadequate. The effect of radial taper is evaluated later.

Assumption two recognizes that many process seals are operating with a very thin film and in fact contact slightly at least during part of their operation. If the hydrostatic pressure is not sufficient to carry the entire load, then the balance of the load must be carried by contact or hydrodynamic effects or both. It is most likely that when the fluid is a liquid, there is a hydro-dynamic like component which provides some of the load support [9, 10]. If the net load is in fact carried by hydrodynamic forces, then the proposed model has some error, but the error will be on the conservative side for critical balance ratio selection. There is reason to believe, however, that for low viscosity, heavily loaded seals that the load is carried by partial contact [11]. When the fluid is a gas, the validity of the assumption is clear. Hydrodynamic effects are minimal and contact forces have to make up the difference. In either case, the film thickness is assumed to be that associated with surface roughness, but this really only affects estimated leakage and is not important otherwise.

Assumption three is validated by the work of Will [3]. He shows on the average a negligible radial variation of the face temperature. Finally, according to Hughes and Chao [4], for thin films, the temperature is controlled by the conduction at the faces and the ample viscous dissipation available to evaporate the small flow so that the change in enthalpy of the fluid as it crosses the face does not effect the temperature significantly.

Assumption four is also made by Hughes and Chao [4]. For a single boiling point fluid, this assumption must follow because in order to have a continuity of flow radially across the seal, there must always be a pressure gradient. Therefore, the region of constant pressure as required for a mixture of liquid and vapor for a single boiling point fluid cannot exist. For fluids that boil over a range of temperatures, this assumption is not used.

Assumptions five is made to keep the model as simple as possible. One can eliminate this assumption, but it is thought that error caused by this assumption is small relative to the errors caused by some of the other essential simplifications.

Assumption six is fundamental to the model. It is well known that at least the predominant hydrostatic mechanisms are associated with geometries which are purely axisymmetric. Even though non axisymmetric effects do occur in the form of waviness and non uniform temperatures, these are considered secondary in a well run-in seal. The equivalent of hydrodynamic lubrication has been shown to occur in the absence of waviness [10].

General Theory

The axisymmetric representation for this simple model is shown in Figure 3. Volumetric laminar flow in the small gap in the radial direction for a complete circumference is given by,

$$q_r = -\frac{\pi r h^3}{6\mu} \frac{dp}{dr} \tag{1}$$

For this problem, it is more useful to use mass flow rate rather than volume flow rate. Thus, multiplying Equation (1) by the density gives radial mass flow rate which must be constant at all radii,

$$\dot{m} = const = -\frac{\pi}{6} \frac{r\rho h^3 dp}{\mu \; dr} \tag{2}$$

By assuption three, mentioned previously, the temperature across the face is assumed constant. Furthermore, to make the subsequent calculations, it is assumed that the face temperature is known. Then, for any fluid, the density and the viscosity may be found as functions of the pressure at the given temperature. The film thickness as a function of radius must also be known. Thus, given the temperature and the functions,

$$T_f, \rho(p,T), h(r), \mu(p,T) \tag{3}$$

then Equation (2) can be integrated to find the pressure distribution $p(r)$ which satisfies the two boundary conditions,

$$p(r_o) = p_o, \; p(r_i) = p_i \tag{4}$$

The correct mass flow is found as a part of this solution.

While not necessary for the general case, for the solutions herein it is assumed that the film thickness is constant at a value equal to the Just touching condition [9, 10] such that the film thickness can be taken as approximately as,

$$h = 3\sigma \tag{5}$$

where σ is the combined RMS roughness of the two surfaces. Given a solution to Equations (2) and (4),

$$\dot{m}, p(r) \tag{6}$$

the amount of load supported by fluid pressure is given by,

$$W_f = \int_{r_i}^{r_o} 2\pi r p \, dr \tag{7}$$

The amount of load which must be supported at the faces for equilibrium is,

$$W^* = \pi(r_o^2 - r_i^2) \left[Bp_o + (1 - B)p_i + p_{sp} \right] \tag{8}$$

for an outside pressurized seal where balance ratio is defined by,

$$B = \frac{r_o^2 - r_b^2}{r_o^2 - r_i^2} \tag{9}$$

O-ring friction is neglected here, but its effects are considered later.

If at this point in the calculation, the load by fluid pressure is greater than the applied load,

$$W_f > W^*, \text{ unstable} \tag{10}$$

then the operation is unstable. That is, the seal cannot in fact operate under the assumed condition.

To find out if the assumed face temperature is correct, first the friction power due to viscous friction is calculated,

$$P_f = \int_{r_i}^{r_o} 2\pi \frac{r^3 \omega^2 \mu}{h} \, dr \tag{11}$$

Then, assuming that the seal faces are touching, the contact force is equal to the difference between the applied load and the fluid pressure supported load,

$$W_m = W^* - W_f \tag{12}$$

Friction power due to contact is given by,

$$P_m = W_m f_c r_m \omega \tag{13}$$

where f_c is the friction coefficient associated with contact of the two face materials under ideal non-hydrodynamic conditions.

Finally, the temperature of the face is given approximately by the product of the total friction power and a constant plus the temperature of the fluid surrounding the seal,

$$T_f = T_\infty + (P_f + P_m)K_{\Delta T} = F(T_f) \tag{14}$$

The constant $K_{\Delta T}$ can be determined by performing a heat transfer analysis on a particular seal design. It depends on the thermal conductiveness of the face materials, the convection coefficient which in turn depends on the speed and sealed fluid, and the geometry of the design. Calculations similar to those shown by Lebeck [12] can be used. Any method of two dimensional heat conduction coupled with convection boundaries and a uniform heat source at the interface can serve as the basis for this calculation. The constant is properly interpreted as the amount of temperature rise of the face above the surroundings per unit of friction power at the face.

Given that a specific value of T_f has to be assumed to make all of the above calculations, therefore, Equation (14) indicates that T_f is a function of itself. This equation may be solved for values of T_f, which make the assumed value to T_f equal to the calculated value given by (14). Such solutions represent equilibrium solutions and represent the model's prediction of the conditions under which actual operation will occur. This sequence of calculations and the equilibrium values of T_f serves as the basis for all subsequent studies herein.

Single Component

The above equations can be solved explicitly for the pressure, load support, mass flow, and phase change radius for a single component fluid which has a well defined boiling point.

For the liquid phase between the outer radius and some phase change radius r_s, Equation (2) may be written as,

$$\dot{m}_l = - \frac{\pi r p_l h^3}{6\mu_l} \frac{dp}{dr} \tag{15}$$

For the gas phase, it is useful to use the perfect gas law to express the density in terms of pressure.

$$p/\rho = RT \tag{16}$$

Thus, the mass flow rate for the gas phase of the seal is given by,

$$\dot{m}_g = - \frac{\pi r h^3 p}{\mu_g RT} \frac{dp}{dr} \tag{17}$$

Assuming that the viscosities are constant with r, the liquid density is constant, and that the seal width is small such that the variable radius can be replaced by the mean radius, both of the above equations can be integrated by separation of variables. The results are,

$$p_s - p_o = \frac{6\dot{m}_l \mu_l}{\pi \rho_l h^3 r_m}(r_o - r_s) \tag{18}$$

$$p_i^2 - p_s^2 = \frac{12\dot{m}_g \mu_g RT_f}{\pi h^3 r_m}(r_s - r_i) \tag{19}$$

where p_s and r_s are the pressure and radius at saturation conditions. Given the isothermal, known temperature assumption, the saturation pressure is known,

$$p_s = f(T_f) \tag{20}$$

Of course, if saturation pressure is not between the inside and outside pressures, then the seal is a single phase vapor or single phase liquid seal for which the following equations still apply if one sets p_s equal to either the inside pressure for an all liquid seal or to the outside pressure for an all gas seal.

By requiring that the mass flows and radius at which saturation occurs in both Equations (18) end (19) be the same, then the radius term can be eliminated and the leakage rate found,

$$\dot{m} = \frac{2\pi r_m h^3}{r_o - r_i} \left[\frac{\rho_l(p_s - p_o)}{12\mu_l} + \frac{p_i^2 - p_s^2}{24\mu_g RT_f} \right] \tag{21}$$

Equations (18), (19), and (21) can also be solved for the radius at which the phase change occurs,

$$r_s = r_m - \frac{r_o - r_i}{2} \left[\frac{\dfrac{\rho_l(p_s - p_o)}{\mu_l} - \dfrac{p_i^2 - p_s^2}{2\mu_g RT_f}}{\dfrac{\rho_l(p_s - p_o)}{\mu_l} + \dfrac{p_i^2 - p_s^2}{2\mu_g RT_f}} \right] \tag{22}$$

Finally by noting that $p_s (r_s)$ is the general pressure as a function of radius, then the pressures from Equations (18) and (19) can be integrated to get the load being supported by fluid pressure,

$$W_f = 2\pi r_m(r_o - r_i) \left[\frac{\dfrac{\rho_l(p_o^2 - p_s^2)}{2\mu_l} + \dfrac{p_s^3 - p_i^3}{3\mu_g RT_f}}{\dfrac{\rho_l(p_o - p_s)}{\mu_l} + \dfrac{p_s^2 - p_i^2}{2\mu_g RT_f}} \right] \tag{23}$$

The friction power can be calculated using the above simplifying assumptions and the application of equation (11).

$$P_f = \frac{\pi(r_o^2 - r_i^2)r_m^2\omega^2}{h(r_o - r_i)} [\mu_l(r_o - r_s) + \mu_g(r_s - r_i)] \tag{24}$$

The method of solution for the equilibrium face temperature T_f is the same as for the general fluid model.

TWO PHASE PRESSURE LOAD SUPPORT

Before solving for equilibrium and stability conditions, it is helpful to understand the nature of the fluid pressure load support which develops under two phase operation. An example is used to illustrate fluid pressure load support. The balanced, hot water seal described by Lebeck [5] is used here. The data for this seal are:

r_o = 50 mm
r_i = 45 mm
p_i = 0.101 MPa
ω = 200 radians/s
σ = 0.167 μ
p_{sp} = 0.200 MPa
R_{H_2O} = 460 J/kg·K
f_c = 0.1
$K_{\Delta T}$ = 0.05 C°/W

Since this is a single component fluid, Equations (20) through (24) are used for the calculations. These equations are readily programed. To make the calculation over a wide range of temperatures and for some of the subsequent stability calculations, one needs to also have a computerized data base which gives p_s, ρ_l, μ_l, and μ_g as functions of temperature.

Fluid pressure load support from Equation (23) is shown in Figure 4 for a range of face temperatures T_f and outside pressures p_o. At a given pressure, as temperature increases, load support increases from the all liquid condition to a relatively high level as part of the seal interface is subject to the vapor
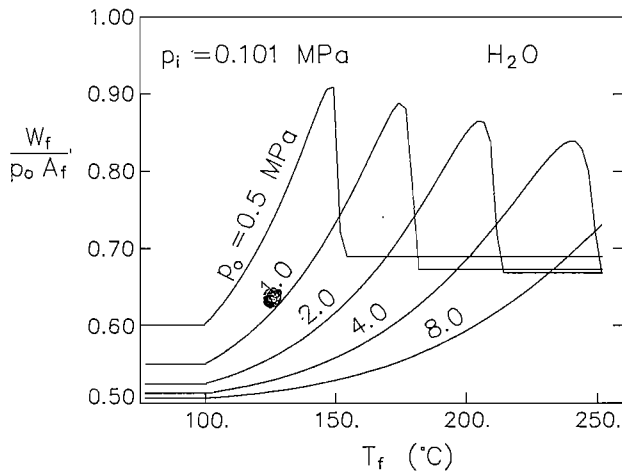
Figure 4. Fluid Pressure Load Support as a Function of Face Temperature for Water.

phase. The reason why load support increases is depicted in Figure 2. There is a distinct peak value of fluid pressure load support. Moving on to the right, load support collapses fairly quickly to a level corresponding to an all gas interface.

The fluid pressure load support shown in Figure 4 is similar in form to the non-dimensional pressure gradient parameter K, which is used in various references on seals [13]. In Figure 4, this factor is somewhat greater than 0.5 for most of the all liquid interface cases, because of the use of absolute inside and outside pressures, coupled with the nondimensionalization with respect to the outside pressure only. Nevertheless, the familiar K factor becomes considerably larger than 0.5 as the temperature increases, as is illustrated in Figure 4.

The ideal way for a seal to operate is at a face load which is somewhat above the fluid pressure load support value shown in Figure 4. This ensures that the seal gap is small, while at the same time minimizing contact forces and wear. It becomes clear from inspection of Figure 4 that by using conventional seal technology, it is difficult to have just the right balance ratio under all operating conditions when there is two phase operation. Some compromises must be made. Thus, it is important to see what the critical limits on balance ratio are.

EQUILIBRIUM SOLUTION

At first glance, it may appear that one could use Figure 4 to choose the balance ratio by simply making sure that the hydraulic closing force plus the spring load are large enough to overcome the maximum fluid pressure load support expected. The difficulty in doing this is that even if one does know the seal environment temperature, the graph is plotted in terms of the face temperature, which is somewhat higher. Thus, to find the needed balance ratio at a known environment temperature one must solve for the equilibrium value of T_f.

Equilibrium face temperature is found by first picking a face temperature and going through the entire set of calculations given by Equations (2) through (13) (or their equivalent for a single component given by Equations (23) and (24)). Then the predicted face temperature is calculated using Equation (14). This procedure is continued until the guessed value is equal to the calculated value.

The equations mentioned are readily programed to solve for the face temperature which satisfies Equation (14). A typical set of results for a given environment temperature is shown in Fig-

ure 5. For a given balance ratio, given a T_f shown by the diagonal line, the computed value is shown by the function. There may be as many as three solutions, A, B, and C on the graph. Solution A is for a case where most of the interface is liquid and represents a desirable operating condition. Solution B cannot occur in reality. It turns out that it is physically an unstable solution, so this point will not be discussed further. Finally, solution C is the case for an all vapor interface. The seal is operating essentially as a gas seal. The physical explanation is that friction heating at the interface causes the temperature rise to be so large that the entire interface is above saturation even though the surrounding fluid may not be.
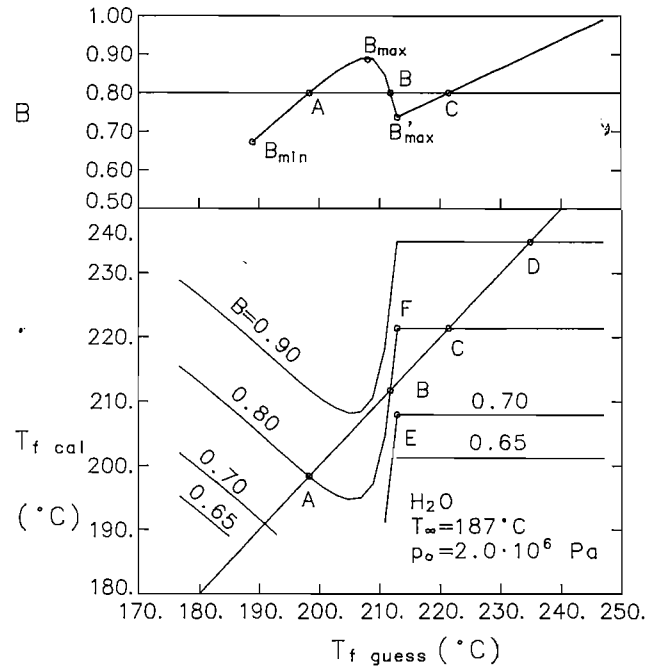

Figure 5. Equilibrium Solutions and Balance Ratio Function for Water.

Whether the seal will operate at A or C probably depends on previous history. Certainly, if by adjustment of pressure or temperature, one caused an all vapor operation to occur, then the seal may continue to operate at C. However, If one brings the pressure up first and then the temperature, then operation is expected at A, the more desirable condition.

At a balance ratio of 0.90 the only solution is at D, an all vapor case. At 0.65 balance ratio, there is no solution. The value of P_m must be negative, which means that the seal is lifted off. Thus, a 0.65 balance seal would blow off at this operating temperature. There is a solution for a 0.70 balance ratio seal having predominantly a liquid interface.

STABILITY LIMITS

Three limits of stability can be identified from Figure 5. The first limit is the value of balance ratio where operation is first possible. In this case the value is between 0.65, which has no solution, and 0.70. This will be termed B_{min}. This is the balance ratio below which the seal will always pop open at the given pressure and temperature. The second limit comes about as the balance ratio is made larger and larger, then the points A and B move toward each other. There is some maximum balance ratio which

causes the curve to become just tangent to the diagonal line. This is called B_{max}. This is the balance ratio above which the seal will always be forced to operate as an all vapor seal (intersection D). The third critical balance ratio is that which causes corner F to just touch the diagonal. If balance ratio is selected below this value, then operation will be forced to occur at intersection A. Thus, no vapor operation is possible so that even if one started the seal in an all vapor mode, it would return the to mixed mode at A. This is termed B'_{max}. The ideal seal operation is between B_{min} and B'_{max}.

Operation in this range assures that the seal interface has liquid and that it cannot operate in the all vapor regime.

To calculate these balance ratio limits for a range of operating temperatures, one can solve Equation (14) for B by substituting first for P_m from (13) and for W_m from (12) and for W^* from (8). This gives $B(T_f)$. This equation has been plotted in Figure 5. Now the points on the curve which correspond to B_{min}, B_{max}, and B'_{max} can be clearly seen. Intersections A, B, and C from the equilibrium solutions below can also be seen.

Once a program has been written to generate the balance ratio curve of Figure 5, then it is a simple matter to find the three balance ratio limits of interest for various seal operating conditions. Two sets of balance ratio limits obtained in this way are presented in Figure 6. Critical balance ratios are shown as functions of the seal environment temperature. For a given pressure, it can be clearly be seen how the minimum balance ratio increases with increasing temperature. The maximum balance ratio decreases with increasing pressure.

One of the questions which often arises with respect to the results of Figure 6 is at what point does most of the seal interface become all vapor. An examination of the details of the results shows that starting at the minimum balance ratio curve the interface is 99 percent + liquid and remains over 50 percent liquid almost all the way to the maximum balance ratio curve. This means that one will have a seal interface which has primarily liquid lubrication over most of the range between the two stability limits. This behavior affects the choice of optimum balance ratio which is discussed later after similar curves for a multicomponent fluid have been developed.

PROPANE-METHANE

Properties

To understand how the above method can be applied more generally, the common case of a multicomponent fluid is considered. The multicomponent fluid has a boiling range rather than just a boiling point. Propane-methane was chosen to serve as an example.

The solution to equation (2) requires that the viscosity and density be known as functions of pressure and temperature. For a mixture of vapor and liquid, each consisting of different fractions of the basic components, the development of a suitable algorithm to give the needed properties is not simple. The basis for and the assumptions used in the algorithm are given here.

• Equilibrium ratios at the given pressure and temperature are found for both methane and propane using double interpolation from equilibrium ratio data given in the "Engineering Data Book" of the Gas Processors Suppliers Association [14].

• Given the mole fraction of methane, the ratio of vapor to liquid is found using the previously mentioned equilibrium ratios.

• At the given pressure and temperature, densities of the gas mixture and the liquid mixture are computed using the equivalent molecular weight of the gas mixture and a volume addition rule for the liquid [14].
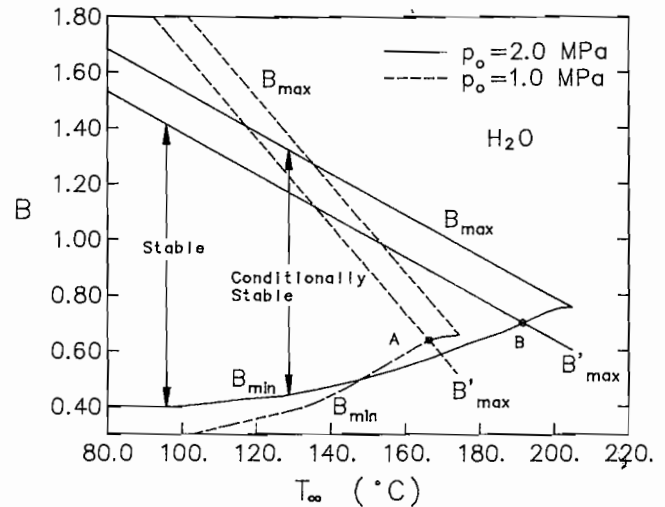

Figure 6. Critical Balance Ratios for Water at Two Pressures.

• Viscosity of the gas component is based on molecular weight. Viscosity of the liquid is taken as that of propane for the given temperature [14].

• The equivalent density is obtained from the densities of the liquid and vapor fractions. The equivalent viscosity is obtained by using a volume ratio averaging technique.

Based on the aforementioned, an algorithm was constructed to give the equivalent viscosity and density for a given pressure, temperature, and mole fraction of methane. While this approach was expedient for the current project, it is clear that to make the proposed critical balance ratio computation method useful for the wide range of multicomponent fluids of interest, it will be necessary to link the balance ratio computation program to one of the more general multicomponent vapor-liquid equilibria computer programs [15].

Fluid Pressure Load Support

Using the calculation methods described, the fluid pressure load support for the propane-methane binary system was calculated as a function of interface temperature and is shown in Figure 7. The case of pure propane is shown compared to a 0.1 mole fraction of methane mixed with propane. The first observation is that the maximum load support for the pure propane is less than that for water shown in Figure 4. Equation (23) shows that the peak pressure is influenced by several fluid properties which are functions of temperature. Thus, the peak load support depends on the specific fluid and the peak for propane is lower than that for water. The second observation is that the peak load support for the multicomponent fluid is even lower than the propane. The peak value, which is so well defined in Figure 4, is not nearly as sharp for the multicomponent case. The reason is that the more gradual transition between liquid and gas simply does not allow the sudden pressure transition and subsequent high load support as shown in Figure 2.

Equilibrium Solutions

Just as the case for water, one may solve for the computed interface temperature as a function of the assumed interface temperature. Results for a 0.1 mole fraction case are shown in Figure 8. The most important difference between Figure 8 and the case for water is that there is only one equilibrium solution possible for a given balance ratio. This is the case because the fluid pressure load support curve does not drop sharply as it does for water.
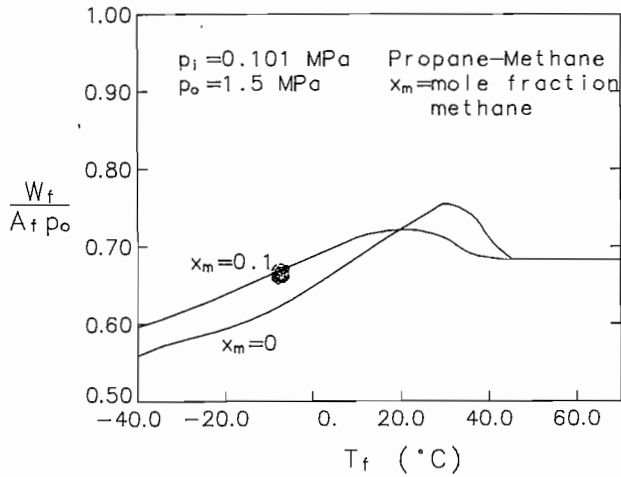
Figure 7. *Fluid Pressure Load Support as a Function of Face Temperature for Propane-Methane.*
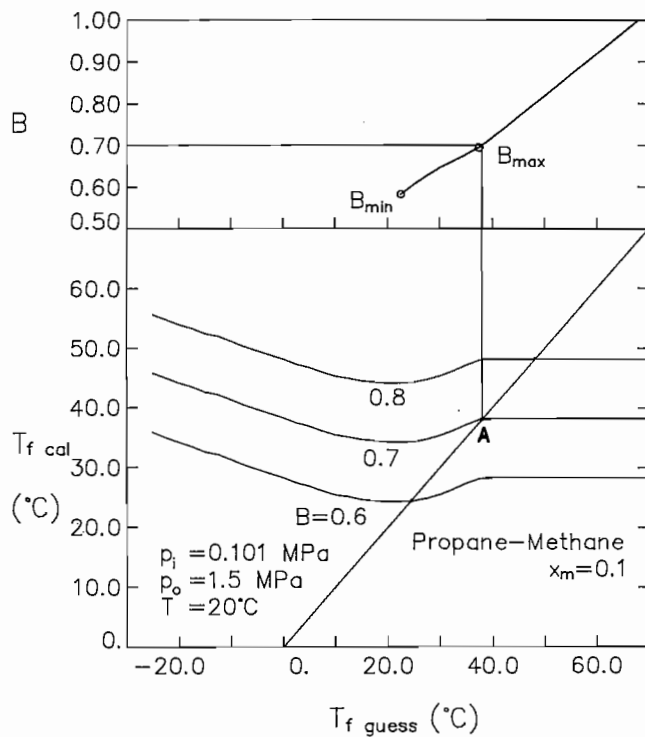
is quite different than the case for water. Here the minimum balance ratio is defined by the first existing point on the curve. The maximum balance ratio cannot be picked off of the curve directly, but it can be determined by noting, in the computation, the face temperature which causes all vapor operation. This becomes the saturation temperature at the sealed pressure of the propane component.

Critical Balance Ratios

Using the above procedures, critical balance ratios were calculated for the propane-methane binary and are shown in Figure 9. These curves are similar in character to the case for water, but, of course, the second maximum balance ratio curve does not exist. The lower curve represents the balance ratio below which the seal will blow open. The upper curve represents the balance ratio above which all vapor operation occurs and wear rate is expected to be high because of the total absence of liquid in the interface. Using the same procedure, the upper and lower critical balance ratios for pure propane are given in Figure 10. It is to be noted comparing the pure case to the mixed case that the pure case gives a slightly higher lower stability limit.

Figure 8. *Equilibrium Solutions and Balance Ratio Function for Propane Methane.*

The location of the critical balance ratios is somewhat different for this binary system. The minimum balance ratio (not shown in the lower part of Figure 8) is found just as the case for water. There is some balance ratio value below which a solution does not exist and this is the minimum critical balance ratio. The maximum critical balance ratio is simply defined as that balance ratio which causes operation to become all vapor. This is a unique value for a given operating temperature. From Figure 8, it can be seen to occur at point A, so that 0.7 is the maximum critical balance ratio for this condition.

The equilibrium balance ratio is shown in the upper part of Figure 8, as a function of interface temperature. This function
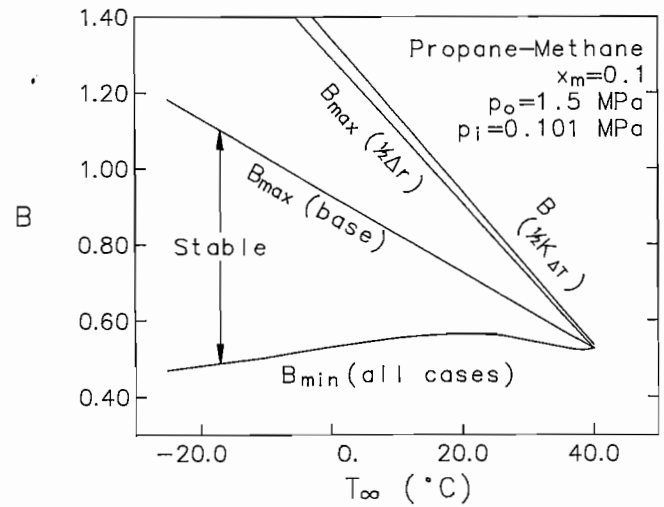

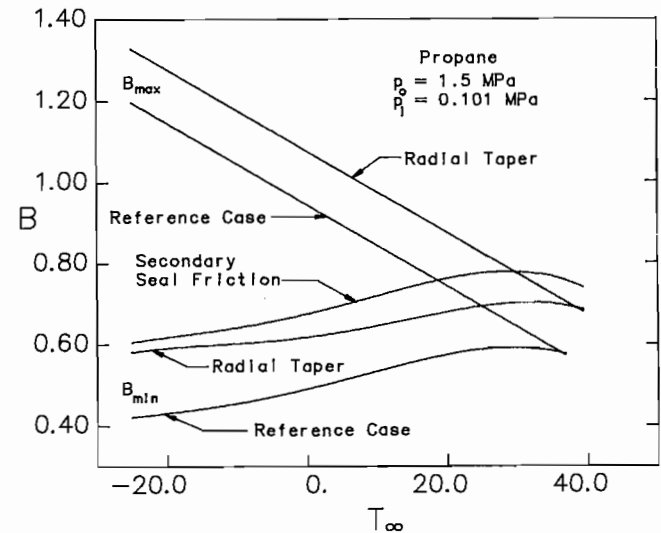Figure 9. *Critical Balance Ratios for Propane-Methane for Various Conditions.*

Figure 10. *Critical Balance Ratios for Propane for Various Conditions.*

EFFECTS OF SEAL DESIGN ON
CRITICAL BALANCE RATIOS

There are several other parameters in addition to the balance ratio over which the seal designer has control which strongly influence the critical balance ratio results. These include face width, temperature rise coefficient, secondary seal type, and radial taper.

Face Width

The critical balance ratio curves are shown in Figure 9 for a case where the face width is made one half of the value of 5.0 mm used for the base case studies herein. It can be seen that the maximum critical balance ratio increases compared to the base case. This is the expected result. This shows that one could operate a narrow faced seal with a given balance ratio at a temperature closer to sealed pressure saturation conditions than a wider faced seal without causing all vapor operation. This supports the conclusions of Will [3].

Temperature Rise Coefficient

A case where the temperature rise coefficient is taken at one half the value of the base case, as might be obtained by using materials having a higher thermal conductivity is also shown in Figure 9. Not surprisingly, the result is almost the same as that obtained by halving the face width. Neither modification has any effect on the lower critical balance ratio. This is because the upper critical balance ratio is strongly influenced by heat transfer while the lower critical balance ratio is primarily controlled by other factors.

Radial Taper

Many seal designs have some radial taper type of distortion which occurs either upon pressurization or due to thermal heating of the interface [16]. The consequence of this distortion is that at least some of the time (either hot or cold, under pressure or no pressure, or having worn or unworn faces) the faces may develop a convergent radial taper. Salant and Key [17] show how seal behavior is affected by such tapers and describe in detail the analysis of such situations. Using the equations already set forth herein with a variable radial film thickness, h, and using a converging radial taper of 100 microradians, the limits of stability were found for the propane case. The lower and upper stability curves shift upward, as shown in Figure 10. This result shows that one might be required to increase the balance ratio about 0.1 compared to the parallel face lower stability limit to assure that excessive leakage does not occur.

Secondary Seal Friction

The axial component of friction of a secondary o-ring seal can act to hang up a face seal. Thus, when operating near the lower stability limit, leakage may occur, even though the balance ratio is clearly above the theoretical lower stability curve. To estimate how large the effect might be, O-ring friction was estimated using references [18] and [19] at 388 N for the example seal at 1.5 MPa. Figure 10 shows how the lower stability limit is shifted upward from the reference case about 0.2. Since actual friction is indeterminate and may vary widely over its possible range, all that can be stated is that the curve shown probably represents a worst case. However, such friction may well be one of the causes of leakage of low balance ratio o-ring type seals in two phase applications. A seal design having a low friction secondary seal would have the potential to operate closer to the theoretical lower stability limit without excessive leakage, other factors being equal.

The previous results herein show how various parameters of seal design can have a strong influence on the limits of critical balance ratio. All of these factors must be carefully evaluated to assess the stability limits of a given seal design.

EMPIRICAL DATA

It is expected that certain seal problems such as short life and blowout are caused by the fact that seal balance ratio and other design features are such that actual operation falls outside the critical limits shown here at least some of the time. Detailed data is needed to test this idea. Buck [20] made one of the few studies which relates seal failures to operating conditions and design. However, the detailed data needed to test the validity of the methods developed herein are not available from the study. About the only information which could be obtained is based on Buck's [21] experience in sealing propane near saturation conditions with an O-ring type of secondary seal. He has observed that at a balance ratio of around 0.65 the seal usually comes open and leaks excessively, at 0.75 the seal will sometimes come open, and at 0.85 the seal usually works but eventually fails by wear out of the faces. This information appears to fit the predictions of Figure 10, if one considers that O-ring friction and converging radial taper probably do raise the lower stability limit at least some of the time. The fact that a 0.85 seal fails by wearing out is also consistent, because Figure 10 shows that all vapor operation would be expected if operation is anywhere near saturation conditions.

This limited information is hardly adequate to validate the model. What is needed is a detailed study comparing predictions to field experience. Seal design data, process conditions and fluid, and failure history data for specific applications are needed. Given this information, critical balance ratio can be predicted and expected performance compared to observed performance. It is thought that making such a study over a wide range of applications and for sufficient number of cases would yield statistical correlations from which useful balance ratio selection guidelines could be developed.

BALANCE RATIO SELECTION

The consequences of operation at either balance ratio extreme are clear. If balance ratio is too low, there is a risk of excessive leakage. If balance ratio is too high, there is a risk of rapid seal wear. In cases where the actual range available is fairly small, then it would seem that one should design for toward the upper limit. At least this will insure that the failure mode will be relatively rapid wear rather than seal blowout. In services which must operate near saturation conditions, it is likely that to avoid the possibility of blowout as might be triggered by O-ring friction or radial taper, then one will probably have to operate above the upper critical balance ratio in the all vapor regime and suffer the consequences, as suggested in Figure 10. A possible better alternative which might allow operation between the close stability limits might be to use a seal which has been carefully designed to maintain parallel faces and give low secondary seal friction, but experience and testing are needed to validate this idea.

For a seal with some room between the upper and lower stability limits. such that even with radial taper and friction one could avoid both limits, then one might consider designing for the middle. The reason for this choice is that friction and taper can push the upper curve down as much as such factors can push the lower curve up. Thus, the point of operation most removed from undesirable conditions is exactly the middle between the two limits. If one could eliminate by design some of the uncertainty caused by radial taper and friction, then it would be ideal to operate closer to the lower limit from a wear standpoint, al-

though it is a fact that the seal face will remain primarily liquid almost to the limit of the upper stability limit curve, and leakage might become a more important consideration than wear. Again, studies are needed to validate these recommendations.

CONCLUSIONS

- Two limits on balance ratio have been identified. Minimum critical balance ratio is that below which the seal will blow open. Maximum critical balance ratio is that above which all vapor operation and accelerated wear are expected to occur. Balance ratio should ideally be chosen between these two limits.

- Results show that the range between critical balance ratio values becomes small as saturation temperature at the sealed pressure is reached.

- Results show that other design variables such as face width, temperature rise coefficient, radial taper, and O-ring friction greatly influence the critical balance ratio curves. These factors can make it impossible to have reliable operation between the two stable limits above. The balance ratio may have to be chosen in the all vapor regime to avoid the possibility of seal blowout.

- A single component fluid displays considerably different critical balance ratio characteristics than a multicomponent fluid, at least for the cases studied here.

- A propane-methane binary fluid has a lower minimum critical balance ratio curve than for propane alone. This occurs in part because in the binary fluid the transition between all vapor and all liquid phases in the interface does not occur over a short radial distance as it does for a pure fluid and the fluid pressure load support does not become as high.

- Propane alone has a lower minimum critical balance ratio curve than for water alone. This occurs in part because propane has different ratios of densities and viscosities than water. The lower critical balance ratio curve is a function of the individual fluid properties.

- Field data is needed to test the usefulness of the model.

NOMENCLATURE

B	Balance ratio
B_{min}	Minimum critical balance ratio below which the seal opens
B_{max}	Maximum critical balance ratio above which the seal operates with all vapor
B'_{max}	Maximum critical balance ratio below which the seal cannot operate as an all vapor seal
f_c	Contact friction coefficient
h	Film thickness (m)
$\dot{m}, \dot{m}_l, \dot{m}_g$	Mass flow (kg/s)
p, p_o, p_i, p_{sp}, p_s	Pressures: outside, inside, spring, saturation (Pa)
P_f, P_m	Friction power: fluid and contact (W)
q_r	Volume rate of flow (m³/s)
r_o, r_i, r_g, r_b, r_m	Radii: outside, inside, saturation, balance, mean face (m)
R_{gas}	Gas constant (J/kg-K)
T, T_∞, T_f	Temperature: seal surroundings, face (C)
W_f, W^*, W_m	Load: fluid, applied, contact (N)
μ, μ_l, μ_g	Viscosity: liquid, gas (Pa·s0
ρ, ρ_l, ρ_g	Density: liquid, gas (kg/m³)
σ	RMS combined roughness (m)
ω	Angular velocity (1/s)

REFERENCES

1. Lymer, A., "An Engineering Approach to the Selection and Application of Mechanical Seals," 4th International Conference on Fluid Sealing, held in conjunction with the 1969 ASLE Annual Meeting, Philadelphia, Pennsylvania (1969).

2. Hughes, W. F., Winowich, N. S., Birchak, M. J., and Kennedy, W. C., "Phase Change in Liquid Face Seals," Journal of Lubrication Technology, 100, (1), pp. 74-79 (1978).

3. Will, T. P., "Effects of Seal Face Width on Mechanical Seal Performance — Hydrocarbon Tests," Lubrication Engineering, pp. 522-527 (September 1984).

4. Hughes, W. F., and Chao, N. H., "Phase Change in Liquid Face Seals II — Isothermal and Adiabatic Bounds with Real Fluids," Journal of Lubrication Technology, 102, (3), pp. 350-359 (1980).

5. Lebeck, A. O., "A Mixed Friction Hydrostatic Face Seal Model With Phase Change," Journal of Lubrication Technology, 102, (2), pp. 133-138 (1980).

6. Lebeck, A. O., and Chiou, B. C., "Two Phase Mechanical Face Seal Operation: Experimental and Theoretical Observations," Proceedings of the Eleventh Turbomachinery Symposium, Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 181-188 (1982).

7. Chiou, B. C., "The Effect of Two Phase Operation on Seal Performance," Masters Thesis, Mechanical Engineering Department, University of New Mexico (December 1980)

8. "Centrifugal Pumps for General Refinery Services, API Standard 610," 6th edition, Washington, District of Columbia: American Petroleum Institute (1981).

9. Lebeck, A. O., "Parallel Sliding Load Support in the Mixed Friction Regime, Part 1 — The Experimental Data," Journal of Tribology (January 1987).

10. Lebeck, A. O., "Parallel Sliding Load Support in the Mixed Friction Regime, Part 2 — Evaluation of the Mechanisms," Journal of Tribology (January 1987).

11. Young, L. A., and Lebeck, A. O., "Experimental Evaluation of a Mixed Friction Hydrostatic Mechanical Face Seal Model Considering Radial Taper, Thermal Taper, and Wear," Journal of Lubrication Technology, p.p 439-447 (October 1982).

12. Lebeck, A. O., "A Mixed Friction Hydrostatic Face Seal Model With Thermal Rotation and Wear," ASLE Transactions, 23, (4), pp. 375-387 (1980).

13. Peterson, M. B., and Winer, W. O., Eds, Wear Control Handbook, New York: American Society of Mechanical Engineers (1980).

14. Engineering Data Book, Volume II, Tenth Edition, Tulsa, Oklahoma: Gas Processors Association, (1987).

15. Prausnitz, J., Anderson, T., Grens, E., Eckert, C., Hsieh, R., and O'Connell, J., Computer Calculations for Multicomponent Vapor-Liquid and Liquid-Liquid Equilibria, New York: Prentice-Hall (1980).

16. Lebeck, A. O., "Contacting Mechanical Seal Design Using a Simplified Hydrostatic Model," Eleventh International Conference on Fluid Sealing, Cannes, France, BHRA, (1987).

17. Salant, R. F., and Key, W. E., "Improved Mechanical Seal Design Through Mathematical Modelling," First International Pump Symposium, Turbomachinery Laboratories,

Department of Mechanical Engineering, Texas A&M University, College Station, Texas, pp. 37-46 (1984).

18. Parker Seals O-Ring Handbook, Lexington, Kentucky: Parker Seal Company (1977).

19. Wensel, R., Cotnam, B., Gentili, H., and Constantinescu, I., "Friction and Axial Force/Displacement Characteristics of Elastomer Seals in Water," Lubrication Engineering, *41*, (9), pp. 550-555 (1985).

20. Buck, G. S., "A Methodology for Design and Application of Mechanical Seals," ASLE Transactions, *23*, (3), pp. 244-252 (1980).

21. Buck, G. S., Private Correspondence (1987).