AN IMPROVED HEAT SOAK CALCULATION FOR MECHANICAL SEALS

by

Gordon S. Buck

Chief Engineer, Field Operations

John Crane Inc.
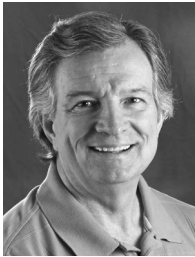
Baton Rouge, Louisiana

and

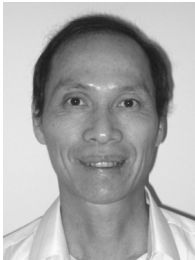**Tsu Yen Chen**

Senior Staff Engineer

John Crane Inc.

Morton Grove, Illinois

Gordon S. Buck is Chief Engineer, Field Operations, for John Crane Inc., in Baton Rouge, Louisiana. Prior to joining John Crane, he held various positions in the chemical processing, refining and pump industries. Mr. Buck has authored several publications on pumps and mechanical seals. As a member of every API 682 Task Force, he helped to write the standard for mechanical seals, and he is a member of the American Society of Mechanical Engineers.

Mr. Buck has a B.S. degree (Mechanical Engineering, 1970) from Mississippi State University and an M.S. degree (Mechanical Engineering, 1978) from Louisiana State University.

Tsu Yen Chen is Senior Staff Engineer for John Crane Inc., in Morton Grove, Illinois. Prior to joining John Crane, he was a senior FEA Specialist for Federal Mogul Corp. (Fel-Pro Inc.).

Dr. Chen has an M.S. degree (Mechanical Engineering, 1984) from the University of New Mexico and a Ph.D. degree (Mechanical Engineering, 1987) from the University of New Mexico.

ABSTRACT

Heat soak is defined as heat transfer from the metal of a seal chamber to the fluid within the seal chamber. Many variables affect heat soak including materials, surface configurations, and film coefficients. Heat soak can be from the seal chamber to the fluid within, or from the fluid to the seal chamber. Typically, the term heat soak is used to describe heat transfer from a hot pump to a cooler liquid within the seal chamber.

During the development of the API 682 seal standard (2004), the API 682 Task Force adopted an equation for estimating heat soak but recommended that the coefficients in the equation should be based on tests. However, there are still little data available and most users simply estimate heat soak using the API 682 default values. For a number of service conditions, the API 682 default equation produces unnecessarily high estimates for heat soak but default estimates are not always high. As a result, auxiliary systems can be incorrectly sized by using the API 682 default equation for heat soak. Improvements are needed in the API 682 heat soak equation.

The first step toward improving the API 682 heat soak equation was to use simple one-dimensional fin theory to derive a heat transfer equation. Surprisingly, this one-dimensional fin equation could be recast into a form that approximated the API equation but included additional parameters. The theoretical fin equation was checked and further improved using two-dimensional finite element analysis and again by using computational fluid dynamics. The predicted heat transfer compares favorably to measurements in the test lab. Based on these studies, the authors have prepared and offer a table of adjustment factors to be applied to the default API 682 heat soak equation.

INTRODUCTION

Figure 1 illustrates the concept of heat soak. In Figure 1, the pump is hot but the flush liquid is cool. As shown in Figure 1, heat is transferred by conduction from the hot pump case into the walls of the seal chamber. Cool liquid is circulated through the seal chamber and absorbs heat from the hot walls by convection. The pump shaft also is hot and contributes to heating the liquid. Flush liquid exits the seal chamber at a higher temperature than its inlet temperature. At the same time, the liquid is cooling the walls of the seal chamber as well as the pump shaft.
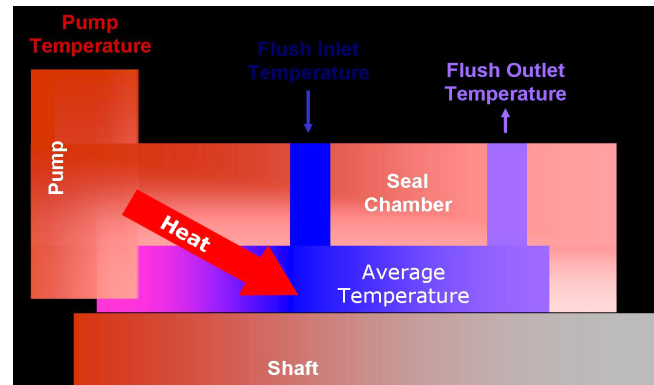

Figure 1. Concept of Heat Soak.

As environment controls and multiple seal systems became popular, there was a need for a systematic method for estimating the heat load imposed on a sealing system. About 1970, Hamaker (1977) consolidated and improved the various methods previously used to estimate heat soak. Hamaker based his method on Plan 23. Hamaker's method was published as a chart and taught during educational sessions of the American Society of Lubrication Engineers. As a result, Hamaker's method, especially his chart for estimating heat soak, was used by many seal manufacturers and application engineers. Years later, Jon Hamaker came to believe that his method produced low estimates for heat soak (Hamaker and Buck, personal correspondence).

In 1986, Adams (1987) published a paper on high temperature sealing that included a chart for determining heat soak. This chart produced heat soak values significantly higher than Hamaker's method. Adam's method appears to have been developed from considerations of a small pump handling hot water and fitted with dual seals and Plan 54.

In 1992, during the development of the API 682 seal standard, there was an open discussion of heat soak. The discussion group included both users and manufacturers of mechanical seals. The general feeling was that the Hamaker method produced low estimates for heat soak. The API 682 Task Force decided to default to a variation of the Adam method chart when pump specific data were not available for heat soak.

API Heat Soak Correlation

The API 682 heat soak equation is:

$$H_s = UA\Delta T \qquad (1)$$

In Equation (1),
H_s = Heat soak, Btu/hr
UA = A default of 12S where S is the seal size in inches
ΔT = Pump temperature-seal chamber temperature, °F

Equation (1) is intended to be an estimate and used only in the absence of data. As will be noted later, it is approximately representative of the actual heat soak, especially for API 682 seals in water at shaft speeds of 1800 rpm. For oils and for slower shaft speeds, Equation (1) overestimates the heat soak; however, for some applications Equation (1) underestimates heat soak.

Dodd and Hinkel Plan 23 Study

Dodd and Hinkel (1997) reported on their success using Plan 23 flush in a refinery. The Dodd and Hinkle report provided some 50 field experiences with data including pump temperature, flush temperature, presence of cooling water, etc. These data included water and hydrocarbon liquids. Unfortunately, Dodd and Hinkle did not provide enough data to correlate to the API heat soak equation; however, by making typical assumptions, the Dodd and Hinkel report indicates that the API default equation is reasonable for water but often high for hydrocarbons.

ONE-DIMENSIONAL FIN MODEL

Consider a one-dimensional fin with the base of the fin at the pump temperature. The equation for the heat soak based on a long fin is:

$$H_s = \sqrt{hpkA_s}\ \Delta T\ tanh(mL) \qquad (2)$$

The preferred simple form is:

$$H_s = U\ A_f\ \Delta T \qquad (3)$$

Since $tanh(mL) \cong 1$, the overall heat transfer coefficient, U, can be reduced to:

$$U = \sqrt{\left(\frac{hk}{L}\right)\left(\frac{t}{L}\right)} \qquad (4)$$

where:
H_s = Heat soak
h = Convective heat transfer coefficient
p = Perimeter of fin
k = Thermal conductivity
A_s = Cross-section area of fin
L = Length of the fin (depth of the stuffing box)
A_f = Convection area
t = Wall thickness

$$m = \sqrt{\frac{hp}{kA_s}}$$

Equation (4) takes into account the thermal conductivity of the pump, the thickness of the pump wall and the convection coefficient of the pump surfaces. Hence, the heat soak value calculated using Equation (3) is likely to be more accurate than the API equation. The long fin theory assumes one-dimensional heat conduction only; however, the heat transfer in a seal chamber is a three-dimensional heat conduction problem. Therefore, the results should be further improved by solving the three-dimensional heat transfer problem.

FINITE ELEMENT ANALYSIS MODEL

Figure 2 illustrates a simplified geometry for a seal chamber. The steady-state heat transfer equation in cylindrical coordinates is:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial}{\partial \theta}\left(k\frac{\partial T}{\partial \theta}\right) + \frac{\partial}{\partial Z}\left(k\frac{\partial T}{\partial Z}\right) + q''' = 0 \qquad (5)$$

where:
k = Thermal conductivity
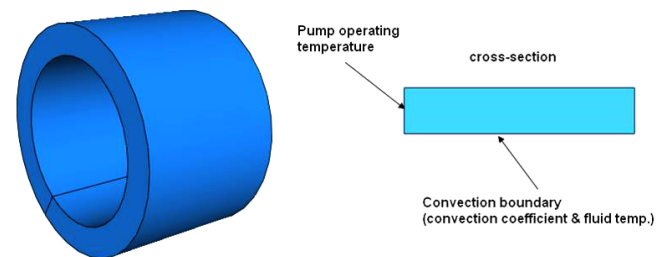T = Temperature
q''' = Heat source

Figure 2. Seal Chamber.

A commercially available finite element analysis program was employed to solve for the heat transfer problem of the pump seal chamber. Figure 3 compares the heat soak as calculated using the four methods mentioned above versus the thermal conductivity of the pump. Figure 3 shows that Hamaker's chart dramatically underestimates heat soak. The API default overestimates heat soak at a low thermal conductivity and underestimates the heat soak for high thermal conductivity. The long fin theory improves the heat soak estimate because it includes the dependence on thermal conductivity. However, the long fin theory predicts a higher heat soak value than does finite element analysis (FEA).
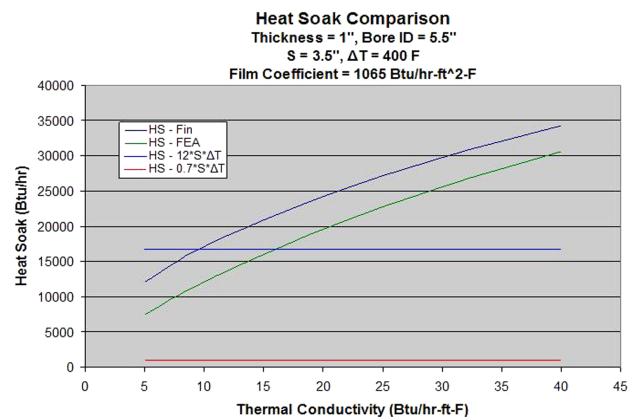

Figure 3. Heat Soak Versus Thermal Conductivity.

Figure 4 plots heat soak versus the thickness of the seal chamber. Again, Hamaker's chart dramatically underestimates the heat soak, and the default API equation overestimates the heat soak value for a thin wall seal chamber. As expected, the long fin theory predicts heat soak well for a thin wall seal chamber but begins to deviate from the FEA prediction as wall thickness increases.
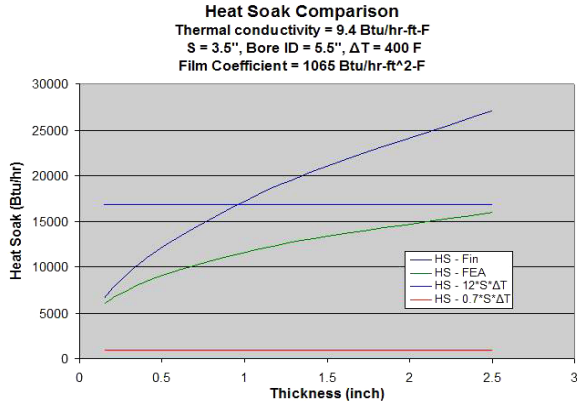
Figure 4. Heat Soak Versus Wall Thickness.

COMPUTATIONAL FLUID DYNAMICS MODEL

As expected, the finite element analysis did appear to improve the heat soak prediction for a seal chamber in comparison to both the API default and fin equations. However, in order to establish confidence in the FEA results, the heat soak prediction from the seal chamber only model (FEA) was compared to that of the whole chamber/seal assembly model using computational fluid dynamics (CFD). Two different chamber/seal assemblies were analyzed using CFD, and those results were compared with the FEA results. The cross-section of these two assemblies is shown in Figures 5 and 6. The areas that are marked "h" in Figures 5 and 6 are where the convection boundary conditions were applied based roughly on CFD results. The total heat soaks on surfaces 1, 2, 3 and surfaces 1 and 2 are calculated for the designs in Figures 5 and 6, respectively. For the design in Figure 5, the total heat soak for the assembly model is 5824 Btu/hr based on CFD, and for the seal chamber only model it is 5693 Btu/hr based on FEA. The difference is less than 2.5 percent. This is because the housing does not contact with the retainer or the sleeve, and it is barely in contact with the gland plate, therefore whether the whole assembly is modeled or not will not affect the heat soak calculation much. However, the design in Figure 6 is a different story. The total heat soak for surfaces 1 and 2 from the assembly model and gland plate only model is 1081 and 1323 Btu/Hr, respectively. The 22 percent difference is because the gland plate contacts both the retainer and housing, the heat is conducted from gland plate to retainer and housing, but the simplified model did not account for these heat conductions. Even though it showed that there is a 22 percent difference of the heat soak prediction for the design in Figure 6, however, by considering the tremendous analytical time that seal chamber only model saved, it is still acceptable and worthwhile to use the simplified model for heat soak calculation. Figures 7 and 8 show the temperature profile and the heat flux (heat soak) of the housing for the design in Figure 5.
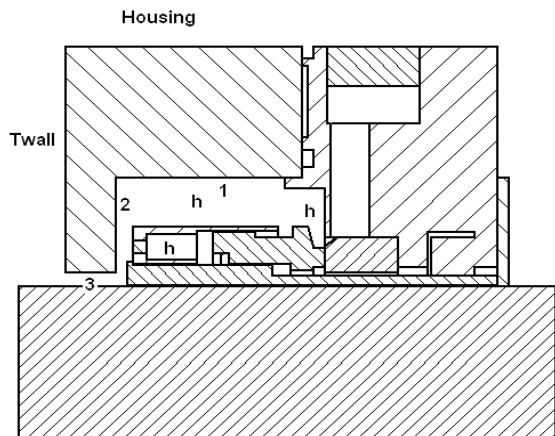

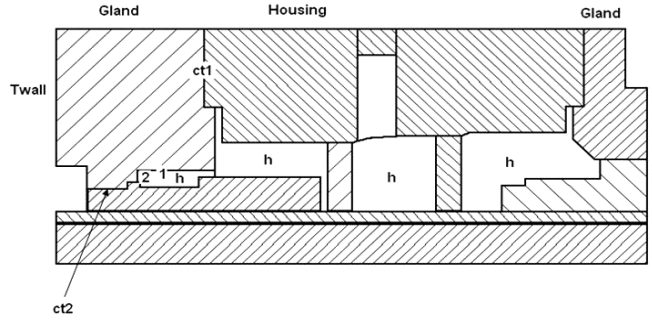Figure 5. Single Seal Model for CFD.

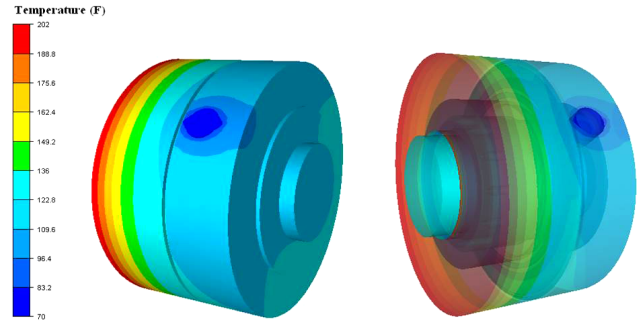Figure 6. Dual Seal Model for CFD.

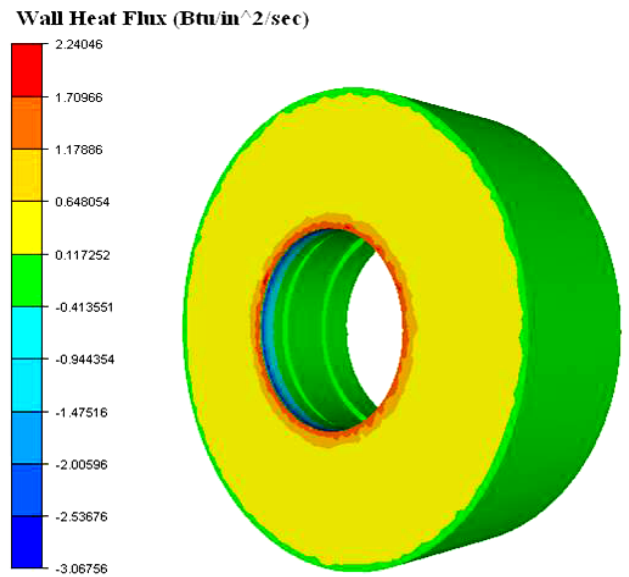Figure 7. Temperature Profile Example for Figure 5.

Figure 8. Heat Flux Example for Figure 5.

These comparisons provide with good guidance when using the FEA analysis on only the seal chamber for the heat soak calculation. Based on the comparison, we can conclude that the heat soak value from the FEA analysis on the seal chamber only is a reasonable and reliable model.

EXPERIMENTAL DATA

Although no experiments were done specifically for this project, there are many laboratory tests and field observations that include the effects of heat soak. Unfortunately, heat soak data remain difficult to obtain because necessary information is often missing. For example, the heat load on a reservoir or exchanger may be readily available but this includes seal face heat generation as well as heat soak. In the field, temperature information might be available but flow rates are often not available. Table 1 was constructed by examining such applications.

Table 1. Heat Soak Data.

Case Number	Temperature, F		Seal Mean Dia Inch	API	Actual	Heat Soak Btu/hr		Parameters not considered in API Equation			
	Pump	Seal				Factor	FEA Factor	Machine	Fluid	thk inch	Piping Plan
1	180	100	2.5	2400	3258	1.36	1.33	rig	water	2	11/52
2	180	88	2.5	2760	2359	0.85	1.30	rig	water	2	11/52
3	180	112	2.5	2040	2750	1.35	1.28	rig	water	2	11/53
4	180	121	4.3	3044	5006	1.64	1.30	rig	water	2	11/52
5	180	136	4.3	2270	4956	2.18	1.33	rig	water	2	11/53
6	180	158	4.3	1135	7637	6.73	1.35	rig	water	2	11/53
7	180	136	4.3	2270	3630	1.60	1.32	rig	water	2	11/52
8	500	148	2.2	9293	7900	0.85	0.70	pump	HT fluid	1.5	11/52
9a	495	248	3.9	11560	4698	0.41	0.90	rig	HT fluid	2.5	11/53
9b	499	119	3.9	17784	7210	0.41	0.70	rig	HT fluid	2.5	11/53

Notes
1. For all dual seals, the inner and outer theoretical seal face heat was subtracted from the reservoir heat load.
2. Cases 1 - 7 were similar designs and materials except for size.
3. By inspection, cases 1, 2, 3 would be expected to have the same heat soak. Average factor = 1.19
4. By inspection, cases 4, 5, 6, 7 would be expected to have the same heat soak. Discarding Case 6, average factor = 1.81
5. Cases 9a and 9b were the same seal but 9b included additional cooling. Seal face heat was assumed to be equal for 9a and 9b.

In Table 1, all seal cartridges were the face-to-back orientation, all used stainless steel metals, all operated at 3600 rpm. Only case 8 was from a pump; all other tests were done on a large test rig that tested two sets of seals simultaneously. The procedure used to develop Table 1 was to compute the heat transferred to the reservoir based on the buffer/barrier fluid flow. The heat generation from (both) the seal faces was calculated conventionally and subtracted from the heat load on the reservoir. The remainder is heat soak. Although Table 1 shows much scatter, the general trend is somewhat representative of pump to seal heat soak.

In contrast to the data in Table 1, test results using a smaller rig were also reviewed but not included because the results were not considered representative of heat soak from a pump. A tentative conclusion is that the mass of the pump should be considerably more than that of the seal in order for the API heat soak procedure to be representative.

RECOMMENDED IMPROVEMENTS
TO THE API EQUATION

Without going through the detailed computations of CFD, FEA or even the fin analogy, sufficiently accurate heat soak estimates can be obtained by making slight modifications to the API 682 heat soak equation as follows:

$$H_s = m_1 m_2 m_3 m_4 m_5 m_6 UA \Delta T \tag{6}$$

In Equation 6,
H_s = Heat soak, Btu/hr
UA   = The API default of 12S where S is the seal size in inches
ΔT = Pump temperature-seal chamber temperature, °F
m_1 = Speed factor, (rpm/1800)0.26, for example
        = 1.0 for 1800 rpm (base case)
        = 1.2 for 3600 rpm
        = 0.90 for 1200 rpm
m_2 = Thermal conductivity factor
        = 1.0 for stainless steel (base case)
        = 2.3 for carbon steel and cast iron
        = 1.4 for 12 percent chrome steel
m_3 = Thickness factor
        = 1 for 1 inch thick seal chamber wall (base case)
        = 0.81 for 0.5 inch thick seal chamber wall
        = 1.13 for 1.5 inch thick seal chamber wall
        = 1.24 for 2.0 inch thick seal chamber wall
m_4 = Bore factor, (seal chamber bore/API bore)
        = 1.0 minimum
        = 1.0 for API 610 standard bore and seal sizes
m_5 = Viscosity factor, $(.4/u)^{0.15}$, for example
        = 1.0 for 0.4 cP (base case)
        = 1.11 for 0.2 cP
        = 0.87 for 1 cP
        = 0.68 for 5 cP
        = 0.62 for 10 cP
m_6 = Combined fluid properties factor excluding viscosity
        = 1 for water (base case)
        = 0.78 for synthetic oil barrier fluids

        = 0.72 for conventional lube oils
        = 0.65 for nonvaporizing hydrocarbon mixtures
        = 0.53 for vaporizing hydrocarbon mixtures

Instead of using the fin analogy, all base case reference values, coefficients and exponents recommended for use in Equation (6) were obtained by curve fitting the FEA simulations to selected representative cases.

The speed factor, m_1, recognizes that convective heat transfer is a function of shaft speed. Since convection heat transfer increases with shaft speed, heat soak also increases with shaft speed. Because the best base case reference point was found to be 1800 rpm, pumps operating at 3600 rpm would have a higher heat soak if all other parameters were equal.

The thermal conductivity factor, m_2, recognizes that heat transfer increases with the thermal conductivity of the metal in the seal chamber wall. The base case reference point is stainless steel as noted in Adams (1987). Since the thermal conductivity of carbon steel is more than that of stainless steel, heat soak will be more for carbon steel. However, when a stainless steel dual seal cartridge is bolted to a carbon steel pump, the appropriate walls are those of the stainless steel seal chamber.

The thickness factor, m_3, recognizes that the wall thickness influences heat transfer. For thin walls, the fin analogy equation becomes increasingly accurate. In practice, the wall thickness of the pump seal chamber is not constant and representative thickness data may not even be readily available. For dual seal cartridges, the appropriate thickness is the wall thickness of the gland plate. Likewise, for some pumps, the thickness of the gland plate is somewhat representative of the pump wall to which it is bolted.

The seal chamber bore factor, m_4, will not normally be a consideration. This factor is included only to account for those rare applications in which the radial clearance between the seal parts and seal chamber bore is very large. The point of reference is the standard bore and seal relationship in API 610 (2004). As a very rough guideline, consider that the diametral clearance between shaft outer diameter (OD) and seal chamber bore ranges between 2 to 3 inches for many API pumps. The minimum diametral clearance between seal chamber bore and seal OD is ¼ inch. It turns out that, for many API pumps, the seal chamber bore is in the range of ½ inch to 1½ inch more than the representative seal diameter. The heat soak might be proportionally larger if the bore is significantly larger than the expected API size.

As can be seen from the typical values, the viscosity factor, m_5, can have a very significant effect on the computed heat soak. Viscosity has a strong effect on convection heat transfer. The viscosity factor will be particularly important when computing heat soak from viscous process fluids and buffer/barrier fluids. In general, heat soak to fluids more viscous than water will be less than the default API estimate for heat soak.

Except for viscosity, all other fluid properties can be combined into the fluid property factor, m_6. This factor includes density, fluid thermal conductivity and specific heat. Sometimes these physical properties are not readily available and little loss in accuracy results from combining typical values into a single factor.

EXAMPLE AND CALCULATIONS

A 3.5 inch mean diameter dual seal per API 682 is to be used in a 500°F pump at 3600 rpm. The bulk temperature of the synthetic oil barrier fluid in the reservoir is to be 150°F. The viscosity of the oil is 5 cP at 150°F. The seal chamber is stainless steel and the walls are 1.5 inches thick. Evaluate the heat soak for this application.

For purposes of comparison, the estimate for heat soak is first obtained from Equation (1) using the default coefficients. If the bulk temperature of the barrier fluid is 150°F and the pump temperature is 500°F, then the default API value for heat soak is:

AN IMPROVED HEAT SOAK CALCULATION FOR MECHANICAL SEALS

H_s = 12 (3.5) (500 – 150)
H_s = 14700 Btu/hr

Next, consider the various multiplier factors and obtain an improved estimate for heat soak.

For 3600 rpm, m_1 = 1.2
For stainless steel, m_2 = 1.0
For 1.5 inch wall thickness, m_3 = 1.13
For the API dimensional envelope, m_4 = 1.0
For viscosity of 5 cP, m_5 = 0.68
For synthetic barrier fluids, m_6 = 0.78

The product of all these factors is 0.719; therefore an improved heat soak estimate is:

H_s = (0.719)(14700)
H_s = 10569 Btu/hr

In comparison, the heat soak computed using the FEA method is 11143 Btu/hr. Taking the FEA results as the basis, the API default method for estimating heat soak is high for this example by about 24 percent. The heat soak computed using the fin analogy is about the same as the API default value for this example. The adjusted heat soak using the recommended multiplier factors is about 5 percent less than the FEA result. As noted in the discussion of CFD analysis, the actual heat soak might be even less depending on the configuration of the dual seal.

CONCLUSION

Heat soak prediction can be simple for some pump/seal systems but complex for others. The authors were unable to develop a simple, universal equation that is accurate for a wide range of services but have recommended modifying factors for the default API equation to produce improved estimates for heat soak. In particular, heat soak to typical buffer/barrier oils is likely to be less than estimated by the default API equation.

REFERENCES

Adams, W. V, January 1987, "Better High Temperature Sealing," *Hydrocarbon Processing Magazine*, pp. 53-59.

API Standard 610, 2004, "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industry," Tenth Edition, American Petroleum Institute, Washington, D.C.

API Standard 682, 2004, "Pumps—Shaft Sealing Systems for Centrifugal and Rotary Pumps," Third Edition, American Petroleum Institute, Washington, D.C.

Dodd, V. R. and Hinkel, T. E., 1997, "Mechanical Seal Reliability Improved with API Piping Plan 23," *Proceedings of the Fourteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 1-6.

Hamaker, J., May 1977, "Mechanical Seal Lubrication Systems," ASLE Seal Education Course, Montreal, Quebec, Canada.

Hamaker, J. and Buck, G. S., Personal Correspondence.

BIBLIOGRAPHY

Buck, G. S., 2008, "Guidelines for Application of High Temperature Dual Seals: A Tutorial," *Proceedings of the Twenty-Fourth International Pump Users* Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 69-74.

"Circulation Systems for Single and Multiple Seal Arrangements," MMTC-602, John Crane Inc., Morton Grove, Illinois.

"Sizing and Selecting Sealing Systems," 1980, Durametallic Corporation, SD-1162A.

ACKNOWLEDGEMENT