ANALYSIS OF THE THERMAL BEHAVIOR IN THE SEALING CAVITY OF AN OVERHUNG API PROCESS PUMP

by

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Dr. Nelik is an author of several publications in the area of centrifugal pumps and hydraulic power recovery turbines, heat transfer, and CAD/CAM/CAE applications for engineering/manufacturing integration.

ABSTRACT

In recent years, cooling requirements for mechanical seals have become a subject of controversial opinions. While the maximum limiting temperatures of mechanical seals are known and specified by seal manufacturers, the actual temperatures in the sealing cavity of the pump stuffing box during pump operation are not readily available. A conservative approach has been to provide extra cooling to the stuffing box. However, the availability of clean cooling water is often scarce in many installations, and therefore, knowledge of the thermal behavior of the sealing cavity region becomes important.

A developed analytical method, which utilizes the finite element analysis (FEA) technique to determine the thermal behavior of the pump stuffing box in the sealing region is presented. A single-stage overhung hot water (590°F) boiler circulation pump was analyzed. The stuffing box of the pump described herein the paper was equipped with API plan 23.

The developed method is not limited to a specific API stuffing box piping plan, but can be used for a variety of cooling/flushing stuffing box configurations, such as API Plan 2, or others.

The values of the heat transfer coefficients used in the method were found iteratively, based on published correlations for natural convection (Nusselt number vs Grashof number).

INTRODUCTION

A cross section of a typical high-pressure, high-temperature single-stage overhung pump used for the boiler circulation water is shown in Figure 1. The operating conditions, related to the nature of the problem, are:

Pumped liquid:	boiler feed water
Max. suction temperature	590°F
Max. suction pressure	1450 psia
Pump speed	1450 rpm
Pump flow	1840 gpm
Pump head	156 ft
Seal Plan 23- pumping ring with cooler	
Pumping ring flow	0.67 gpm
Heat generated at the seal faces	1708 BTU/hr (per seal supplier)
Cooling water temperature	104°F
Pump material	carbon steel

If the temperatures in the sealing cavity were found to be excessively high, additional stuffing box cooling might be required, such as in API plan C. Therefore, the main objective

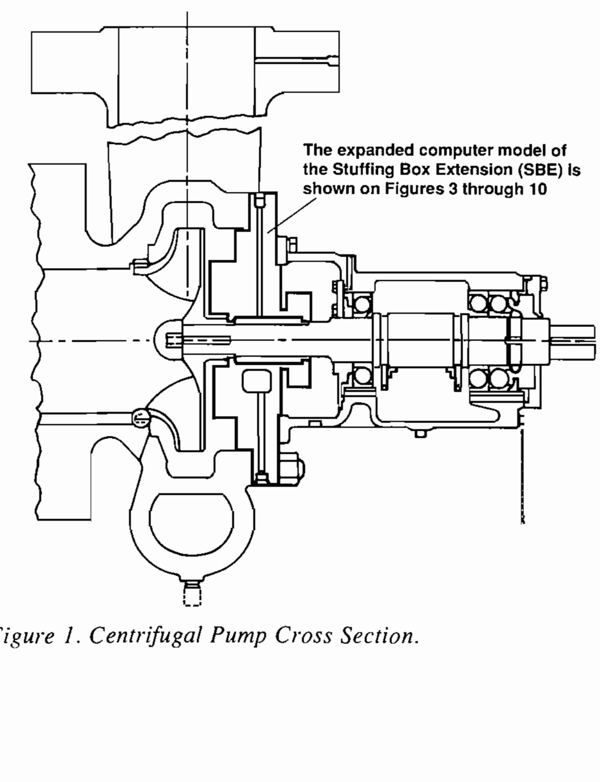

Figure 1. Centrifugal Pump Cross Section.

was to determine the water temperature in the sealing cavity to ensure that the temperature does not exceed the maximum allowable, as specified by the seal manufacturer.

As a result of the study, the water temperature in the sealing cavity was found to be below the maximum allowable, and no additional cooling of the stuffing box was required. Follow up testing of the pump verified the pump temperatures to be within allowable limits.

As the heat from the pumped liquid propagates through the stuffing box, it reaches the seal cavity, and the temperature of the liquid in the seal cavity tends to increase. A portion of this heat is rejected to the ambient air. A pumping ring, mounted on the shaft, pumps the hot liquid from the cavity to the cooler, and cooled liquid is returned to the sealing cavity. This is shown in Figure 2. The heat balance, therefore, establishes a certain liquid temperature in the sealing cavity. It was found in the analysis that this temperature was equal to 196°F, which is less than the maximum allowable temperature of 320°F. The distribution of metal temperature, deflections, and stresses in the stuffing box also were determined. Details of the method are described in the following sections.

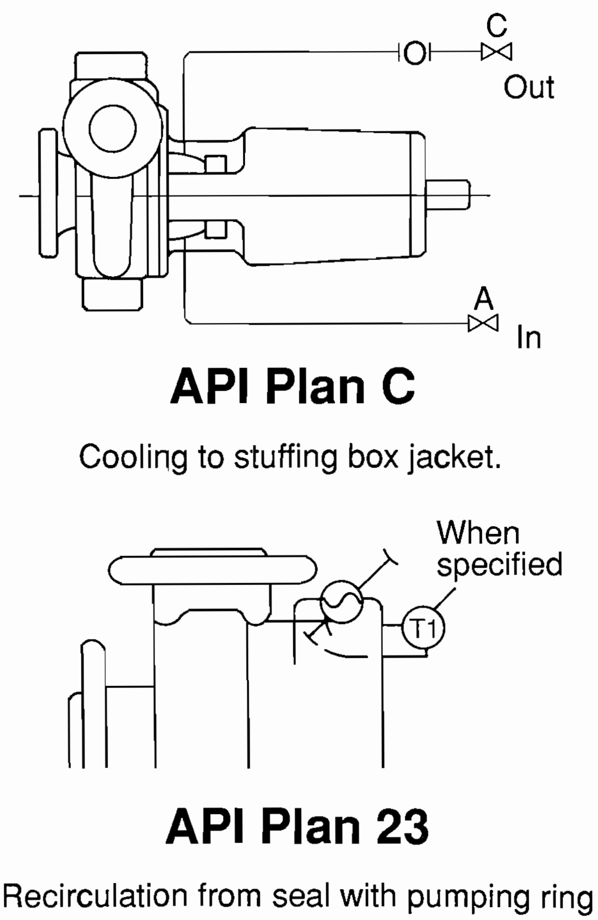

Figure 2. API Seal Circulation and Cooling Plans.

METHOD OF APPROACH AND ASSUMPTIONS

A computer model (meshed) of the stuffing box extension (SBE) used in the finite element analysis (ANSYS) program is shown in Figure 3. Since the studied geometry represents a body of revolution with circumferential uniform boundary conditions, an axisymmetric model approach was chosen. Such an approach requires considerably fewer finite elements as compared to a full 3-D model.

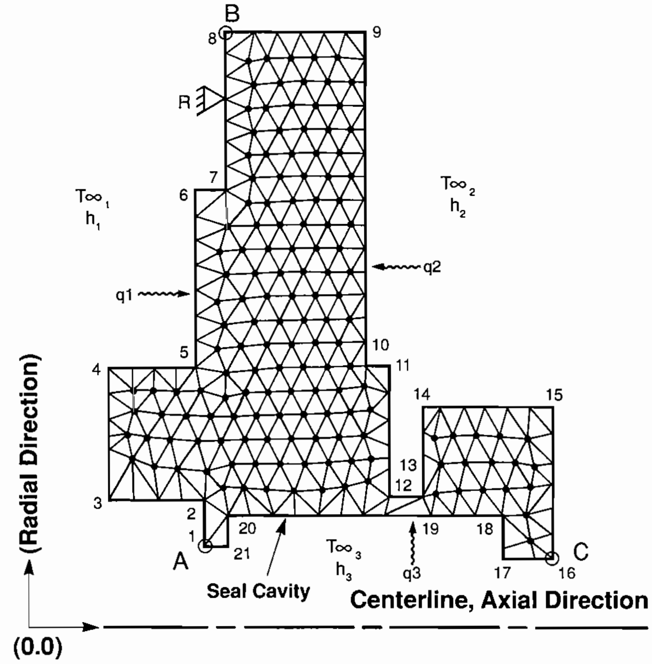

Figure 3. Stuffing Box Meshed Model Is the Seal Cavity Bulk Liquid Temperature.

Points 1 through 21 represent the auxiliary points used to construct the computer model and to mesh it. The x-y coordinates (where x represents radial, and y axial coordinates) are listed in Table 1. The water temperature on the impeller side was assumed conservatively equal to the maximum suction temperature of $T_{\infty 1}$ = 590°F. This side is shown on Figure 3 as extended from A to B (Face 1). The side from B to C (Face 2) is exposed to ambient air with bulk temperature $T_{\infty 1}$ = 68°F, and the side from C to A (Face 3) is in contact with water in the sealing cavity. But the temperature, $T_{\infty 3}$, is not known and needs to be determined. Therefore, the thermal boundary conditions are specified as the bulk temperatures $T_{\infty 1}$, $T_{\infty 2}$, $T_{\infty 3}$, and the average heat transfer coefficients h1, h2, h3, on the corresponding sides of the stuffing box. The geometrical boundary conditions are zero axial deflections at the mesh nodes in the region of the bolting attachement of the SBE to the pump casing, as shown in Figure 3. Note that the the body of the SBE is not restricted to the x direction at the bolting region and, therefore, is allowed to grow radially. The heat transfer coefficients are found from the standard published relationships and data for Nusselt number vs Grashof number, Nu=f(Gr) [1]. The Grashof number for the free convection at the vertical wall (air side) is given as:

$$\mathrm{Gr} = \frac{g\beta(T_{w2} - T_{\infty 2})}{\nu^2}$$

and the gravity force is causing the convective currents.

For the impeller side, the gravity force is replaced by the centrifugal force acting on the liquid neighboring the stuffing box, and is given as:

$$\mathrm{Gr}_{IMP} = \frac{r\omega^2\beta(T_{w1} - T_{\infty 1})}{\nu^2}$$

The Nusselt number is given in [1] as:

$$Nu_L = 0.024 \left[\frac{Pr^{1.17}}{1.0 + 0.494\, Pr^{2/3}}\, Gr \right]^{2/5}$$

The heat transfer in the sealing cavity is a convection in the horizontal cavity and can be described by the formulae from [2]:

$$Gr_{SEAL} = \frac{a}{g} \left(\frac{g\beta\rho^2}{\mu^2} \right) b^3 (T_{w3} - T_{\infty 3})$$

$$Nu_{SEAL} = 0.0426\, Gr_{SEAL}^{0.37}$$

Since the sealing cavity bulk temperature is not known a priori, an iterative process is required. The essence of the logic of the iterations is demonstrated below.

Table 1. Model Geometry Coordinates.

Point No.	X, in (Radial Coordinate)	Y, in (Axial Coordinate)
1	1.75	4.00
2	2.75	4.00
3	2.75	2.00
4	5.50	2.00
5	5.50	3.80
6	9.20	3.80
7	9.20	4.50
8	12.50	4.50
9	12.50	7.50
10	5.50	7.50
11	5.50	8.00
12	2.75	8.00
13	2.75	8.70
14	4.65	8.70
15	4.65	11.50
16	1.50	11.50
17	1.50	10.40
18	2.40	10.40
19	2.40	8.70
20	2.40	4.50
21	1.75	4.50

LOGIC OF ITERATION

- Assume heat transfer coefficients on all three faces h_1, h_2, and h_3. Also assume that water bulk temperatures in the seal cavity $T_{\infty 1,2,3}$ are known.
- Based on the $T_{\infty 1,2,3}$ and $h_{1,2,3}$, run the computer program and calculate the temperature and heat flux distribution (using FEA analysis) in the body of the SBE.
- Calculate the average wall temperatures on each face of the stuffing box:

$$T_{wi} = \frac{\sum_j T_{wj}}{N_i}$$

where T_{wj} are local temperatures at a given element face, and N_i is the number of mesh nodes on that face.

- Recalculate the heat transfer coefficients $h_{1,2,3}$ based on the corrected average wall temperatures $T_{w1,2,3}$ and rerun the FEA calculations until the problem converges. This constitutes the "inner" convergence on the heat transfer coefficients for a given guessed seal cavity bulk liquid temperature.
- Compute the total heat flux rejected to the seal cavity by the SBE:

$$Q_{REJ} = \frac{\sum Q_{is}}{N_i}$$

Add to this the heat generated on the seal faces (Q_{FACE} = 1708 BTU/HR, known) to get a total:

$$Q_{REJ,\,tot} = Q_{REJ} + Q_{FACE}$$

- Calculate the heat removed by circulation of the water from the seal cavity to the cooler at the thermal gradient of:

$$\Delta T_{COOL} = T_{\infty 3} - T_{COOL}\text{, where}$$
$$T_{COOL} = 104°F \text{ (given)}$$
$$Q_{COOL} = C\rho\, m_{SEAL}\, \Delta T_{COOL}$$

- Compare the total rejected heat, $Q_{REJ,\,TOT}$, to that removed from the seal cavity, and adjust the guessed water temperature in the seal cavity, $T_{\infty 3}$.

Go back to the second step in this process and continue until the following convergence is satisfied:

$$Q_{REJ,\,TOT} = Q_{COOL}$$
(within desired accuracy)

The final seal water temperature is thus found.

TEMPERATURE DISTRIBUTION

The temperature distribution in the body of the SBE is shown in Figure 4. The final converged solution is presented. The SBE deflections would result from this temperature distribution.

DEFLECTIONS

The original (stet line) and distorted (solid line) shapes of the body of the SBE are shown in Figure 5. The distorted shape is consistent with the geometrical boundary conditions (zero axial deflections) at the bolt location on Face 1 as shown on Figure 3.

Further details are shown in Figures 6 and 7 of the radial and axial deflections of the SBE. It can be seen that the maximum radial growth of the SBE is approximately 0.046 in at the outside diameter. The axial deflection reaches a maximum of 0.011 in at the air side and 0.012 in on the impeller side. In the seal cavity region, the radial deflection reaches the maximum of 9.0 in. However, in the immediate proximity of the seal faces, the radial deflection is only 0.002 in. This value is within the acceptable limits, specified by the seal manufacturers.

A detailed finite element analysis of the seal gland distortions has been done in the past by some seal manufactures [3]. In general, if the deflections at the seal faces would exceed the maximum allowable values, the seal manufacturer can be contacted for the possibility of the alternate design of the seal gland and stuffing box. If the alternate design is not feasible, the liquid temperature would have to be decreased to lower deflections at the seal faces.

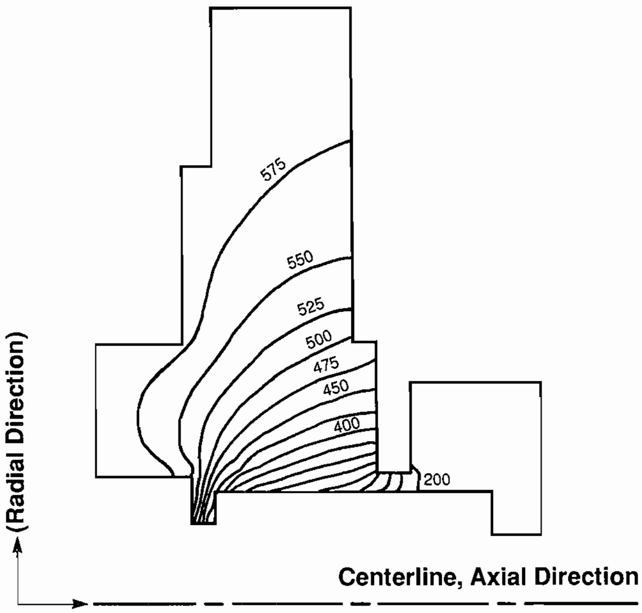

Figure 4. Temperature Distribution in the SBE.

API 610, 7th Edition, carefully addresses the importance to maintain parallelism of the seal faces. For example, the register-fit surface is specified to be concentric to the shaft and must

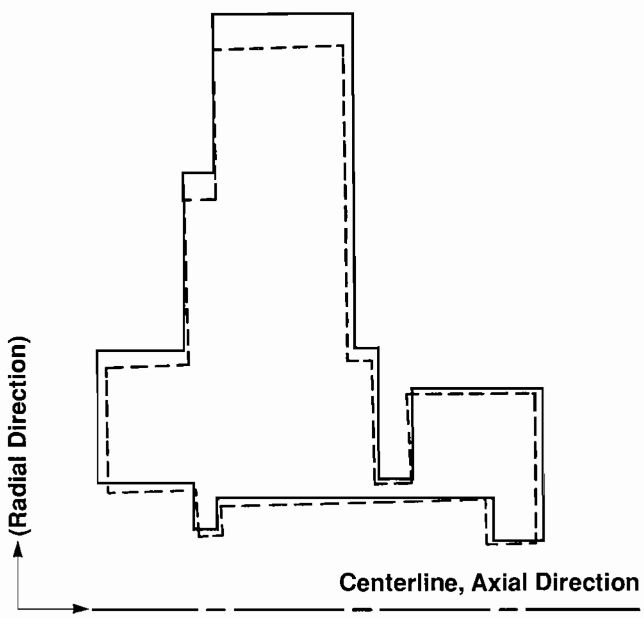

Figure 5. SBE Deformed Shape.

have a total indicated runout of not more than 0.005 in [4]. Such practices ensure the optimum seal life. The described method can be applied to enhance the study of the seal life as a function of the seal deflections. In certain cases, a joint research in this subject area between the pump manufacturers and seal vendor may be beneficial.

STRESSES

The radial, axial, and Von Mises stresses in the stuffing box (Figures 8, 9, and 10), are due to the temperature distribution in

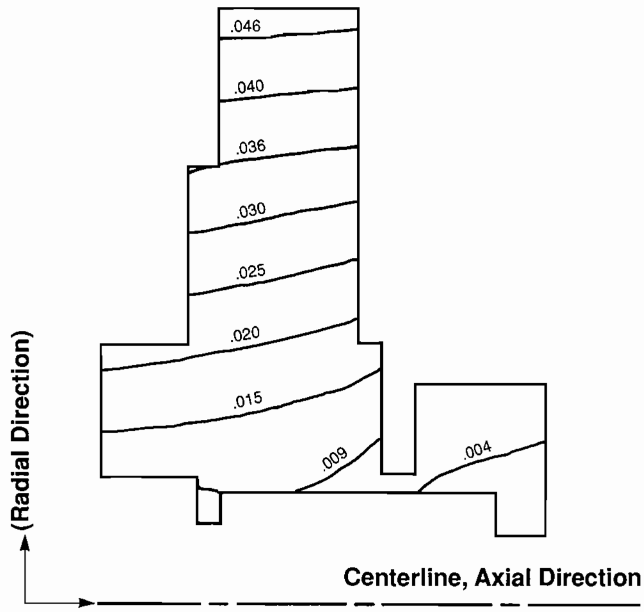

Figure 6. Radial Deflection, UX (inches).

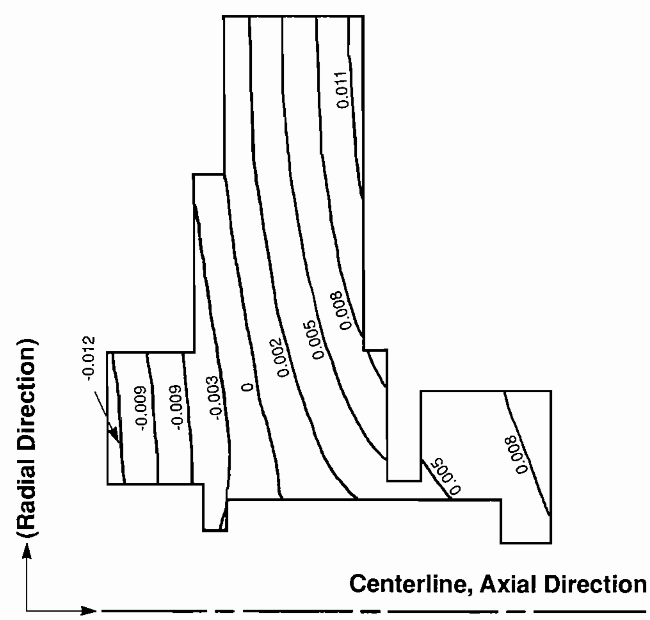

Figure 7. Axial Deflection, UY (inches).

the body of the SBE and subject to the boundary conditions of zero axial deflections and bolt locations.

It can be seen that the maximum Von Mises stress is 42477 psi. This is a localized phenomenon and does not exceed allowable limits as defined by the rules of the ASME Pressure Vessels' Codes.

TEST INFORMATION

The devised analytical method described in this paper allowed prediction of the seal cavity liquid bulk temperature, which was computed to be within the allowable values. No additional stuffing box cooling was determined necessary. It was of interest to correlate the predicted values with actual test data. High-

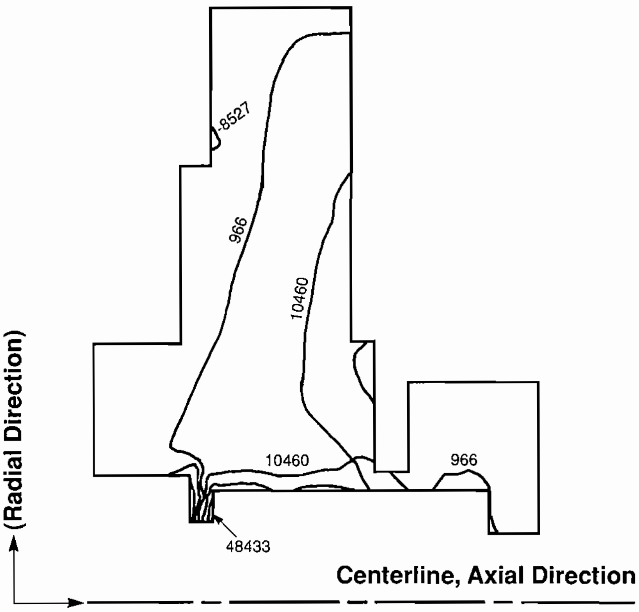

Figure 8. Radial Stress, SX (psi).

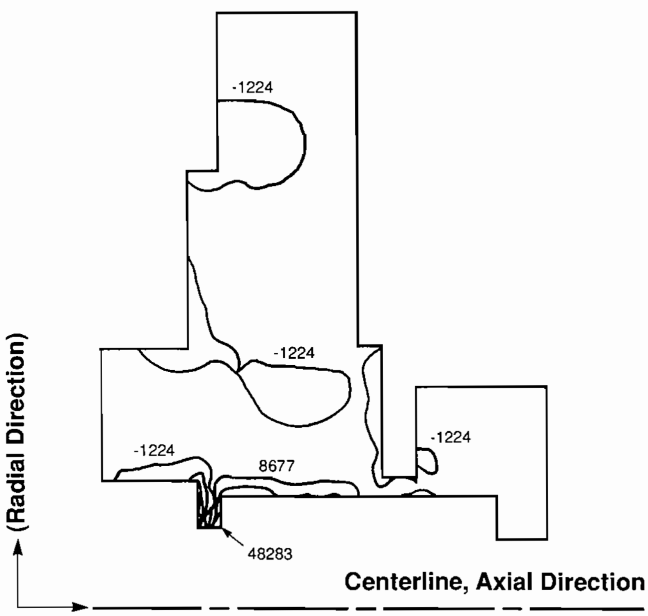

Figure 9. Axial Stress, SY (psi).

temperature, high-pressure field testing was performed, with the seal cavity temperature monitored. The pump was equipped with API plan 23, in the same way as described in the analysis. A set of thermocouples was used to record seal cavity temperature (as shown in Figure 3). The measured temperature in the seal cavity was found to be 203°F, which agrees with the computed results (196°F) within 3.5 percent.

CONCLUSIONS

A developed computational technique allows prediction of the bulk liquid temperatures in the sealing cavity of a centrifugal pump. In addition, temperature, deflection and stress distribution were determined. A finite element analysis is used as a backbone of the method and is enhanced by auxiliary computer

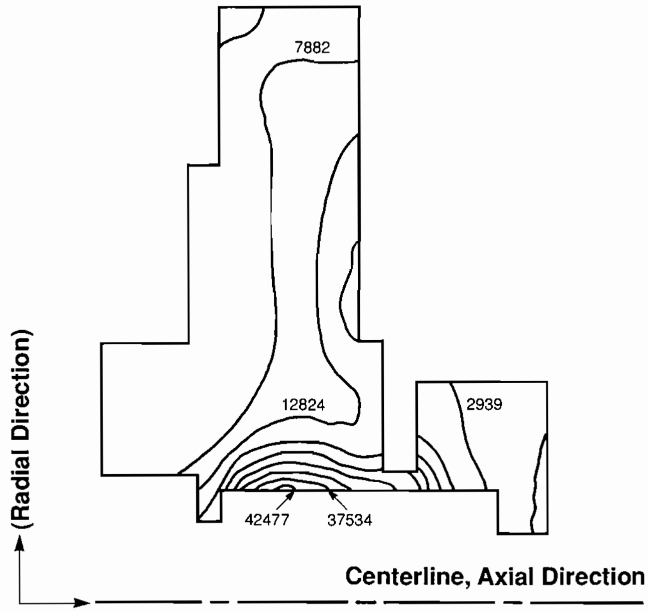

Figure 10. Von Mises Stress, SIGE (psi).

programs to provide iterative convergence. As a result of the application of this method, a seal cavity water temperature was computed and found to be less than the maximum allowable values. Therefore, no cooling of the stuffing box was deemed to be required. The analytical method was verified by test results, showing close agreement.

Modifications and enhancements of the described method are currently used at the author's company for routine applications to heat transfer phenomena encountered in centrifugal pumps.

In addition, the ability to predict deflections in the area of the seal stuffing box may be used as a tool to optimize seal life. Such study is suggested to be conducted in the future between the pump manufacturers and seal suppliers.

NOMENCLATURE

a	centrifugal acceleration, ft/sec**2
b	characteristic gap in seal cavity, inches (OD-ID)/2
Cp	liquid specific heat, BTU/lbm*hr
g	gravitational constant, ft/sec**2
Gr	Grashof number
h_1	heat transfer coefficient, impeller side, BTU/hr*in**2*F
h_2	--------------------, air side, ----------------------
h_3	--------------------, seal cavity,--------------------
$\dot{m}_{seal}$	seal circulation mass flow, lbm/HR
N_i	number of elements on a given face
N_u	Nusselt number
P_r	Prandtl number
r	characteristic dimension (radius), inches
T_{w1}	average wall temperature, impeller side, °F
T_{w2}	--------------------, air side, ----------------------
T_{w3}	--------------------, seal cavity, ------------------
$T_{\infty 1}$	liquid bulk temperature, impeller side, °F
$T_{\infty 2}$	--------------------, air side, ----------------------
$T_{\infty 3}$	--------------------, seal cavity, ------------------
T_{cool}	cooler liquid temperature, deg. F (104°F given)
Q_{rej}	heat flux rejected to liquid, BTU/HR
Q_{is}	heat flux on a given element i of a given face BTU/hr
Q_{face}	heat flux generated on the seal face, BTU/hr

Greek Letters

β	bulk modulus, l/F
ω	rotation velocity of liquid, l/sec
υ	kinematic viscosity, ft/sec**2
μ	dynamic viscosity, lbm/ft*sec

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