Leak Rate Analysis of the Westinghouse Reactor Coolant Pump

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Prepared by T. Boardman, N. Jeanmougin, R. Lofaro, J. Prevost

Energy Technology Engineering Center

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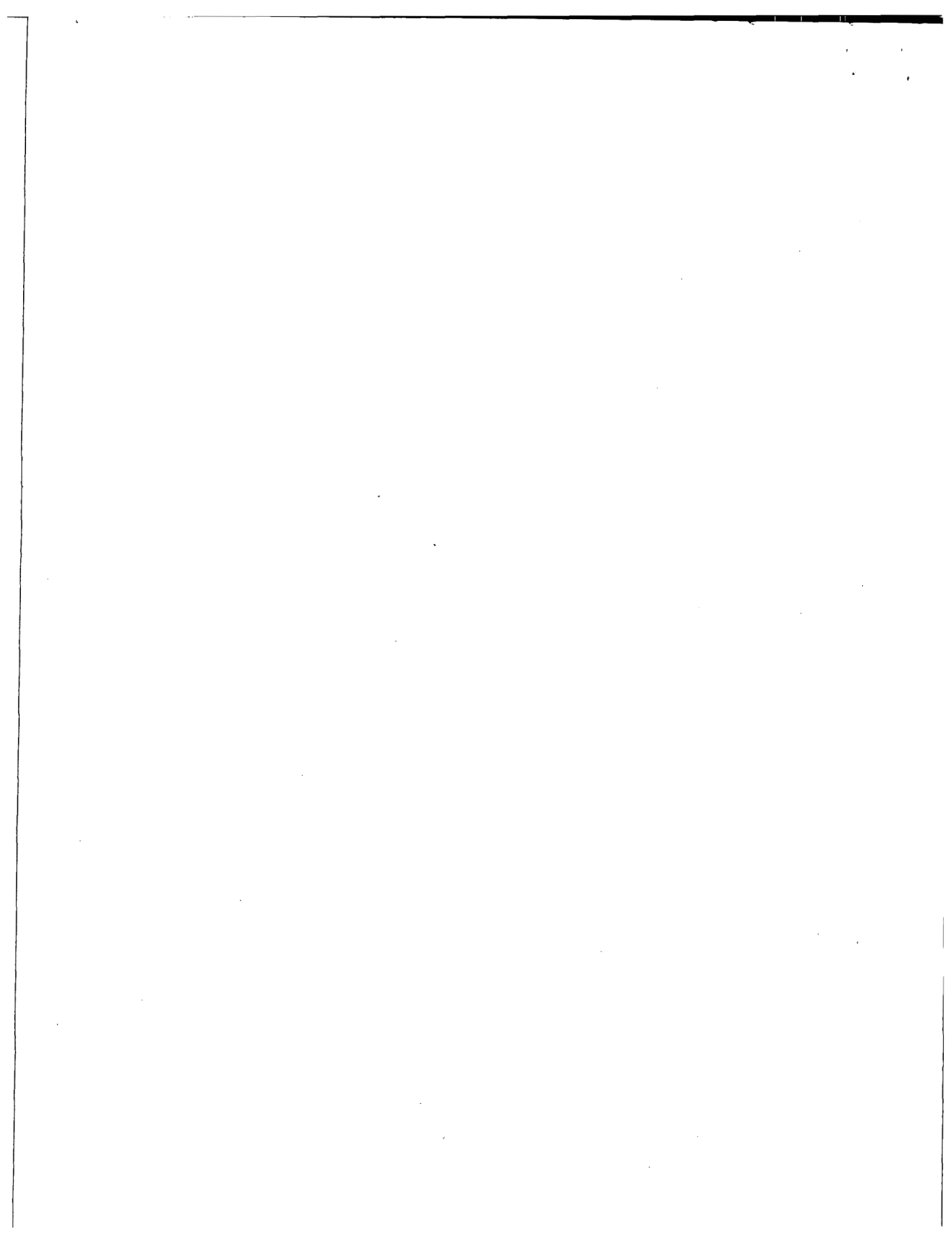
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ABSTRACT

An independent analysis was performed by ETEC to determine what the seal leakage rates would be for the Westinghouse Reactor Coolant Pump (RCP) during a postulated station blackout resulting from loss of ac electric power. The object of the study was to determine leakage rates for the following conditions:

Case 1: All three seals function.

Case 2: No. 1 seal fails open while Nos. 2 and 3 seals function.

Case 3: All three seals fail open.

The ETEC analysis confirmed Westinghouse calculations on RCP seal performance for the conditions investigated. The leak rates predicted by ETEC were slightly lower than those predicted by Westinghouse for each of the three cases as summarized below.

Case 1: ETEC predicted 19.6 gpm, Westinghouse predicted 21.1 gpm.

Case 2: ETEC predicted 64.7 gpm, Westinghouse predicted 75.6 gpm.

Case 3: ETEC predicted 422 gpm, Westinghouse predicted 480 gpm.

TABLE OF CONTENTS

FIGURES

TABLES

EXECUTIVE SUMMARY

This work was performed by the Energy Technology Engineering Center (ETEC) in response to Nuclear Regulatory Commission (NRC) concerns regarding leakage rates of the Westinghouse Reactor Coolant Pump (RCP) during station blackout. The object of the study is to independently determine leakage rates for the postulated functional state of the assembly without consideration to predicting failure or identifying the failure mechanism. During the loss of ac electric power the pump would be stationary and all cooling functions would be lost. It was assumed that pump rotation stops immediately after a station blackout occurs.

Figure 1 is a schematic of the RCP seal assembly. The RCP is a vertical shaft pump with three seals. The no. 1 seal is a film-riding, controlled leakage seal. The no. 2 and 3 seals are rubbing face type.

The scope of the study includes leakage rate predictions for the following cases.

1. All three seals function.

2. No. 1 seal fails open while no. 2 and 3 seals function.

3. All three seals fail open.

The analysis also addresses the following concerns:

1) effect of system cooldown and depressurization on leakage rates

2) sensitivity of leakage to dimensional and thermal hydraulic parameters.

Force balance equations were developed for the no. 1 seal faceplates to predict seal gap, pressure drop and flow. Finite element structural models were developed for all three seals to predict thermal and pressure distortions. Two-phase flow correlations based on the Dukler constant slip flow model were used to determine seal leakage. The flow model included the following assumptions:

1) thermodynamic equilibrium between phases

2) adiabatic process

3) uniaxial steady state flow.

Results of the independent ETEC analysis confirm Westinghouse calculations on seal performance for the conditions investigated.

1. An initial leakage rate of 19.6 gpm was calculated for the case with all three seals functioning.

2. An initial leakage rate of 64.7 gpm was predicted for the case with no. 1 seal failure and no. 2 and 3 seals functional.

FIGURE 1: RCP SEAL ASSEMBLY
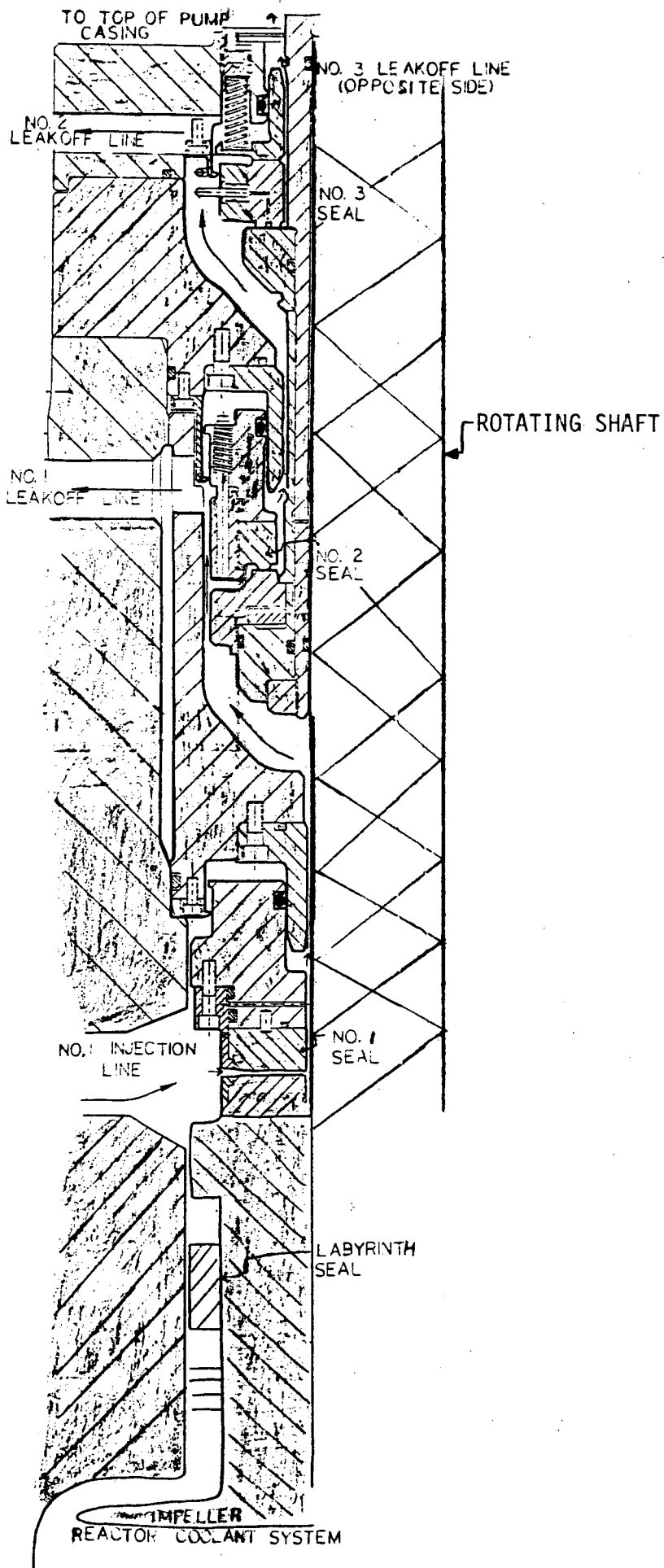
3. An initial leakage rate of 422 gpm was calculated for the case with all three seals failed.

4. The no. 2 seal will rotate closed if the no. 1 seal fails.

5. Leakage rates decrease due to reactor depressurization (2250 psia to 1250 psia)

    a. 19.6 gpm decreases to 16.9 gpm

    b. 64.7 gpm decreases to 33.7 gpm.

The following recommendations are made regarding future investigation.

1. Determine Experimentally the Probability of Seal Failure

    Analysis indicates that seal failure is an important factor in determining the time required to uncover the core. The probability of seal failure cannot be accurately determined by analysis, therefore the probability of seal failure should be determined experimentally.

2. Evaluate the Effect of Boil-Off Downstream of No. 2 Seal

    ETEC and Westinghouse analyses predict that the temperature gradient across the No. 2 seal will cause it to rotate closed during a station blackout. However, if all the water downstream of the no. 2 seal flashes to steam and "boils-off," the heat sink provided by the water will be lost and the temperature gradient across the no. 2 seal will decrease. The predicted leakage rate is close to the boil-off rate, therefore an analysis of the no. 2 seal, assuming boil-off, would provide valuable information on seal stability and closure.

3. Determine Experimentally the Labyrinth Seal Flow Resistance

    The dominant resistance for the case when all three seals fails is the labyrinth seal located between the impeller and the first stage seal. To identify the worst case leakage rate when all three seals fail open, the resistance factor for the labyrinth should be determined experimentally.

4. Evaluate Time to Uncover Core for Specific Plants

    An important result of the leakage rate analysis is the time required to uncover the core. This time should be determined for specific plants based on their individual system thermal hydraulic response characteristics and hardware designs.

5. Evaluate No. 2 Seal Leakage with Failure of No. 1 Seal

    Analysis indicates that the no. 2 seal will rotate closed during a station blackout due to thermal gradients across the seal. This behavior limits the overall leak rate, especially if the no. 1 seal

3

fails open. As presented in section 3.2.1, the calculated leak rate past the no. 2 seal when it rotates closed is on the order of .1 gpm. However, analysis can not verify whether the no. 2 seal can withstand full system pressure and high temperature gradients for an extended time period. Therefore, testing of the no. 2 seal subjected to these conditions--preferrably during a test of a full scale seal assembly-- is recommended. If testing of a RCP seal assembly is not conducted, then tests of the O-rings and channel seals, under pressure and temperature conditions which would exist if the no. 1 seal failed, should be considered.

1.0 INTRODUCTION

The Energy Technology Engineering Center (ETEC) conducted an independent analysis of the leakage rates from the primary coolant system through the Westinghouse Reactor Coolant Pump (RCP) shaft seals and associated downstream piping for a postulated station blackout condition. A station blackout would cause loss of ac power to the plant, loss of RCP rotation, and loss of cooling and injection water to the RCP seal system subjecting the seals to off-design conditions. There is concern that leakage through the RCP seal system would cause the core to be uncovered during a lengthy station blackout. The ETEC study was funded by the Nuclear Regulatory Commission (NRC) as a part of the research on this issue. Proprietary documents detailing the RCP dimensions and construction were provided by Westinghouse but are not included as part of this report.

The scope of the study encompassed prediction of leakage rates during a proposed station blackout for the following cases.

1. All three seals function.

2. The no. 1 seal fails open while the nos. 2 and 3 seals function.

3. All three seals fail open.

Leakage was calculated for nominal reactor coolant system conditions (2250 psia and 550°F) during normal plant operation. The effect of reactor coolant system (RCS) depressurization and cooldown on leakage was also investigated. In addition, the sensitivity of the calculated leakage rates to changes in various parameters was evaluated.

2.0 MODELS

The computer models developed for calculating the leakage rates were, 1) a seal model, 2) a flow model, and 3) a structural model. The seal model determines the gap and pressure distribution between the faceplates of the no. 1 film riding seal for various flow conditions. The flow model calculates the pressure drop for single and two-phase flow through the leak paths such as the labyrinth seal and the leakoff lines. The structural model determines the rotation of the seal faceplates due to thermal and pressure gradients across the seals. Results of the three models were integrated to obtain the final leakage rates.

2.1 SEAL MODEL

The Westinghouse RCP has three seals. The no. 1 seal is a film riding, controlled leakage seal whereas the no. 2 and 3 seals are rubbing face-type seals. The leakage across the no.1 seal cools the seal assembly. The high pressure, subcooled leakage is supplied by an injection system upstream of the no. 1 seal. Part of the injection water flows through the seal assembly. The remainder flows into the reactor coolant system as make-up water. Backup cooling is provided by a water-to-water heat exchanger parallel to the labyrinth seal. During a station blackout, both injection and cooling water would be lost. High temperature reactor coolant water would then flow into the seal system. This condition will affect the angle between the faceplates of the RCP seals and the gap between the faceplates of the no. 1 film riding seal. The seal model calculates this gap and the pressure distribution across the no. 1 seal. The structural model, described in Section 2.3, is used to calculate the thermal and pressure distortions.

The gap between the no. 1 seal faceplates is determined by a force balance. The inputs to the seal model include flow rate, angle between the faceplates of the seal ring and runner, enthalpy, inlet pressure, and estimate of the gap. The model calculates the pressure distribution along the seal face. This fluid pressure profile between the seal faceplates determines the opening force on the seal. The closing force on the seal is proportional to the differential pressure across the seal and acts on the upper surface of the seal ring. A schematic of these opposing forces is shown in Figure 2.

The seal model compares the calculated opening and closing forces. If these forces are unbalanced, a new gap is estimated and the procedure is repeated until the forces are balanced. The included angle between the seal faceplates is determined by the structural model based on the thermal and pressure gradients across the seal. The seal model incorporates the two-phase flow correlations used for the flow model described in Section 2.2. A more detailed description of this model is given in Appendix A.

2.2 FLOW MODEL

The reactor coolant water during normal plant operation is subcooled and is nominally 2250 psia and 550°F. If a station blackout occurs this water will leak into the RCP seal assembly. As it flows through the RCP seal system, the pressure drop due to frictional losses causes the fluid to enter the two-phase region. Two-phase flow correlations were used to calculate pressure gradients

FIGURE 2

#1 FILM-RIDING SEAL FORCE BALANCE

UPSTREAM PRESSURE

DOWNSTREAM
PRESSURE

SHAFT

FLUID PRESSURE
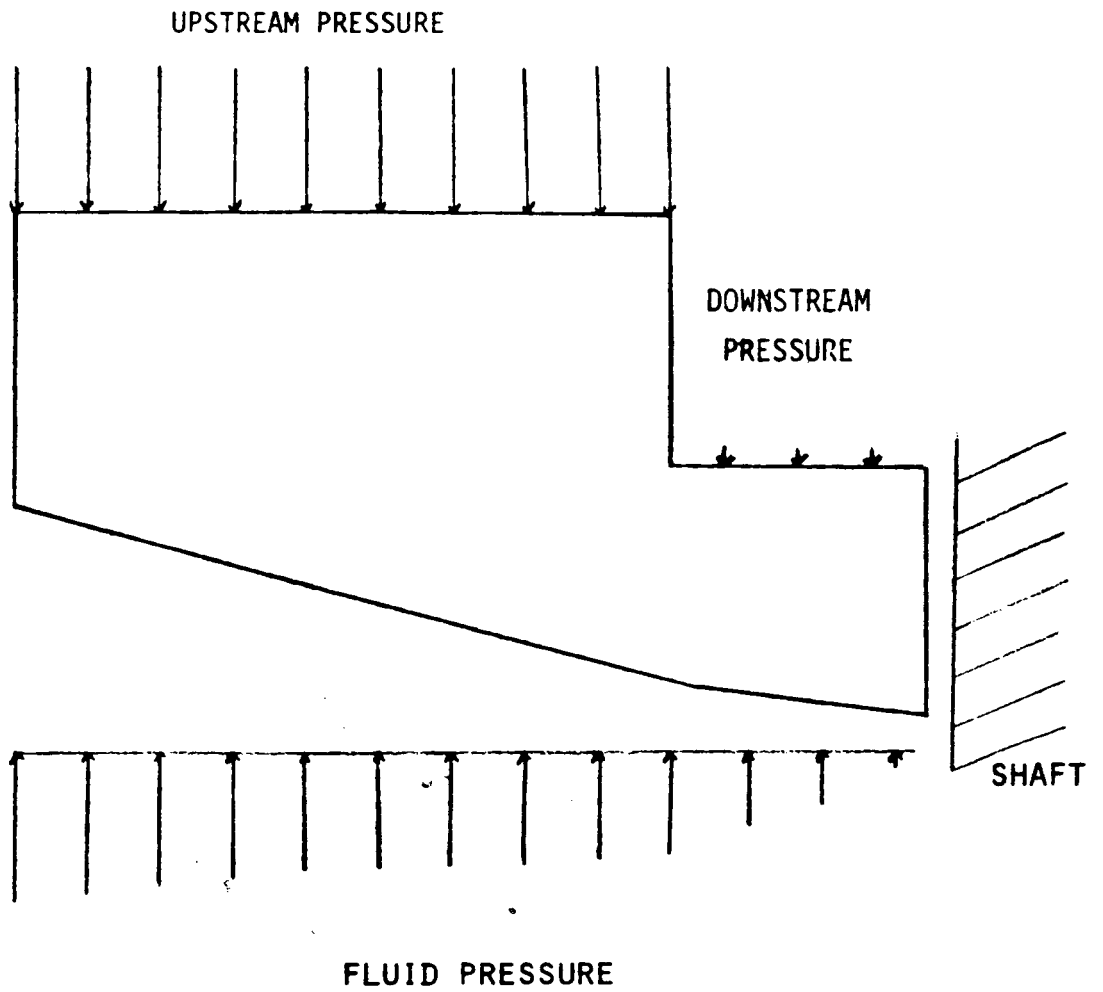
in this region. The correlations are based on the Dukler constant slip model, (ref. 2), which assumes that the ratio of the phase velocities to the average velocity is constant across a cross section. The flow model is based on the following assumptions:

1) thermodynamic equilibrium between phases

2) adiabatic

3) one dimensional flow in the axial direction

4) steady state flow

A detailed discussion of the flow model is given in Appendix B.

The flow path is along the pump shaft through the labyrinth seals, through the cartridge seal internals, and either through one of the three seal leakoff lines or out the top of the pump casing (see Figure 1). If the no. 1 seal is functional, the fluid enters the two-phase region as it passes through the seal. If the no. 1 seal is not functional, two-phase flow may begin in the labyrinth or in the no. 1 leakoff line. Although proprietary dimensional data excluded from this report was provided by Westinghouse so that an accurate analysis could be conducted.

The inputs to the flow model are reservoir pressure and enthalpy, flow path dimensions, and estimated flow rate. The flow rate estimate is modified until the downstream pressure condition is reached or until choked flow occurs. The downstream pressure condition for the no. 1 leakoff line was 165 psia. This is the set pressure specified by Westinghouse as typical for the no. 1 seal leakoff line pressure relief valve. For the other leakoff lines and for flow out of the top of the pump casing, ambient pressure, approximately 15 psia, was the downstream pressure boundary condition.

2.3 STRUCTURAL MODEL

The deformations of the three seal assemblies were analyzed using the ANSYS program, reference 3. The ANSYS computer program is a general purpose, finite element computer program for the solution of linear and nonlinear structural problems. The matrix displacement method of analysis based upon finite element idealization is used. The ANSYS program was used to model the three dimensional seal assemblies. Pressure and temperature loading were applied to the seal models.

3.0 RESULTS

The RCP seal analysis addressed the following:

1) the effect seal failure has on RCP leakage

2) the effect of system cooldown and depressurization on leakage rates and

3) the sensitivity of leakage rates to dimensional and thermal hydraulic parameters.

For each of the three cases, it is assumed that pump rotation stops immediately after station blackout occurs. A discussion of the results for (1) all three seals function, (2) no. 1 seal fails open while the no. 2 and 3 seals function, and (3) all three seals fail is presented in Sections 3.1 through 3.3, respectively. Figure 3 graphically summarizes the results for reactor coolant system (RCS) conditions of 2250 psia and 550°F. Section 3.4 presents a brief, general discussion of results pertaining to O-ring and channel seals. Section 3.5 discusses uncertainty associated with the entire analysis.

3.1 ALL THREE SEALS FUNCTION

The RCS pressure and temperature are nominally 2250 psia and 550°F during power plant operation. If loss of ac power and station blackout occur, RCS water at these conditions would leak into the RCP seal assembly. The temperature change would cause a change in leakage rate. ETEC calculated a leakage rate of 19.6 gpm with RCS pressure equal to 2250 psia and a RCS temperature of 550°F assuming all three seals function. This predicted leakage rate is slightly less than the 21.1 gpm leak rate predicted by Westinghouse for the same conditions. The nominal leak rate for the RCP seal assembly during normal plant operation when injection water is available, is 3 gpm.

3.1.1 Initial RCS Conditions

As previously indicated, the initial RCS conditions following a station blackout are 2250 psia and 550°F. In order to calculate a leakage rate through the RCP seal assembly at these conditions, the three models described in Section 2 were used. The flow model was used to calculate the pressure drop due to single and two-phase flow through all flow paths excluding the flow through the seals themselves. The seal model was used to calculate the pressure distribution across the seal faceplate and the gap between the faceplates. The structural model was used to determine the included angle between the faceplates due to deformation caused by the temperature and pressure differentials across the seal.

The ANSYS model was run to determine the response of the no. 1 seal to temperature and pressure loadings. The model independently determines response to the pressure and temperature loadings. Curves of seal rotation versus differential pressure and temperature were generated, and are illustrated in Figures 4 and 5. The curves represent linear interpolation of response for pressure or temperature differentials.
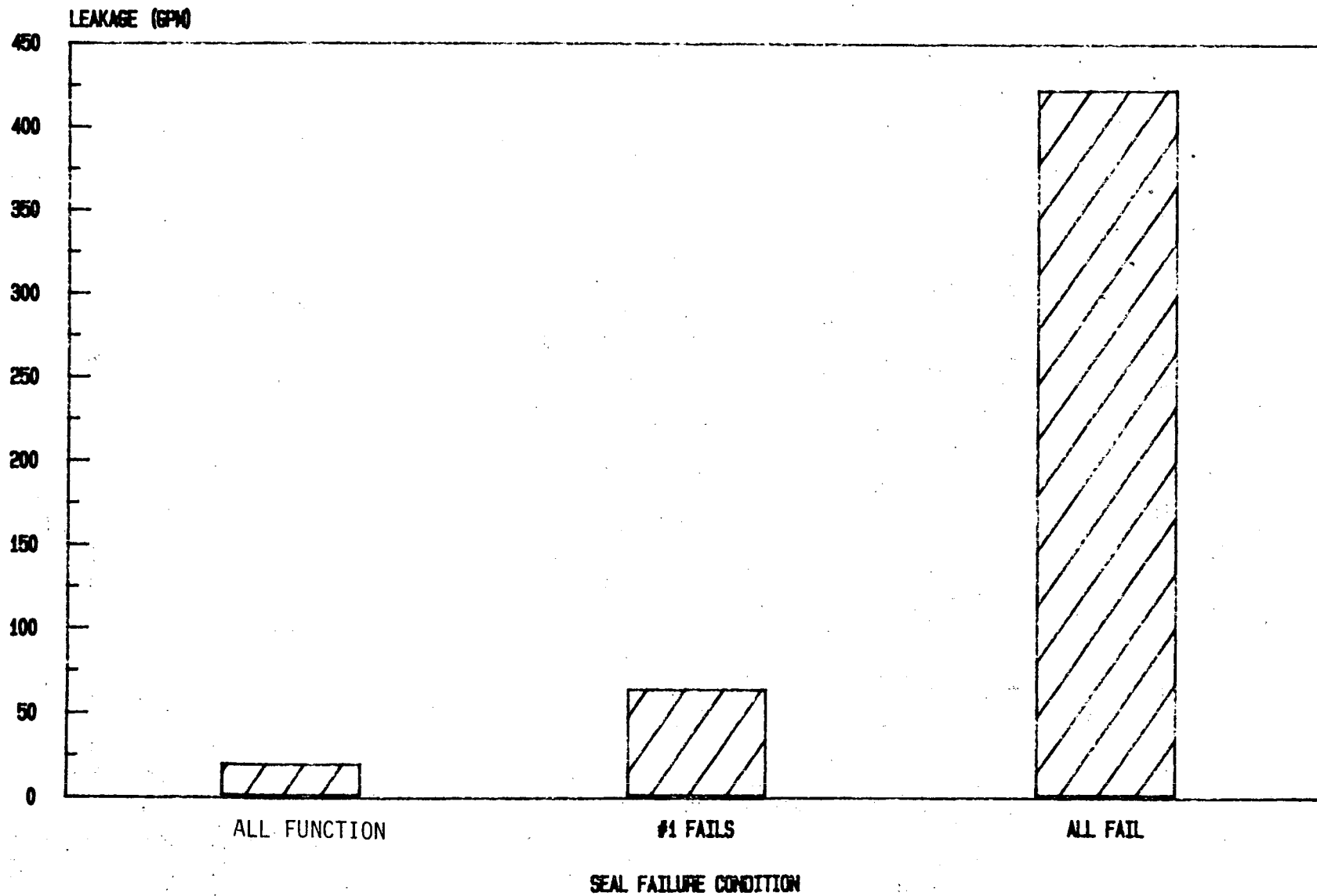
FIGURE 3
COMPARATIVE LEAKAGE RATES

2250 PSIA & 550 F

SEAL FAILURE CONDITION

FIGURE 4:
DECREASE #1 SEAL ANGLE VS DELTA PRESSURE

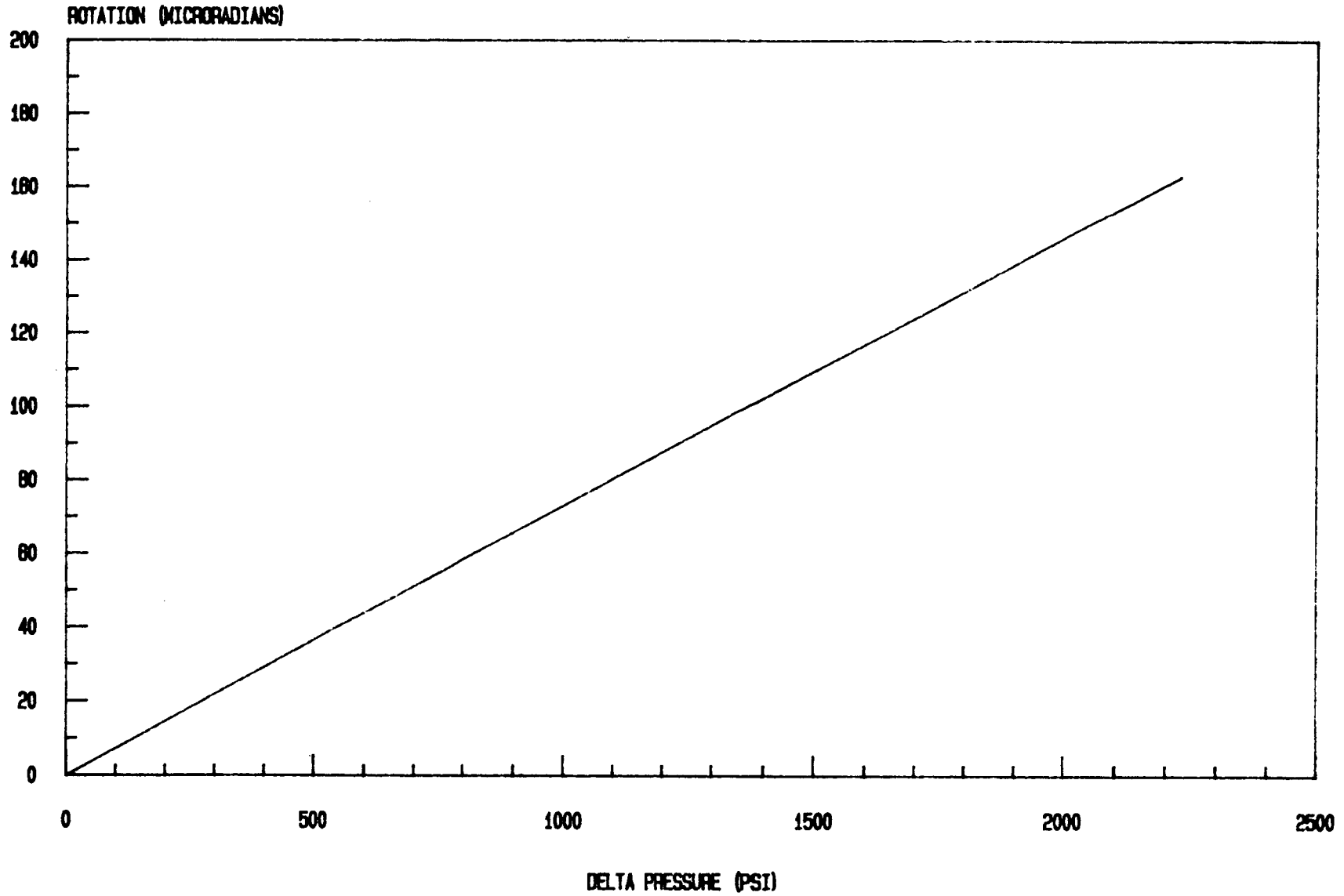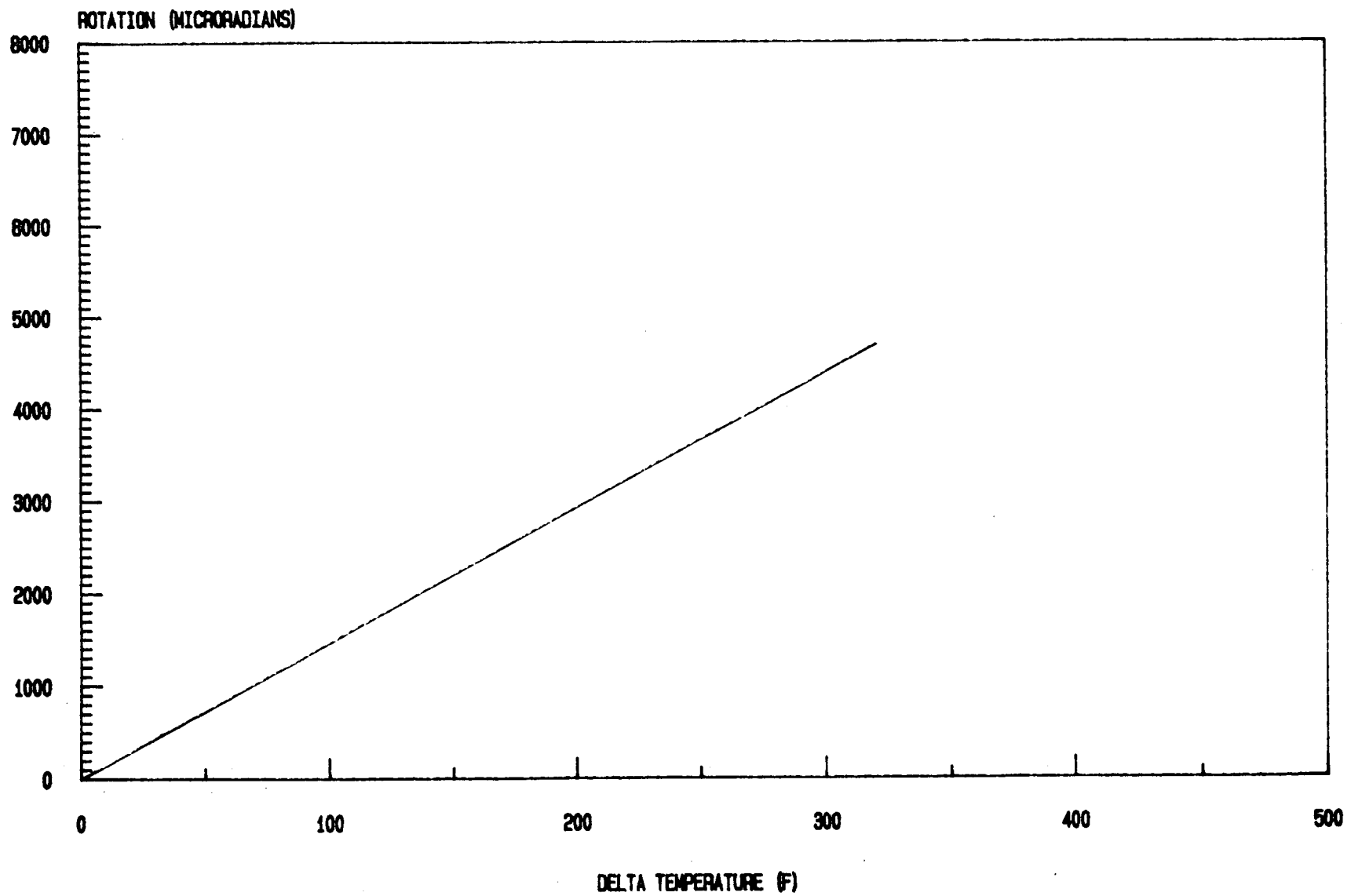ROTATION (MICRORADIANS)

DELTA PRESSURE (PSI)

11

FIGURE 5:
INCREASE IN #1 SEAL ANGLE VS DELTA TEMP.

The temperature and pressure differentials used in the ANSYS structural model were used for developement of Figures 4 and 5. The selection of the temperature differential and pressure distribution used as inputs for the model was arbitary. For the thermal analysis of the no. 1 seal, ETEC assumed 550°F upstream of the seal and 518°F downstream. These conditions were chosen because they are the conditions predicted by Westinghouse for the no. 1 seal when all three seals function during a station blackout. Figure 6 is an isotherm plot of the no. 1 seal subjected to this thermal gradient. As discussed later, ETEC predicted different temperature conditions for the no. 1 seal. However, by selecting the Westinghouse conditions for ANSYS model inputs, ETEC was able to compare results of seal rotation versus thermal gradient across the no. 1 seal. The ETEC results are shown in Figure 5 and are similar to those presented by Westinghouse in reference 1. Note that the included angle increases as the temperature gradient increases.

The nominal pressure conditions during normal pump operation, 2250 psia upstream and 30 psia downstream, were used for analysis of the no. 1 seal rotation due to differential pressure. The pressure gradient along the seal faceplates was obtained from the seal model described in Section 2.1. These boundary conditions were chosen because the pressure gradient across the no. 1 seal during normal operation, that is with a.c. power and cooling water available, is greater than the pressure gradient across the no. 1 seal during station blackout. The seal rotation due to pressure gradients across the faceplates therefore is bounded by the rotation caused by the normal 2220 psia pressure differential and a zero pressure gradient. Figure 4 illustrates the dependency of seal rotation on differential pressure for the no. 1 seal. This graph shows that increases in the differential pressure cause the seal to rotate closed. This is in contrast to the reaction of the no. 1 seal to increases in the temperature gradient. Therefore, the seal opening rotation due to the temperature gradient and the closing rotation caused by the differential pressure tend to offset each other.

Figures 4 and 5 were used to determine angular rotation of the seal faceplates. An iterative process was used to reconcile the results of the flow and seal models with the structural model. The first step was to assume an included angle between the no. 1 seal faceplates for input into the seal model. The seal model predicted a pressure differential and a resultant temperature drop based upon this included angle. A new included angle was then generated based on the upstream and downstream pressures and temperatures predicted by the flow and seal models. Using this method, conditions for the no. 1 seal functioning during station blackout are 2247 psia and 550°F upstream and 935 psia and 535°F downstream.

The no. 2 and 3 rubbing face type seals rotate closed under normal power plant operating conditions when injection water is available to cool the seal assembly. During station blackout the no. 2 and 3 seals are subjected to higher temperatures which increase the gradient across the seals.

ETEC analyzed the no. 2 seal assuming 518°F upstream and 212°F downstream to determine if an increase in the temperature gradient would make the seal

13

FIGURE 6:

ISOTHERM MAP OF #1 SEAL

518 F (FLUID TEMP.)

520 F →

550 F (FLUID TEMPERATURE)

548 F ISOTHERM
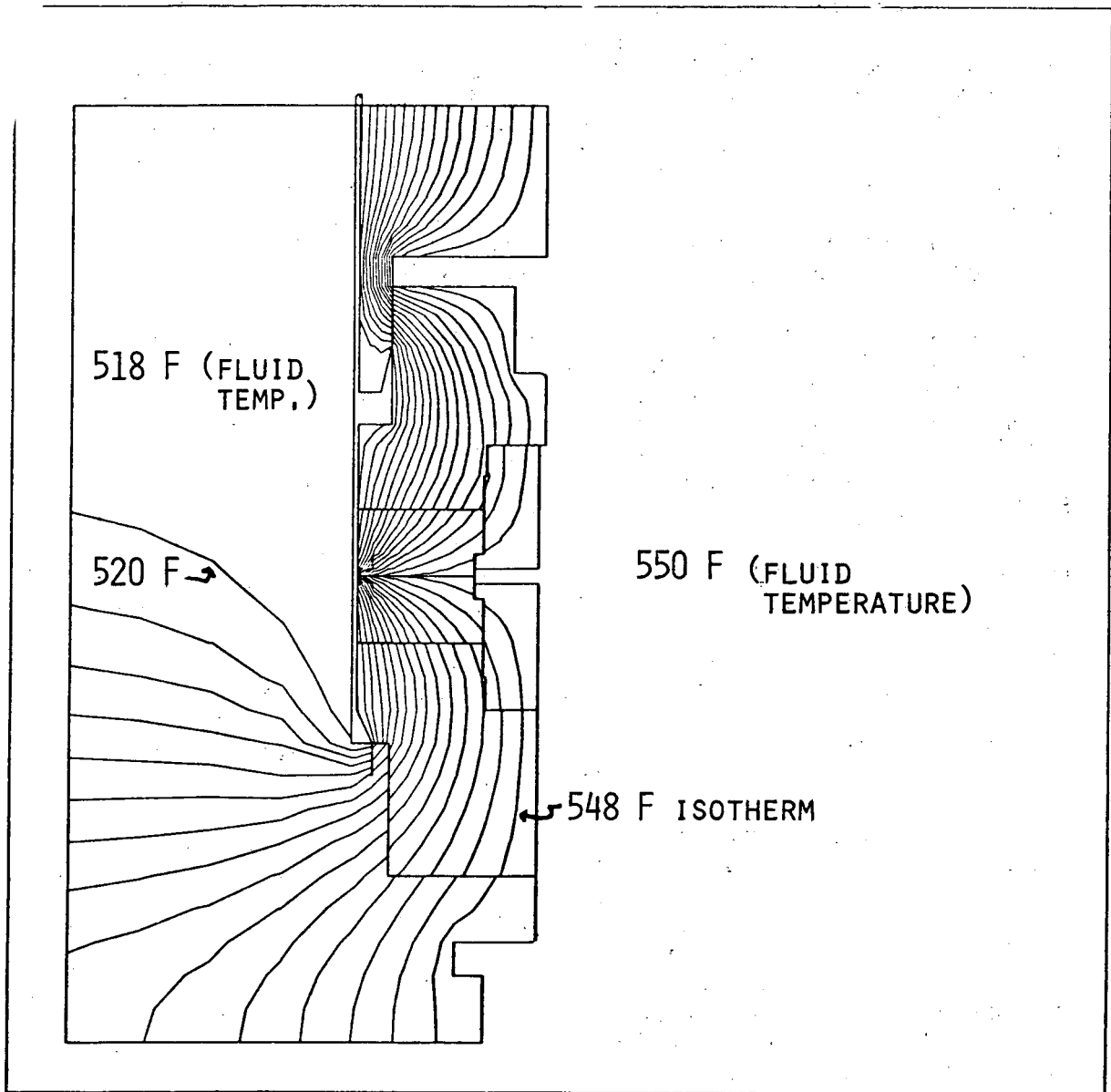
rotate closed. The temperatures used for the analysis were selected to identify whether an increase in temperature gradient caused the normally closed seal to rotate open or remain closed. If an increase in the temperature differential caused the seal to rotate open, it would be necessary to evaluate the flow through the no. 2 seal at each upstream condition. However, if an increase in temperature gradient produced a closing rotation of the seal, then only a minimal amount of flow will leak through regardless of the upstream conditions.

The isotherm plot from the thermal analysis of the no. 2 seal is shown in Figure 7. ETEC's analysis verified the Westinghouse prediction that the no. 2 seal's closing rotation would increase with differential temperature. Thus, the no. 2 seal will rotate closed whenever the temperature gradient exceeds the gradient experienced by the seal under normal operating conditions. This includes all the conditions predicted during a station blackout.

The actual conditions upstream of the no. 2 seal are determined by the reactor coolant system temperature and pressure and by the pressure drop across the labyrinth seal and the no. 1 seal. The ETEC analysis predicts that the temperature upstream of the no. 2 seal will be 535°F when the no. 1 seal is functioning. The temperature downstream of the no. 2 seal should be 212°F, the saturation temperature of water at ambient pressure. Westinghouse predicted 518°F upstream and 212°F downstream of the no. 2 seal. Therefore, the ETEC analysis indicates a larger temperature gradient across the no. 2 seal and therefore larger closing rotation.

3.1.2 RCS Cooldown and Depressurization

The leakage rate of 19.6 gpm reported for the case when all three seals function during a station blackout was based on the initial RCS conditions of 2250 psia and 550°F. However, the reactor coolant system pressure and temperature will decrease during the station blackout due to decreasing volume and possibly due to operator action. If the operators initiate cooldown procedures, Westinghouse predicts the reactor coolant system pressure and temperature will respond as shown in Figures 8 and 9 (ref. 1). The leakage rate with all three seals functioning was calculated at a number of these conditions. The results are tabulated in Table 1. These data were used to estimate the accumulated leakage versus time as shown in Figure 10.

If the operators do not initiate system cooldown procedures, the reactor coolant system pressure will still decay during the station blackout although Westinghouse predicts that the temperature will be fairly constant. These system conditions are presented in Figures 11 and 12. The calculated leakage rates are given in Table 2. For comparison, the accumulated leakage versus time corresponding to the no cooldown case is presented in Figure 10 alongside the curves of accumulated leakage for the cooldown case. The graph emphasizes the impact that operator action can have on the accumulated leakage and thus the time required to expose the core. The leakage rates for the cooldown and the no cooldown case would be the same for identical RCS conditions. The large difference in accumulated leakage is because the RCS remains at high temperature and pressure conditions which result in higher leakage rates.

FIGURE 7:
ISOTHERM MAP OF THE #2 SEAL

212 F (FLUID TEMP.)

518 F (FLUID TEMP.)

220 F ISOTHERM

500 F ISOTHERM
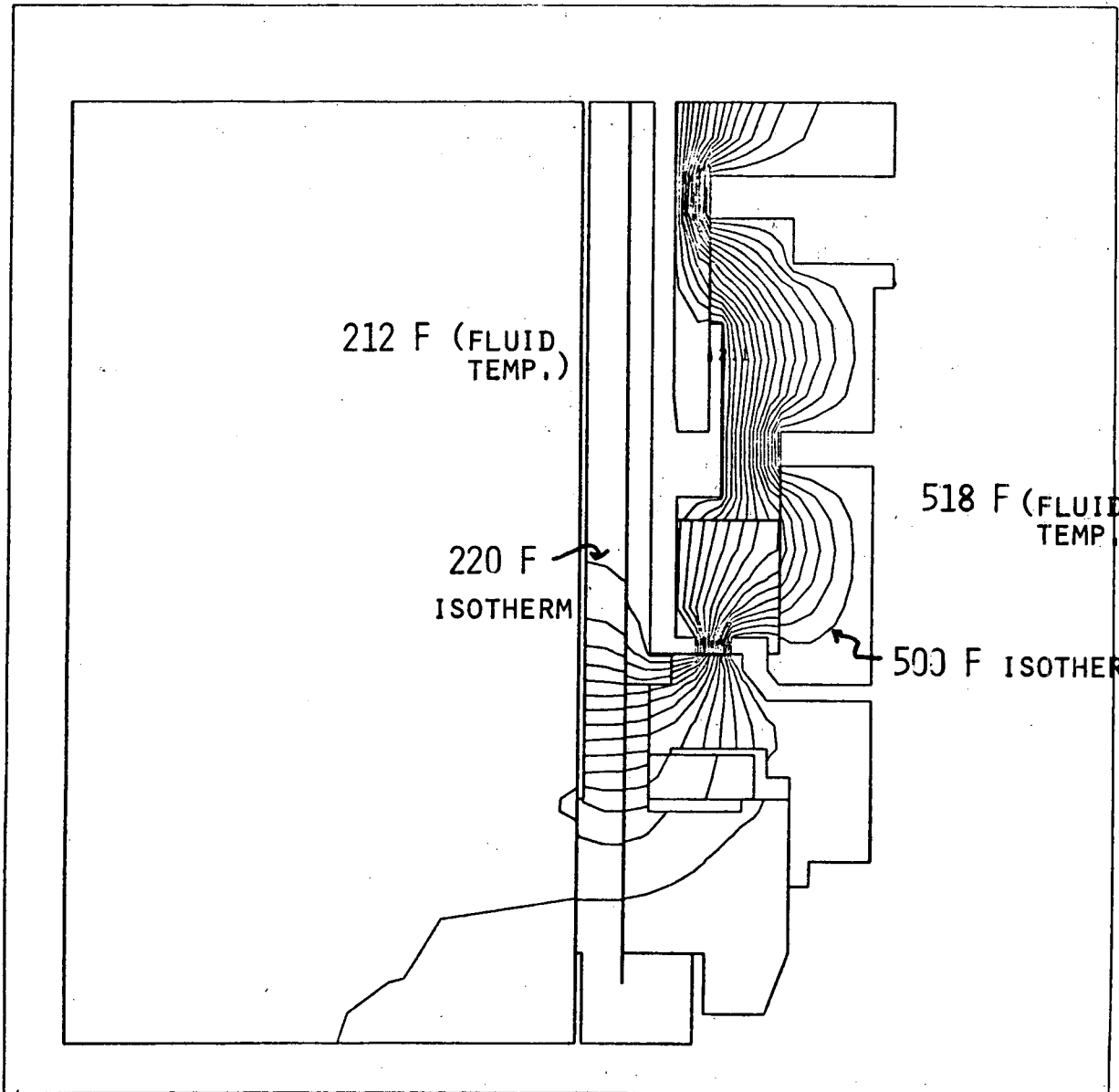
FIGURE 8:
REACTOR COOLANT SYSTEM PRESSURE VS. TIME
OPERATOR INITIATED COOLDOWN

PRESSURE (PSIA)

TIME (HR)
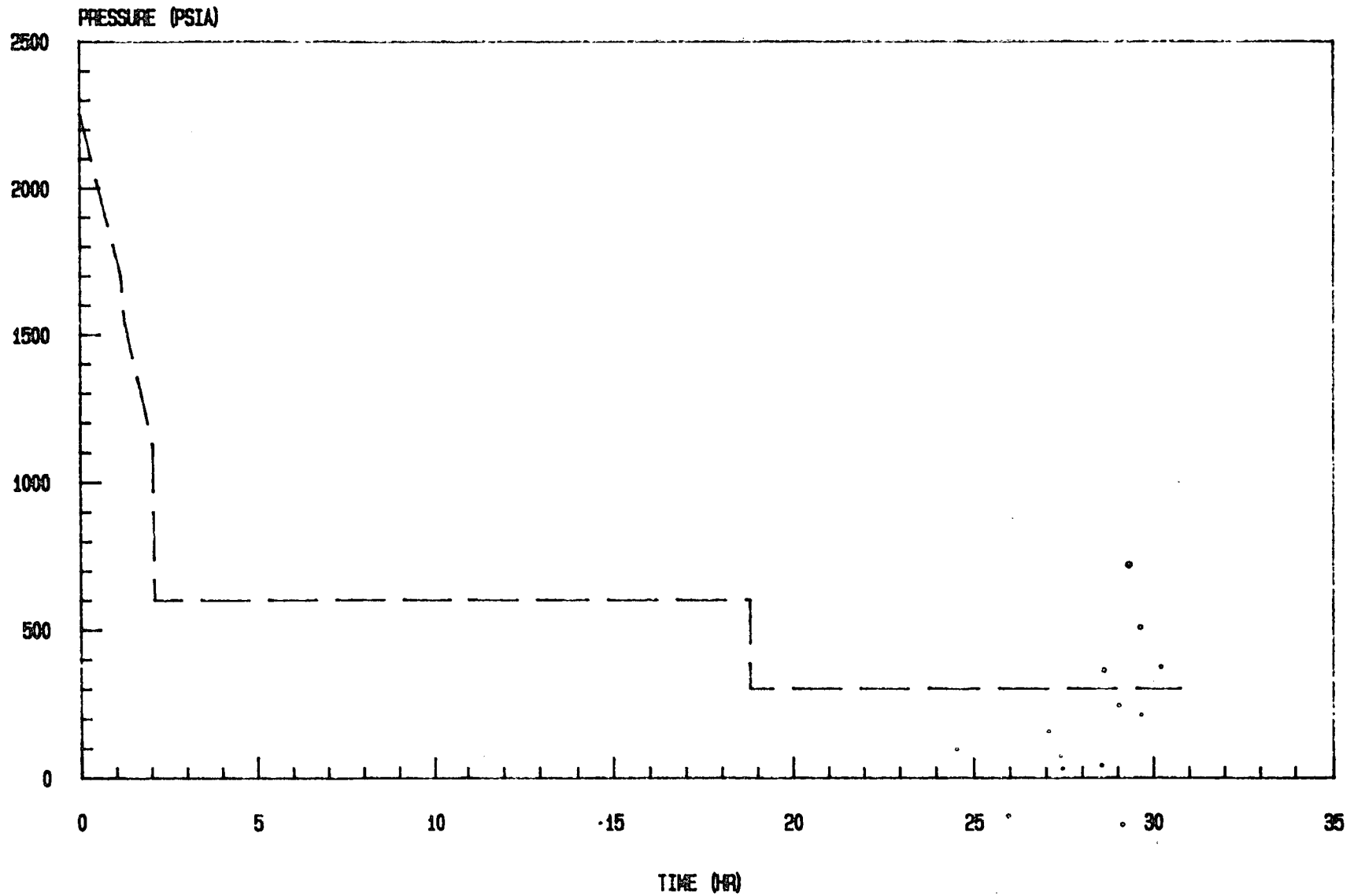
FIGURE 9:
RCS TEMPERATURE VS. TIME

OPERATOR INITIATED COOLDOWN

TEMPERATURE (F)

TIME (HR)
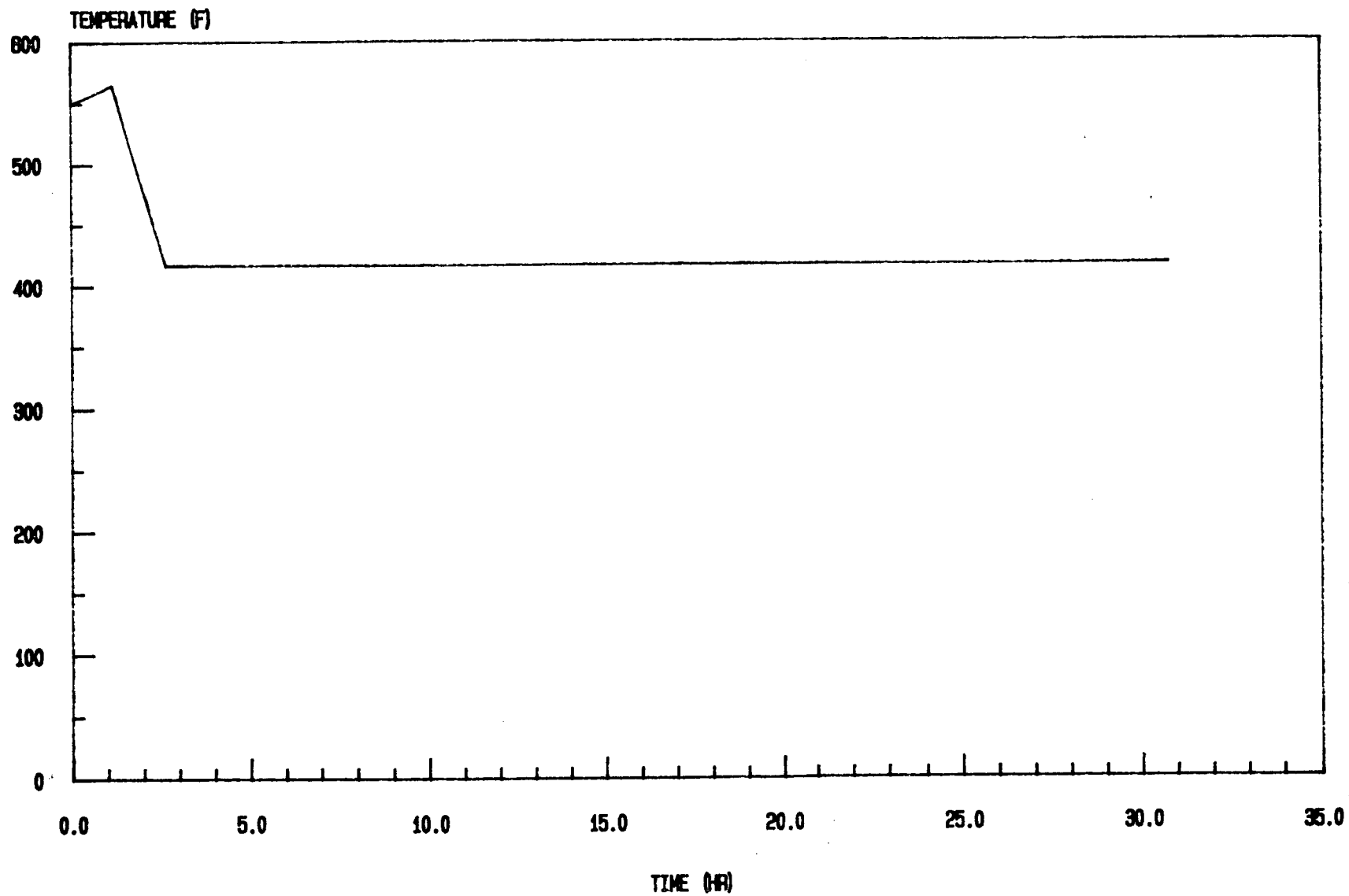
TABLE 1: LEAKAGE RATES DURING REACTOR COOLANT COOLDOWN
ALL SEALS FUNCTION

| Reactor Coolant System | | Leakage Rate | |
Pressure (psia)	Temperature (^{O}F)	lbm/sec	gpm*
2250	550	2.05	19.6
600	444	.92	8.0
300	417	.46	3.9

*Based on Reactor Coolant System density

FIGURE 10:
ACCUMULATED LEAKAGE VS. TIME
ALL SEALS FUNCTION
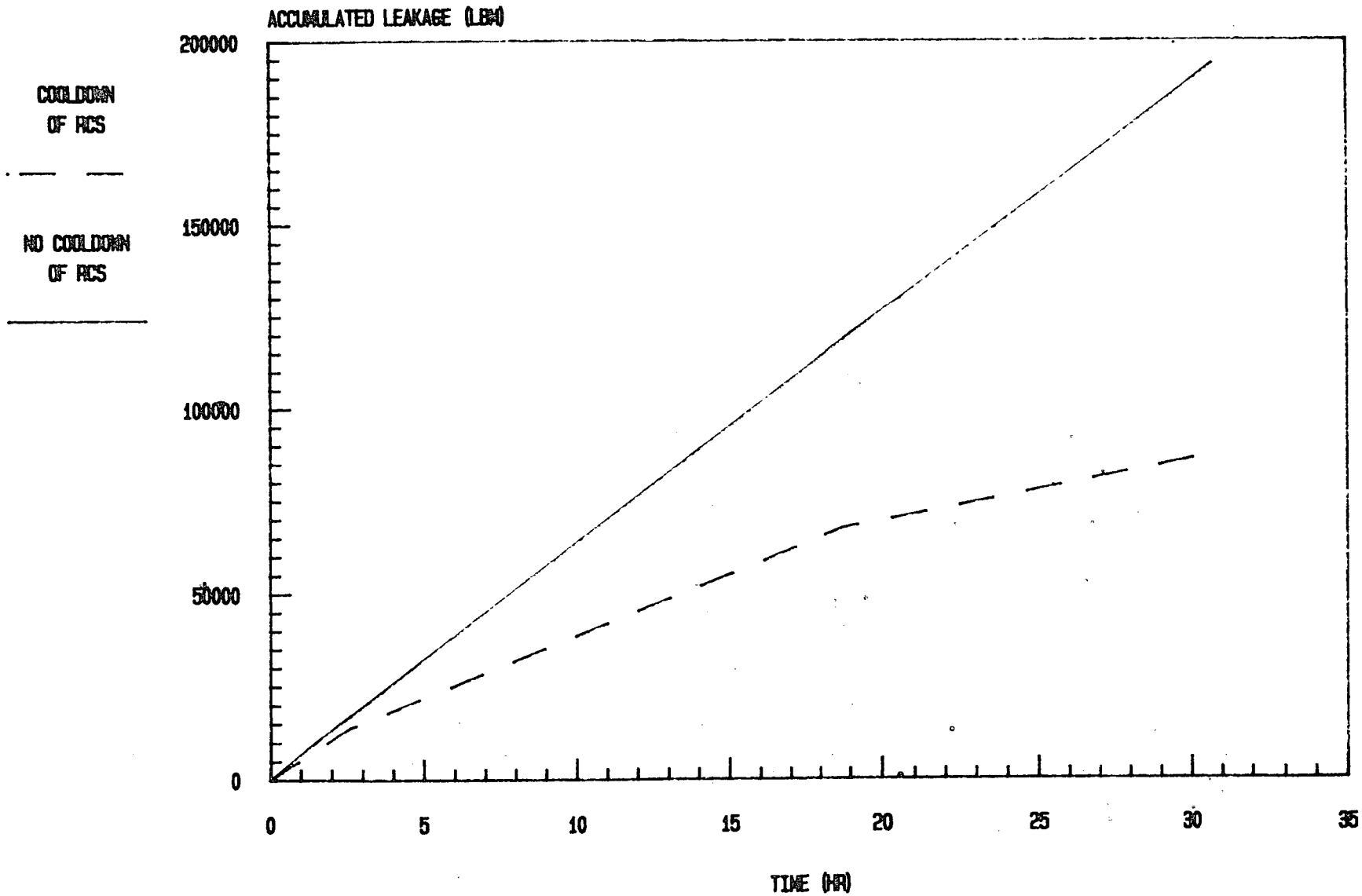
FIGURE 11:
REACTOR COOLANT SYSTEM PRESSURE VS. TIME

NO COOLDOWN

PRESSURE (PSIA)

TIME (HR)
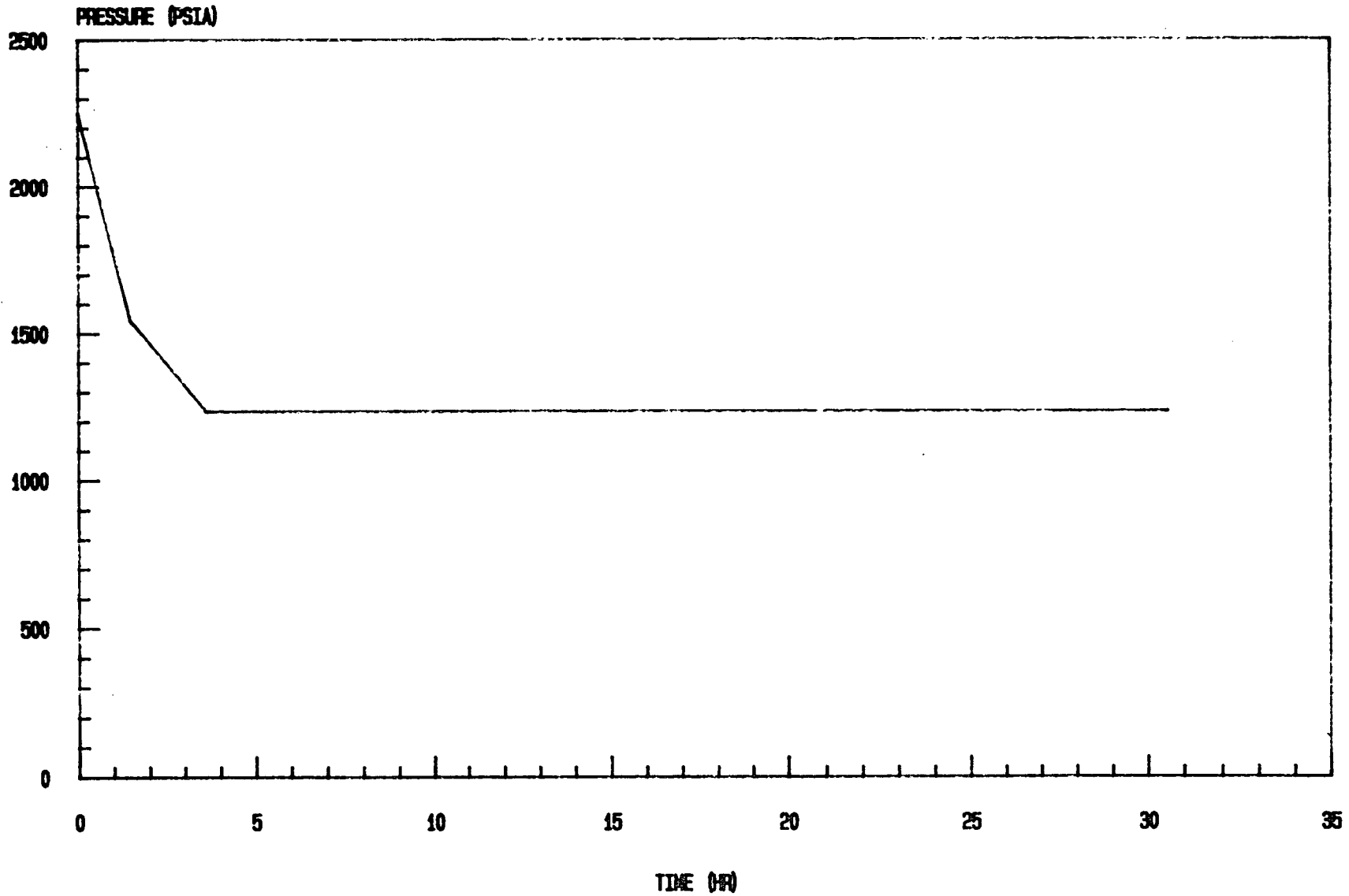
FIGURE 12 :
RCS TEMPERATURE VS. TIME
NO COOLDOWN

TEMPERATURE (F)

TIME (HR)
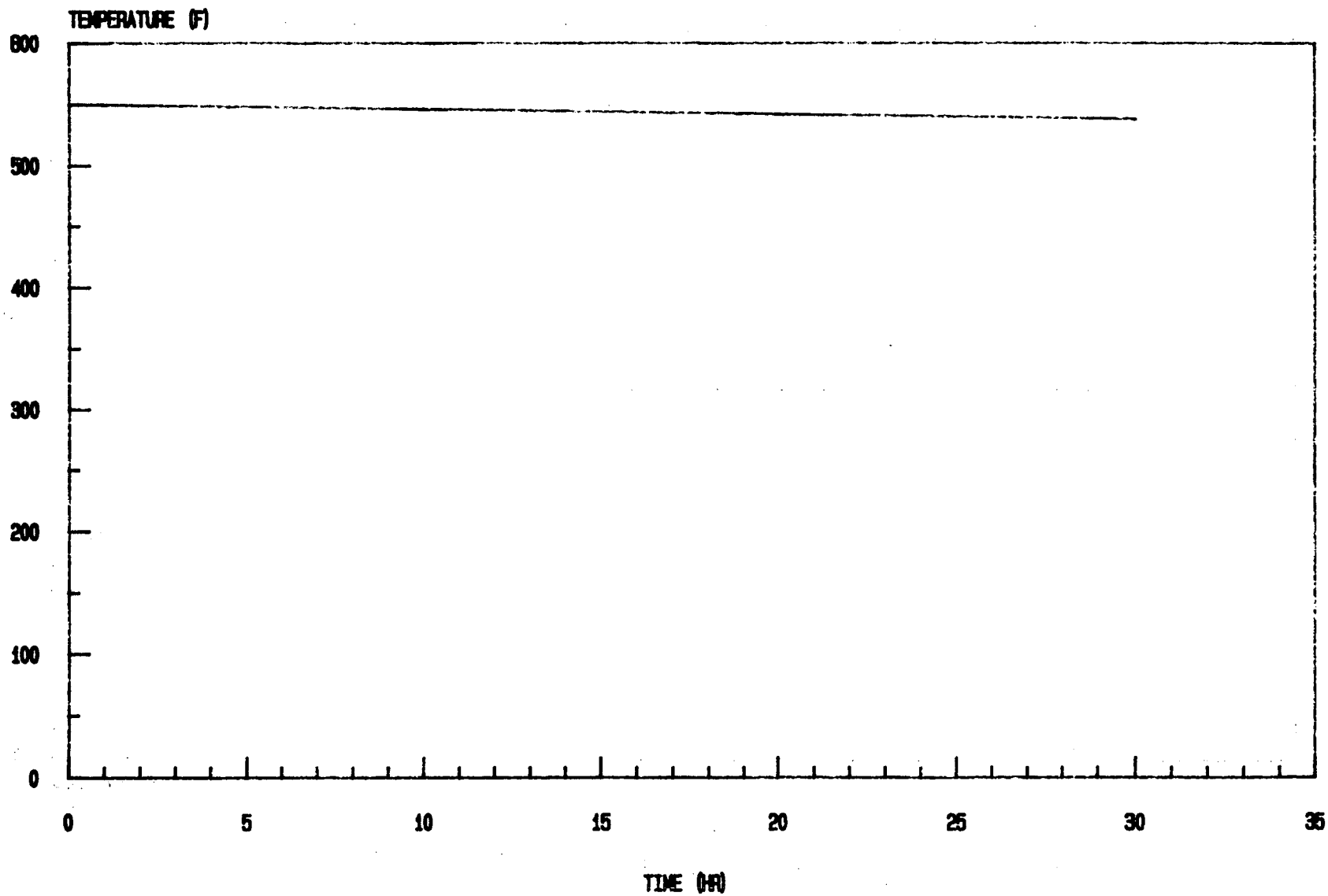
TABLE 2: LEAKAGE RATES WITHOUT REACTOR COOLANT SYSTEM COOLDOWN
          ALL SEALS FUNCTION

| Reactor Coolant System | | Leakage Rate | |
Pressure (psia)	Temperature (^{O}F)	lbm/sec	gpm*
2250	550	2.05	19.6
1545	549.3	1.75	16.9
1235	548.6	1.74	16.9

*Based on Reactor Coolant System density

3.1.3 Sensitivity Studies

The ANSYS structural model was used to determine the influence of several thermal parameters on seal faceplate included angle and the calculated leakage rate. The first parameter investigated was the heat transfer coefficient used at the fluid/seal interface. ETEC initially used a heat transfer coefficient of 14 Btu/(hr*in*in*F), a value in the typical range for water/metal heat transfer. Through discussions with Westinghouse and examination of the isotherms Westinghouse presented for the no. 1 and no. 2 seals, see reference 1, it became evident that Westinghouse had used a higher heat transfer coefficient. ETEC therefore repeated the ANSYS analysis using a heat transfer coefficient of 1000 Btu/(hr*in*in*F). Use of this very high heat transfer coefficient resulted in approximately equal fluid and metal temperatures. The temperature drop across the seal was larger when the higher heat transfer coefficient was assumed. The larger temperature drop produced a larger included angle between the seal faceplates, causing a larger leak rate. ETEC used the high heat transfer coefficient, 1000 Btu/(hr*in*in*F), because it produced more conservative results.

The assumption of isothermal seal faceplates was compared to the assumption of no heat transfer across the faceplate, see Figures 13 and 14. The no heat transfer assumption means the temperature of the faceplates is the same as the temperature of the fluid in contact with the faceplates. The assumption of isothermal seal faceplates means that the entire surface of the seal faceplate is at the same temperature, the temperature of the incoming fluid. The isothermal faceplate assumption gives a greater opening rotation of the seal faceplates, which allows a greater leak rate through the seal. ETEC determined that the assumption of isothermal faceplates was reasonable during their study. The seal model, see section 2.1, predicted that flashing of the leakage water will not occur until the fluid reaches the downstream edge of the no. 1 seal. Before the fluid flashes, it is subcooled and so the temperature remains constant as the pressure decreases. Therefore, even if the no heat transfer assumption was made, the no. 1 seal faceplates would still be virtually isothermal. ETEC assumed the faceplates would be isothermal because this assumption appears to approximate the true situation but is somewhat more conservative. Westinghouse also assumed the faceplates are isothermal.

The effect of machining tolerance on the leakage rate through a functioning no. 1 seal was examined. The rolling stiffness of the support ring with the faceplate is predicted to change approximately 1.1% from a seal with the nominal machined dimensions. The rolling stiffness influences the rotation of the no. 1 seal. The effect of dimensional tolerances on the axial force balance was also calculated. The axial force balance determines the gap between the seal faceplates. The allowable dimensional deviation is predicted to change the axial force by only 0.73% from the nominal case. Machining tolerances therefore would have only a minor impact on the calculated leakage rates.

3.2 NO. 1 SEAL FAILURE

For the case assuming that the no. 1 seal fails open while the no. 2 and 3 seals continue to function, ETEC predicted a leak rate of 64.7gpm through the

FIGURE 13

ISOTHERM MAP OF #1 SEAL
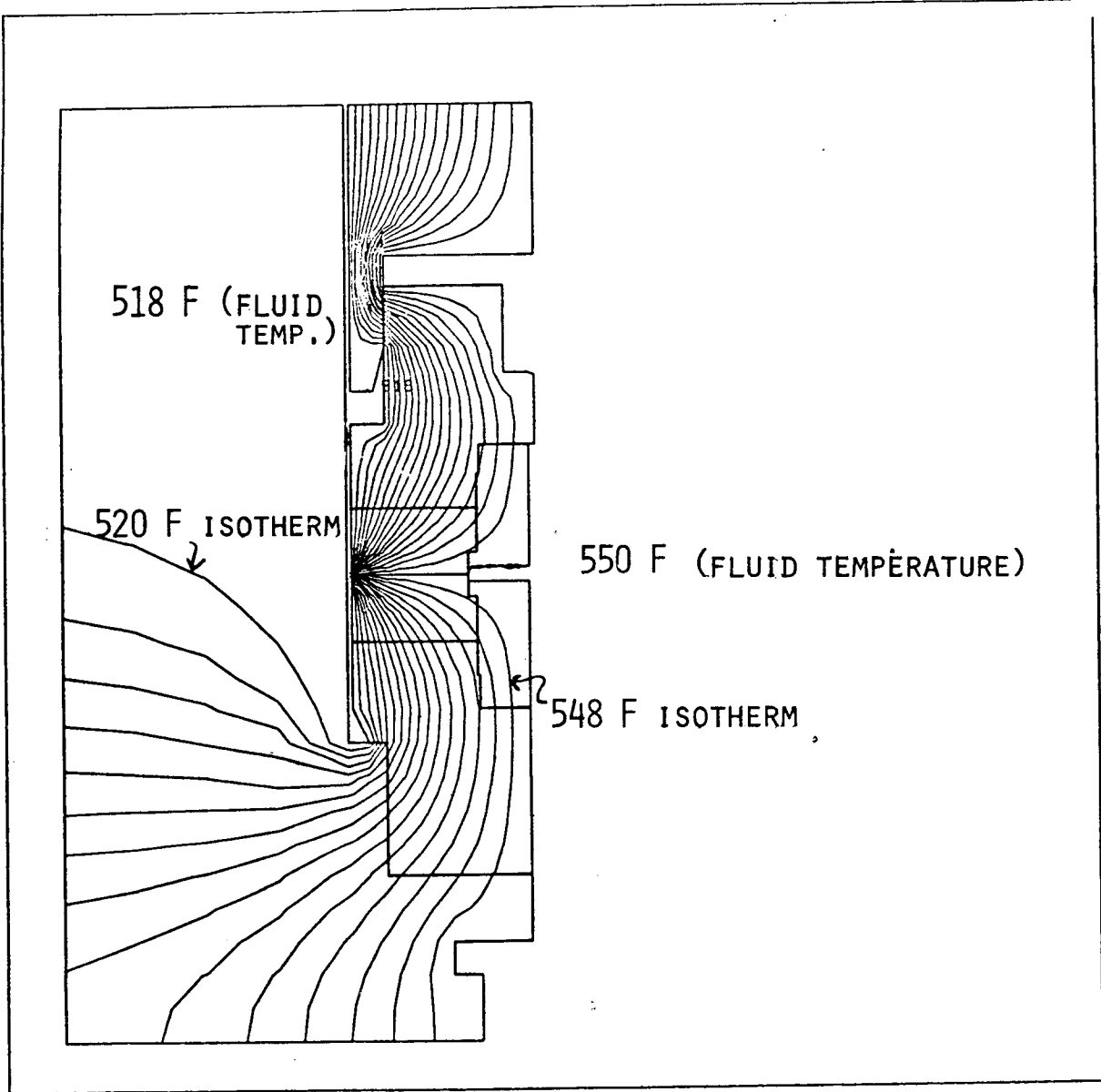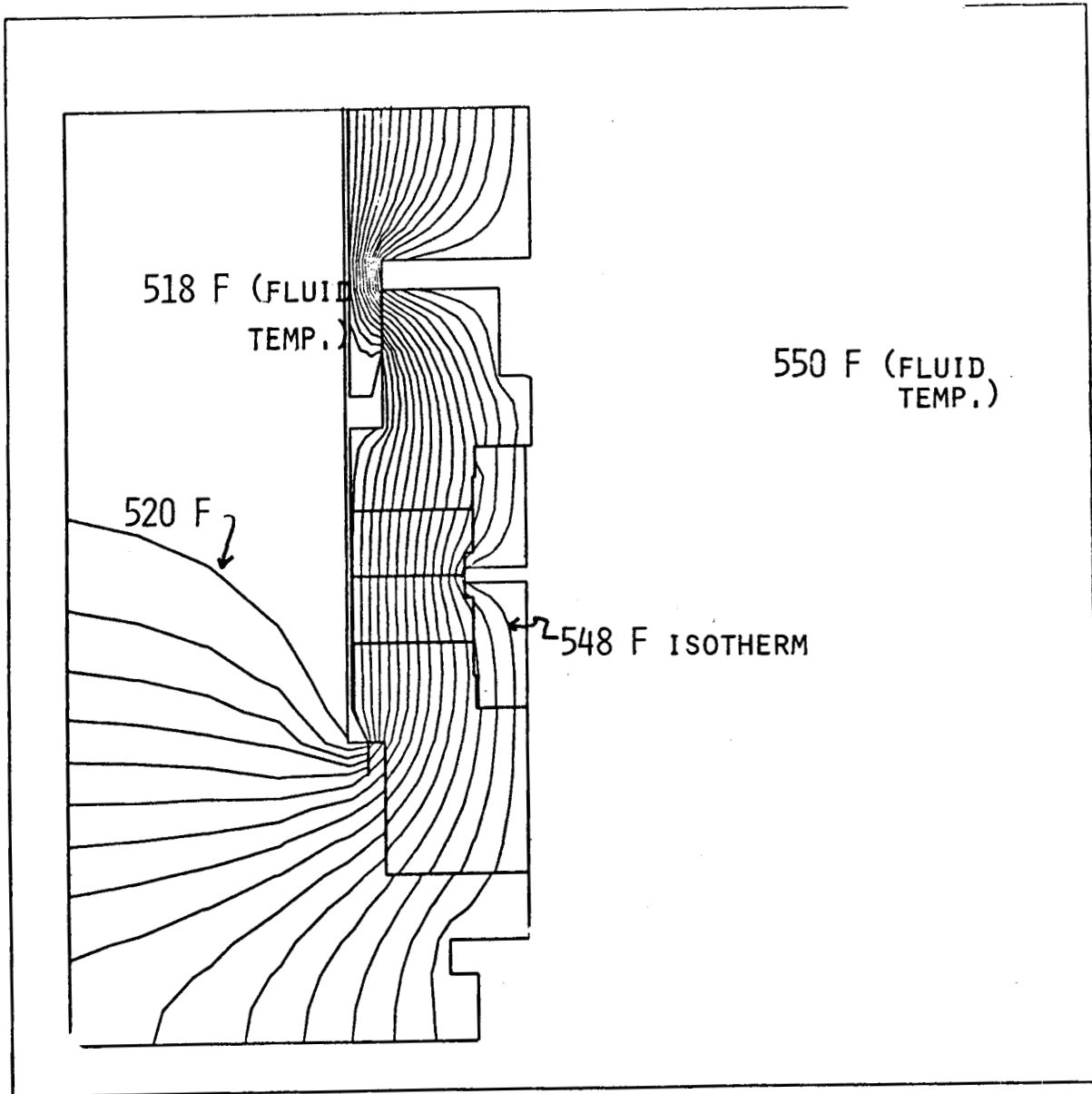### ISOTHERMAL FACEPLATES

518 F (FLUID TEMP.)

520 F ISOTHERM

550 F (FLUID TEMPERATURE)

548 F ISOTHERM

FIGURE 14

ISOTHERMAL MAP OF #1 SEAL

NO HEAT TRANSFER ACROSS FACEPLATES

518 F (FLUID TEMP.)

550 F (FLUID TEMP.)

520 F

548 F ISOTHERM

RCP seal assembly with RCS at 2250 psia and 550°F. This is comparable to the 75.6 gpm leak rate predicted by Westinghouse for the same conditions.

3.2.1 Initial RCS Conditions

The initial RCS conditions during a station blackout were assumed to be 2250 psia and 550°F. These are the nominal RCS conditions during normal power plant operation. It also was assumed that failure of the no. 1 seal occurred immediately upon station blackout and that pump rotation has stopped. This is a conservative assumption which gives the maximum leak rate.

Two of the models described in section 2 were used, for this analysis. The seal model, which defines the gap and pressure distribution across the no. 1 film-riding seal, was not required. The structural model was used to determine the response of the no. 2 and 3 rubbing face seals. As for the case when all three seals function, the temperature gradient across the no. 2 seal causes it to rotate closed. The conditions upstream of the no. 2 seal are approximately 2220 psia and 550°F. Downstream of the no. 2 seal, the fluid will be at saturation; 15 psia and 212°F. The temperature gradient across the no. 2 seal will be greater than if the no. 1 seal functions so the closing rotation will be larger.

The primary leak path is through the no. 1 seal and the no. 1 leakoff line with only minimal flow through the no. 2 seal. If the no. 2 seal faceplates are machined perfectly flat, there would be no leakage across a closed no. 2 seal. In practice, there will be some leakage due to machining tolerances. The leakage past the no. 2 seal when it rotates closed was therefore estimated. Based on the maximum permitted circumferential 'wave' for the no. 2 seal runner face, a leakage rate of approximately 0.1 gpm was calculated. Upstream conditions of 2250 psia, 550°F and downstream conditions of 15 psia, 212°F were chosen for the analysis to give the maximum leakage rate. This leakage of 0.1 gpm is insignificant compared with the overall leakage rates. Since the no. 3 seal is downstream of the no. 2 seal, leakage across the no. 3 seal is limited by no. 2 seal leakage.

3.2.2 RCS Cooldown and Depressurization

As discussed in section 3.1.2 for the case in which all three seals function, the leak rate through the RCP seal assembly will decrease due to system cooldown and depressurization. If the operators initiate cooldown procedures, Westinghouse predicts the RCS temperature and pressure profiles versus time will be as shown in Figures 8 and 9. These temperature and pressure responses are dependent on the volumetric capacity of the equipment and on the time it takes to drain the various portions of the system. The time it takes to drain depends on the leakage rate at different RCS pressures and temperatures. The RCS conditions versus time presented in Figures 8 and 9 are based on a four-loop plant and the leak rates Westinghouse predicted, assuming that all three seals function throughout the station blackout. ETEC used these same conditions in the analysis of accumulated leakage versus time for the case in which the no. 1 seal fails open while the no. 2 and 3 seals function. Since the leakage rates are higher when the no. 1 seal fails, some error is introduced with this method. In fact, the depressurization of the system would occur more quickly if the no. 1 seal failed because the system would be draining faster. This indicates that the accumulated leakage versus time

27

calculated by ETEC for the case with the no. 1 seal failed and the no. 2 and 3 seals functioning is conservative. That is it gives greater accumulated leakage.

The leakage rates calculated by ETEC for various RCS conditions following operator initiated cooldown are summarized in Table 3. These data were used to estimate the accumulated leakage versus time as shown in Figure 10. As discussed above, these results should be conservative.

If the operators do not initiate system cooldown procedures, Westinghouse predicts the RCS will respond as shown in Figures 11 and 12. These pressure and temperature time histories are also based on the leakage rates calculated when all three seals function. Therefore, the error discussed above extends to ETEC's calculation of accumulated leakage versus time without operator initiated cooldown for the case in which the no. 1 seal fails while the nos. 2 and 3 function. The leakage rates at various RCS conditions are given in Table 4. The accumulated leakage versus without operator initiated cooldown is presented in Figure 15 with accumulated leakage for operator cooldown.

3.2.3 Sensitivity Studies

The dominant flow resistance when the no. 1 seal fails open and the no. 2 and 3 seals function is the no. 1 seal leakoff line. ETEC calculated a pressure drop of approximately 2035 psia across this leakoff line. These lines vary from plant to plant. ETEC used a 'typical' configuration provided by Westinghouse. An isometric of this line is shown as Figure 16. Since there is variation in these lines among plants, ETEC evaluated the response of the leakage rate to changes in the line configuration.

Two alternative configurations illustrated in Figures 17 and 18 were examined. The response of the leakage rate to modification of the leakoff line configuration is illustrated in Figure 19. The results indicate that the leakoff line arrangement, as well as the overall equivalent flow resistance (L/D) affects the leakage rate. The leakoff configuration shown in Figure 17 differs from that shown in Figure 16 only by elimination of the last section of 2" diameter pipe. This modification has no effect on the leakage rate. However, if the section of 1 1/4" diameter piping located upstream of this 2" piping is removed, as shown in Figure 18, the leakage rate increases by 13%. This is because flow chokes in the final section of 1 1/4" pipe, and thus the relatively large pipe downstream does not limit the leakage rate.

3.3 ALL SEALS FAIL

The worst case leakage rate through the RCP seal assembly during a station blackout would occur if all three of the seals fail. For this case ETEC predicted a leakage rate of 422 gpm with RCS conditions of 2250 psia, 550°F. Westinghouse predicted a 480 gpm leakage rate under the same conditions.

3.3.1 Initial RCS Conditions

It was assumed that seal failure occurred immediately after loss of ac power and therefore the RCS conditions should be approximately the same as normal plant operation. The assumption of immediate failure gives the largest leakage rate.

TABLE 3: LEAKAGE RATES DURING REACTOR COOLANT SYSTEM COOLDOWN
#1 SEAL FAILS, #2 & #3 FUNCTION

| Reactor Coolant System | | Leakage Rate | |
Pressure (psia)	Temperature (^{O}F)	lbm/sec	gpm*
2250	550	6.76	64.7
1700	565	4.82	47.6
1540	555	4.53	44.2
1124	474	4.84	43.0
600	471	2.26	20.1
600	417	3.50	29.6
300	417	.95	8.1

*Based on Reactor Coolant System Density

TABLE 4: LEAKAGE RATES WITHOUT REACTOR COOLANT SYSTEM COOLDOWN
#1 SEAL FAILS, #2 & #3 SEALS FUNCTION

| Reactor Coolant System | | Leakage Rate | |
Pressure (psia)	Temperature (^{O}F)	lbm/sec	gpm*
2250	550	6.76	64.7
1545	549.3	4.66	45.1
1235	548.6	3.47	33.7

* Based on Reactor Coolant System density

FIGURE 15
ACCUMULATED LEAKAGE VS. TIME
#1 SEAL FAILS, #2 & 3 FUNCTION

ACCUMULATED LEAKAGE (LBM)

COOLDOWN
OF RCS

NO COOLDOWN
OF RCS

TIME (HR)
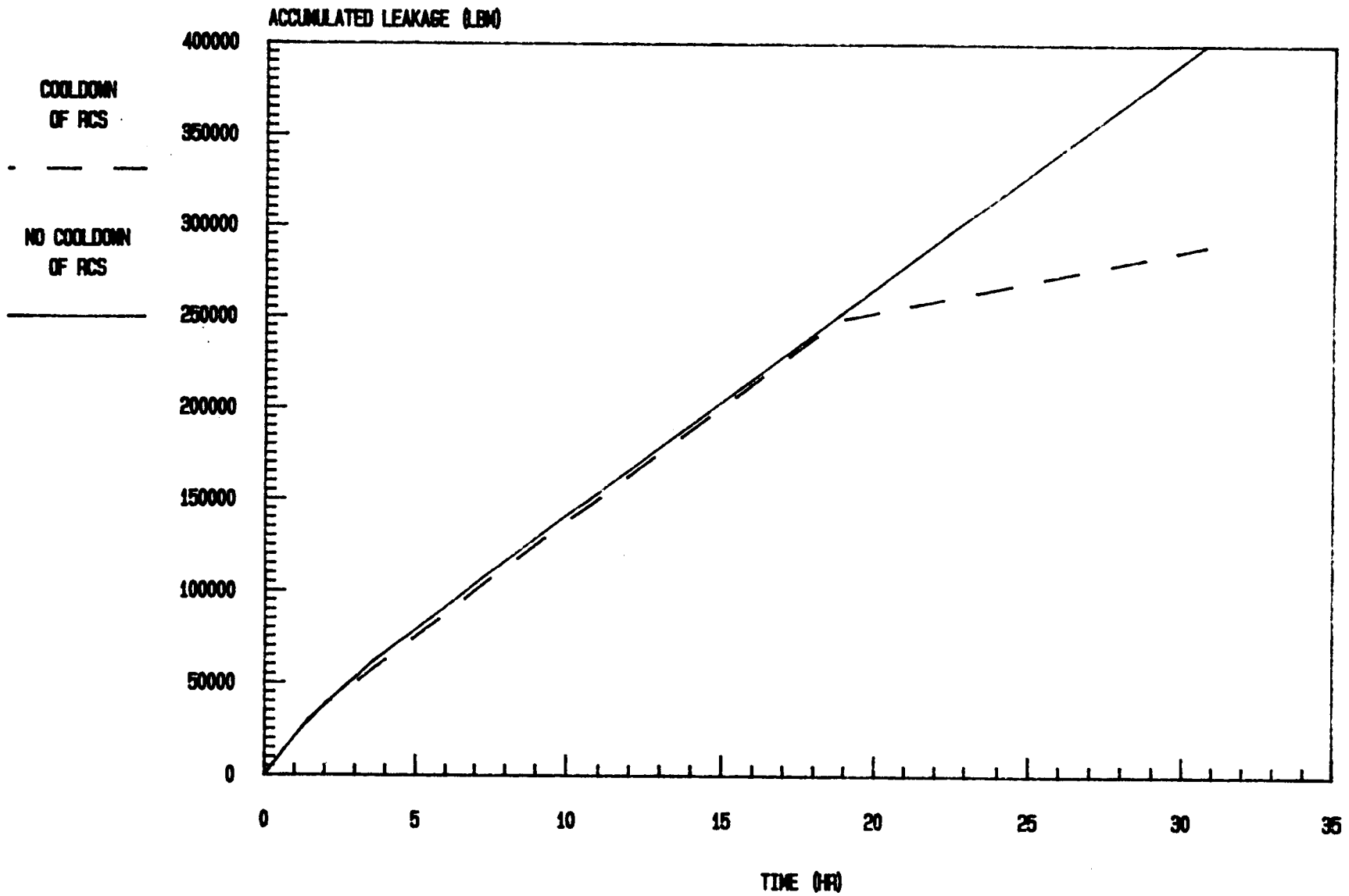
FIGURE 16

TYPICAL #1 SEAL LEAKOFF PIPING

TO ATMOSPHERE

6.55

START 2.0 IN.
SCH. 160
ID = 1.689 IN.

6.01

1.27

4.04

1.27

7.46

0.75

1.35

I/D = .340

1.0 EST

1.0 EST

1.81

4.18

30°

3.56

START 0.75 IN.
SCH. 160
ID = 0.614

3.33

4.5

1.08

START 0.75 IN.
SCH. 160
ID = 0.614

START 1.25 IN.
SCH. 80
ID = 1.278 IN.

1.97

24.6

1.167

5.88

START 2.0 IN.
SCH. 160
ID = 1.689 IN.

2.17

2.19

2.99

2.99

6.53

2.85 EST

CONNECTION
TO PCP NO. 1

SYMBOL KEY:

REDUCER OR EXPANDER

ORIFICE

VALVE

PIPE

NOTE:

ALL DIMENSIONS IN FEET

32
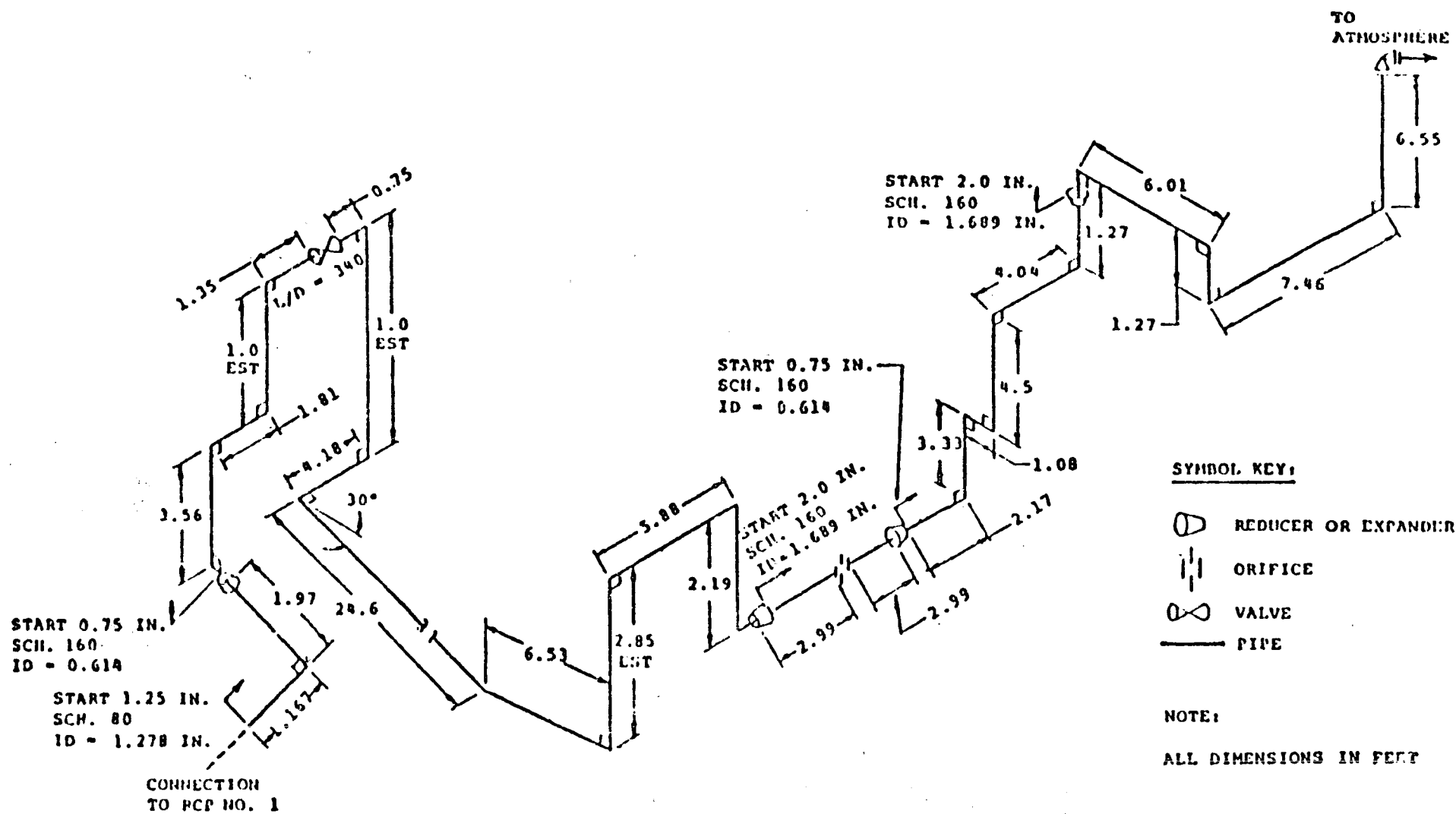
FIGURE 17

#1 SEAL LEAKOFF PIPING
WITHOUT FINAL SECTION OF 2" PIPING
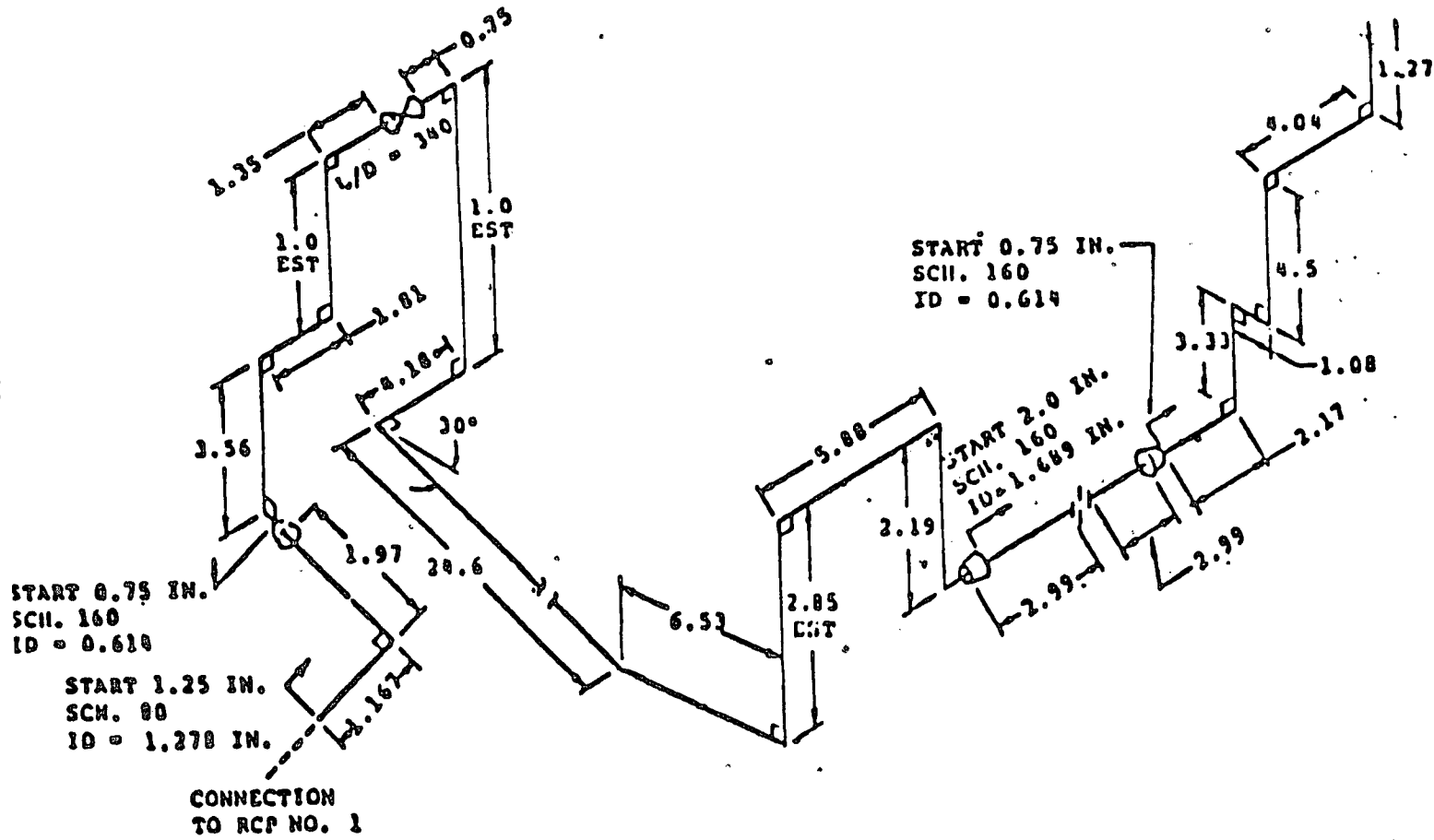
FIGURE 18

#1 SEAL LEAKOFF PIPING

WITHOUT FINAL SECTION OF .75" PIPING

START 0.75 IN.
SCH. 160
ID = 0.614

START 1.25 IN.
SCH. 80
ID = 1.278 IN.

CONNECTION
TO RCP NO. 1

START 2.0 IN.
SCH. 160
ID = 1.689 IN.

FIGURE 19
LEAKAGE VS. #1 LEAKOFF LINE RESISTANCE

#1 SEAL FAILS (2250 PSIA 550 F)

*Figure number gives isometric corresponding to leakoff line L/D

When all three seals fail the primary leakage path is through the open no. 1, 2 and 3 seals and out the top of the pump casing. This flow path offers less resistance to flow than the alternate leak paths; the no. 1, 2 and 3 leakoff lines. The controlling resistance for this case is the labyrinth seal. ETEC predicts flow will choke at this point and therefore the downstream flow path will not affect the leakage rate.

3.3.2 RCS Cooldown and Depressurization

No attempt was made to calculate accumulated leakage versus time for the case in which all three seals fail. For this case the initial leak rate is more than 20 times greater than the leak rate when all seals function. The RCS will drain and depressurize faster when all three seals fail. Because of the large discrepancy between the leakage rates for the two cases, the error in using the RCS pressure and temperature histories render the results almost meaningless.

3.3.3 Sensitivity Studies

In the case in which all three seals fail, the dominant resistance factor is the labyrinth seal at the seal assembly inlet. A pressure drop of approximately 1235 psia is predicted across this seal. It is also predicted that flow will choke at this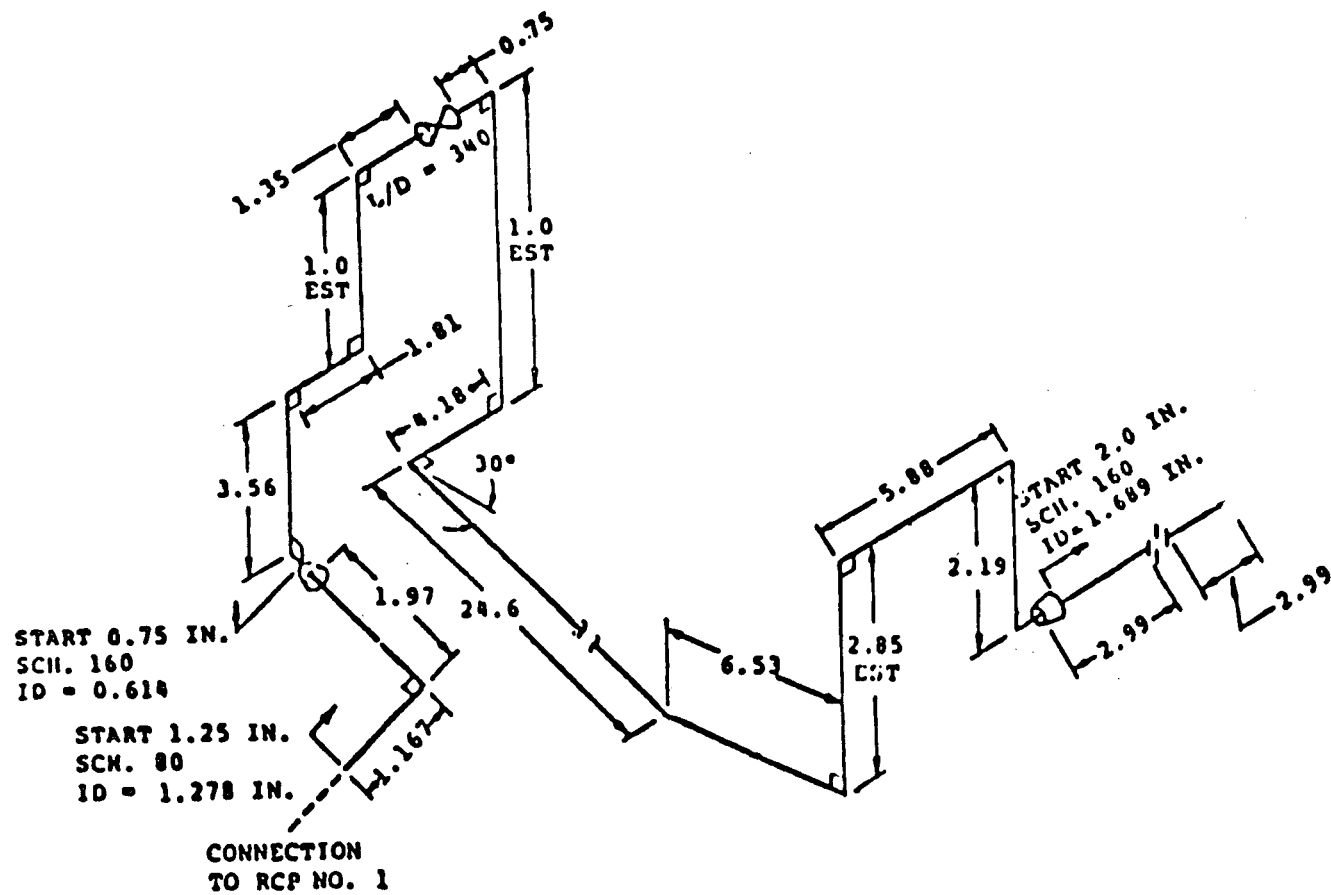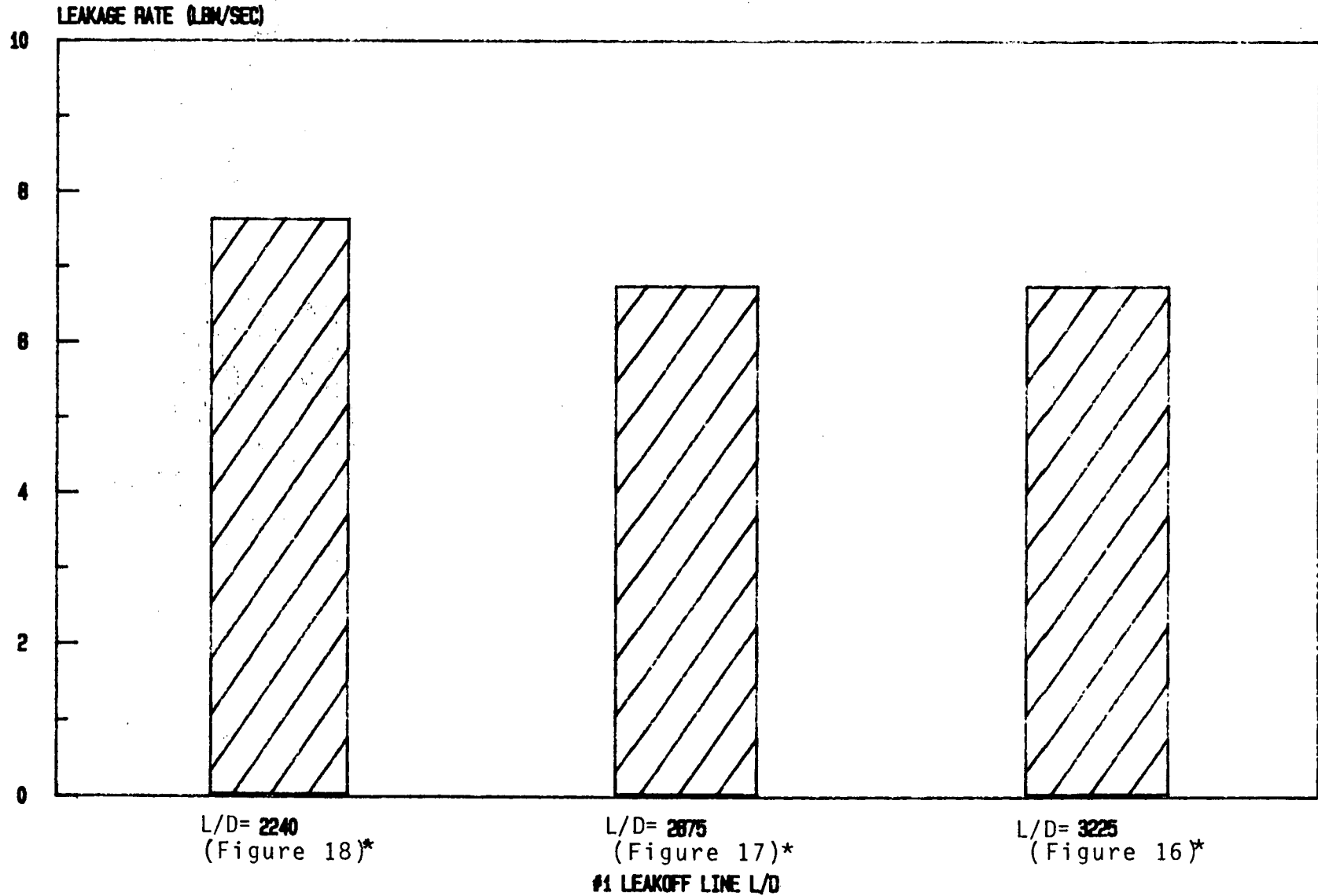 point. Three different resistance, or K, factors were used to determine the sensitivity of leakage rate to this paramater. The lowest K factor (4.7) corresponds to that used by Westinghouse in their analysis. During a meeting between ETEC and Westinghouse, ETEC was informed that this low K factor was calculated based only on entrance and exit losses. The highest K factor (15.4) corresponds to the resistance factor reported by Westinghouse during a February 28, 1984 meeting on RCP seals. The third K factor (10.0) is an average of the two values. Figure 20 illustrates the variation in leakage rate with labyrinth seal K factor. All cases were run with reactor coolant pressure of 2250 psia and temperature of 550°F. As shown on Figure 20, a lower K factor results in a larger leak rate. Therefore, ETEC chose to use the K factor of 4.7 for their analysis as it gave the most conservative results.

To verify that the ETEC assumption was conservative, that is that the true labyrinth seal K factor is not less than 4.7, ETEC made an independent calculation of the labyrinth seal K factor based on proprietary drawings of the Westinghouse RCP. The labyrinth was approximated as a small annulus between the shaft outside diameter and the inner edge of the labyrinth seal. The calculated resistance factor was slightly greater than 4.7, indicating that the K factor of 4.7 used in the ETEC and Westinghouse analyses is less than the true labyrinth seal resistance factor. Thus, the leak rates calculated by ETEC and Westinghouse for the case in which all seals fail are conservative. Since labyrinth seal resistance can not be accurately calculated, the recommendations (Section 4) include testing to determine the labyrinth seal hydraulic resistance .

FIGURE 20
LEAKAGE VS. LABYRINTH SEAL RESISTANCE
ALL SEALS FAIL (2250 PSIA 550 F)
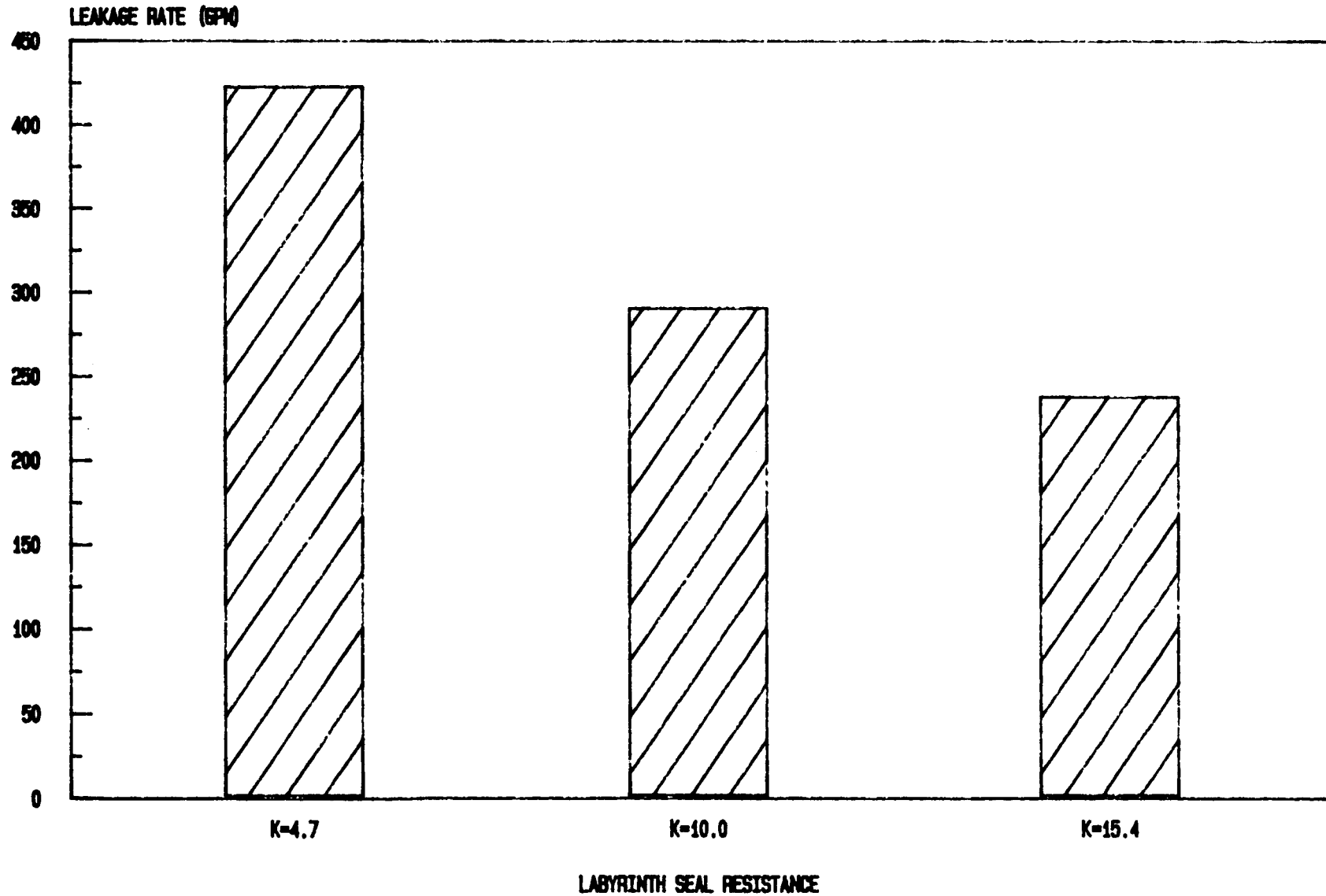
3.4 O-RING AND CHANNEL SEALS

As a part of the structural analysis of the no. 1 seal assembly, the change in O-ring and channel seal gland gap size was evaluated. No attempt was made to
.e
determine whether these changes would cause failure or blowout of the O-rings and channel seals. Table 5 and Figure 21 present the results of this work.

As part of the structural analysis to determine the response of the seals to station blackout conditions, the ANSYS model was used to generate isotherms across the seals. The isotherm plot of the no. 2 seal for 518°F upstream and 212°F downstream is presented as Figure 22. ETEC used the isotherm plot to check the O-ring and channel seal temperatures predicted by Westinghouse, Ref. 1. Most of the Westinghouse values appeared reasonable. However, for the channel seal circled on Figure 22, the 240°F temperature reported by Westinghouse seemed low. From Figure 22, the average temperature for this channel seal appears to be closer to 340°F.

3.5 UNCERTAINTY

In addition to the effect of the parameters examined in the sensitivity studies, Sections 3.1.3, 3.2.3 and 3.3.3, the analytical correlations also must be considered. The Dukler two-phase flow correlations have a reported uncertainty of 20%, reference 2. However, experience with other two-phase correlations indicate that this uncertainity is probably low. A 50% uncertainty in the calculated flow rates appears more reasonable. The uncertainty in flow correlations is much greater than that which results from machining tolerances or leakoff line configuration.

TABLE 5: O-RING AND CHANNEL SEAL GLAND GAP CHANGES --1 SEAL ASSEMBLY

O-Ring or Channel Seal	Change Due to ΔP (2200 psia to 30 psia)	Change Due to ΔT (550°F to 518°F)
1-1	.56 mils closed	.48 mils open
1-2	.48 mils closed	.91 mils open
1-3	.16 mils closed	.79 mils open
1-4	.15 mils open	.73 mils open
1-5	.02 mils open	.77 mils open
1-6	.18 mils closed	.97 mils open
1-7	.4 mils closed	.84 mils open
1-8	.18 mils open	.92 mils open

FIGURE 21:
#1 SEAL ASSEMBLY: O-RING AND CHANNEL SEALS

FIGURE 22:
ISOTHERM MAP OF THE #2 SEAL

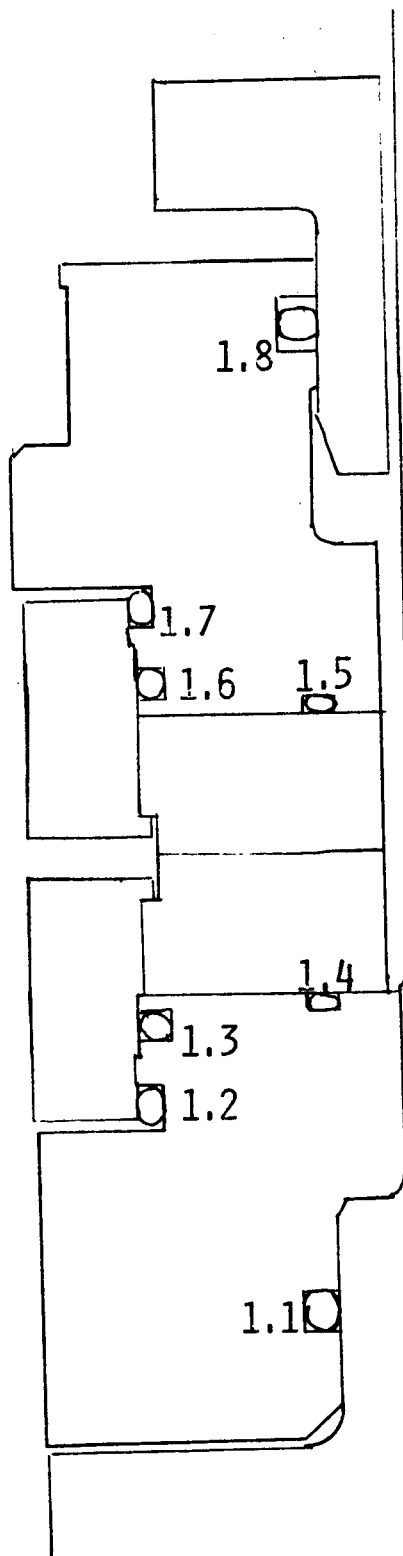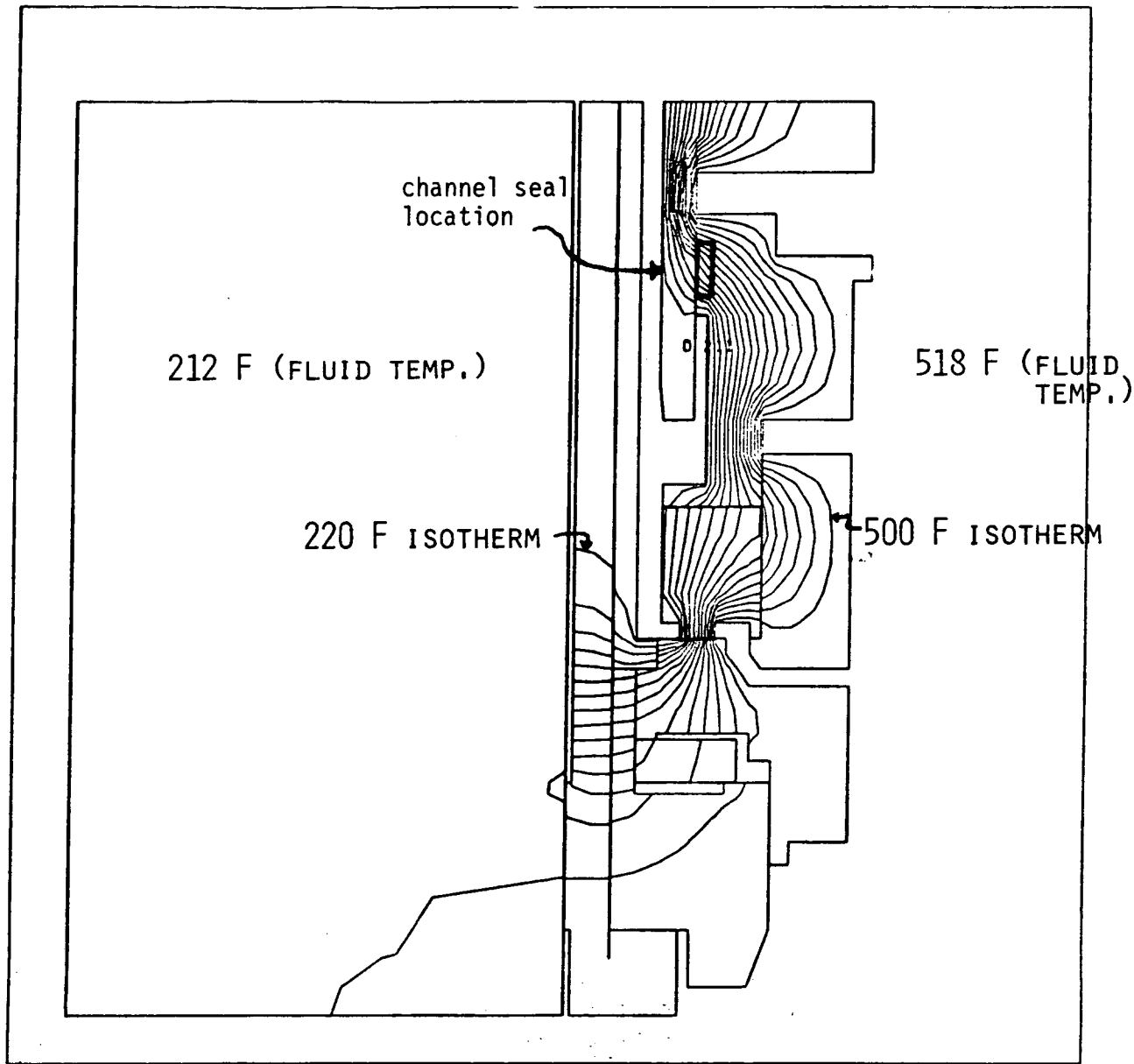channel seal
location

212 F (FLUID TEMP.)

518 F (FLUID TEMP.)

220 F ISOTHERM

500 F ISOTHERM

4.0 CONCLUSIONS AND RECOMMEDATIONS

The independent ETEC analysis of leakage through the Westinghouse RCP seal assembly confirmed the results of a previous Westinghouse analysis. The results of the two analyses are compared in Table 7.

ETEC evaluated the Westinghouse modeling assumptions and concluded they were reasonable or gave conservative results. The major difference between ETEC and Westinghouse seal models dealt with two-phase flow modeling assumptions. ETEC's constant slip two-phase flow assumption is more realistic and also less conservative than Westinghouse's homogenous flow representation and therefore, ETEC predicted leak rates are generally less than Westinghouse values.

The scope of the ETEC study did not include predicting RCS pressure and temperature conditions after initial station blackout. ETEC used the RCS conditions predicted by Westinghouse for a 'typical four loop plant', see reference 1. Westinghouse predicted RCS conditions based on two assumptions: (1) operators initiated reactor cooldown procedures and (2) operators failed to initiate cooldown procedures. Operator initiated cooldown of the RCS significantly reduced accumulated leakage during station blackout, see Figures 10 and 11. The Westinghouse calculations of RCS cooldown, presented in reference 1, were reviewed by ETEC and appear reasonable. However, since cooldown has a significant impact on accumulated leakage, these calculations should be verified.

The following recommendations are made regarding future investigation.

1. Determine Experimentally the Probability of Seal Failure

    Analysis indicates that seal failure is an important factor in determining the time required to uncover the core. The probability of seal failure cannot be accurately determined by analysis, therefore the probability of seal failure should be determined experimentally.

2. Evaluate the Effect of Boil-Off Downstream of No. 2 Seal

    ETEC and Westinghouse analyses predict that the temperature gradient across the No. 2 seal will cause it to rotate closed during a station blackout. However, if all the water downstream of the no. 2 seal flashes to steam and "boils-off," the heat sink provided by the water will be lost and the temperature gradient across the no. 2 seal will decrease. The predicted leakage rate is close to the boil-off rate, therefore an analysis of the no. 2 seal, assuming boil-off, would provide valuable information on seal stability and closure.

3. Determine Experimentally the Labyrinth Seal Flow Resistance

    The dominant resistance for the case when all three seals fail is the labyrinth seal located between the impeller and the first stage seal. To identify the worst case leakage rate when all three seals fail open, the resistance factor for the labyrinth sould be determined experimentally.

TABLE 6:   LEAKAGE RATES AT INITIAL REACTOR COOLANT
           SYSTEM CONDITIONS (2250 psia, 550°F)

| Seal | | | ETEC | | Westinghouse | |
#1	#2	#3	(lbm/sec)	(gpm)	(lbm/sec)	(gpm)
FN	FN	FN	2.05	19.6	2.20	21.1
FAIL	FN	FN	6.76	64.7	7.90	75.6
FAIL	FAIL	FAIL	44.1	422	50.2	480

FN = Function
FAIL = Fail Open

4. Evaluate Time to Uncover Core for Specific Plants

An important result of the leakage rate analysis is the time required
to uncover the core. This time should be determined for specific
plants based on their individual system thermal hydraulic response
characteristics and hardware designs.

5. Evaluate No. 2 Seal Leakage with Failure of No. 1 Seal

Analysis indicates that the no. 2 seal will rotate closed during a
station blackout due to thermal gradients across the seal. This
behavior limits the overall leak rate, especially if the no. 1 seal
fails open. As presented in section 3.2.1, the calculated leak rate
past the no. 2 seal when it rotates closed is on the order of .1 gpm.
However, analysis can not verify whether the no. 2 seal can withstand
full system pressure and high temperature gradients for an extended
time period. Therefore, testing of the no. 2 seal subjected to these
conditions--preferrably during a test of a full scale seal assembly--
is recommended. If testing of a RCP seal assembly is not conducted,
then tests of the O-rings and channel seals, under pressure and
temperature conditions which would exist if the no. 1 seal failed,
should be considered.

REFERENCES

1. Westinghouse Owners Group Report, WCAP 10541, "Reactor Coolant Pump Seal Performance Following a Loss of all AC Power", April 1984.

2. "Chemical Engineering Aspects of Two-Phase Flow", A. E. DeGance and R. W. Atherton, Chemical Engineering Vol. 77, March 23, 1970 - February 22, 1971.

3. ANSYS, Swanson Analysis Systems, Inc., Rev. 4.0, 1982.

APPENDIX A
SEAL FORCE BALANCE MODEL

1.0 RADIAL FLOW BETWEEN CONVERGING/DIVERGING ANNULAR PLATES (GENERAL SOLUTION)
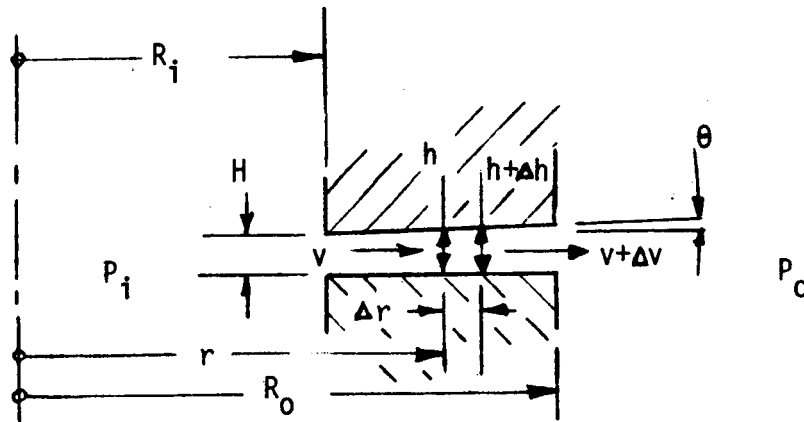
d_{eq}	Equivalent diameter, local [= 2 h]	in.
f	Friction factor, local	-
g	Gravitational constant	in/sec^2
h	Gap between plates, local	in.
H	Gap between plates at inner radius	in.
K	Pressure loss factor (entrance or exit)	-
p	Pressure, local	lb/in^2
P	Pressure, inlet or outlet	lb/in^2
Q	Volumetric flowrate	in^3/sec
r	Radius, local	in.
R	Radius, inlet or outlet	in.
Re	Reynolds number	-
v	Velocity, local	in/sec
V	Velocity, inlet or outlet	in/sec
w	Fluid specific weight	lb/in^3
Δ	Denotes change	
θ	Included angle between plates	(radians)
μ	Absolute viscosity of fluid	$lb \cdot sec/in^2$

Subscripts:
i	Inner Radius of Seal Face
o	Outer Radius of Seal Face
1	Inner Radius of Seal Face Downs-ream of Chamfer
2	Radius of Seal Face at Which Included Angle
3	Outer Radius of Seal Face Upstream of Chamfer
4	Radius at Which Sliding O-Ring is Located

1.1 GENERAL SOLUTION

A. Local gap between plates (h)

$h = H + (r - R_i)\theta$ Where the small angle approximation
$(\theta = \tan\theta)$ is assumed

B. Local velocity between plates (v)

$v = Q/2\pi rh$

$$v = \frac{Q}{2\pi r[H+(r-R_i)\theta]}$$

C. Local pressure gradient ($\Delta p/\Delta r$)

$$\frac{\Delta p}{\Delta r} = \left(\frac{\Delta p}{\Delta r}\right)_1 + \left(\frac{\Delta p}{\Delta r}\right)_2$$

Where $(\Delta p/\Delta r)_1$ = Pressure gradient due to
fluid friction
$(\Delta p/\Delta r)_2$ = Pressure gradient due to
velocity change

C.1 Friction pressure gradient

$$\left(\frac{\Delta p}{\Delta r}\right)_1 = -w\frac{f}{D_{eq}} \cdot \frac{v^2}{2g}$$

For flow through parallel plates, $D_{eq} = 2h$

$$= \frac{-w\,f}{4gh}\left(\frac{Q}{2\pi rh}\right)^2$$

$$\frac{\Delta p}{\Delta r}_1 = -\frac{w\,f\,Q^2}{16\,\pi^2\,g}\left\{\frac{1}{r^2[H+(r-R_i)\theta]^3}\right\}$$

NOTE: The above equation indicates a negative pressure gradient (pressure decreases with radius) based on the initial assumption of radial outflow. For radial inflow, the sign of the gradient is reversed.

C.2 Velocity pressure gradient

$$\left(\frac{\Delta p}{\Delta r}\right)_2 = -\frac{w}{\Delta r} \cdot \Delta\left(\frac{v^2}{2g}\right) = -\frac{w}{\Delta r} \cdot \frac{(v + \Delta v)^2 - (v)^2}{2g}$$

$$\left(\frac{\Delta p}{\Delta r}\right)_2 = -\frac{w}{g}\left(\frac{v \cdot \Delta v}{\Delta r}\right) \text{ where the term } (\Delta v)^2 \text{ in the expansion is assumed to be negligable}$$

$$v = \frac{Q}{2\pi r h}, \quad \Delta v = \Delta\left(\frac{Q}{2\pi r h}\right)$$

$$\cong \frac{-Q \cdot \Delta(2\pi r h)}{(2\pi r h)^2}$$

$$= -\frac{Q \cdot 2\pi(r\Delta h + h\Delta r)}{(2\pi r h)^2}$$

$$\Delta v = -\frac{Q(r\Delta h + h\Delta r)}{2\pi r^2 h^2}$$

$$\text{but: } h = \left[H + (r - R_i)\theta\right]$$

$$h = \theta \cdot \Delta r$$

$$\Delta v = -\frac{Q(r\theta \cdot \Delta r + h \cdot \Delta r)}{2\pi r^2 h^2}$$

$$\left(\frac{\Delta p}{\Delta r}\right)_2 = -\frac{w}{g}\left(\frac{Q}{2\pi r h}\right)\left[-\frac{Q(r\theta \cdot \Delta r + h \cdot \Delta r)}{2\pi r^2 h^2 \cdot \Delta r}\right]$$

$$= \frac{wQ^2}{4\pi^2 g}\left[\frac{\theta}{r^2 h^3} + \frac{1}{r^3 h^2}\right]$$

$$\left(\frac{\Delta p}{\Delta r}\right)_2 = \frac{wQ^2}{4\pi^2 g}\left[\frac{\theta}{r^2\left[H + (r - R_i)\theta\right]^3} + \frac{1}{r^3\left[H + (r - R_i)\theta\right]^2}\right]$$

C.3 Total pressure gradient $= (\Delta p/\Delta r)_1 + (\Delta p/\Delta r)_2$

$$\left(\frac{\Delta p}{\Delta r}\right) = \frac{w}{g} \left(\frac{Q}{2\pi}\right)^2 \frac{(\theta + .25f)}{r^2 \left[H + (r - R_i)\theta\right]^3} + \frac{1}{r^3 \left[H + (r - R_i)\theta\right]^2}$$

D. Entrance and exit loss factors (K_i and K_o)

Entrance and exit pressure losses are taken as proportional to the local (entrance or exit) velocity head:

$$\Delta p_{loss} = \frac{w}{g} K \frac{v^2}{2g}$$

D.1 Entrance loss factor (K_o)

Considering the inlet edge configuration, chamfered or rounded, the inlet loss factor is assumed at one-half the maximum value (0.5) given for a sudden contraction.

$$K_o = .5 \times 0.5$$

$$K_o = 0.25$$
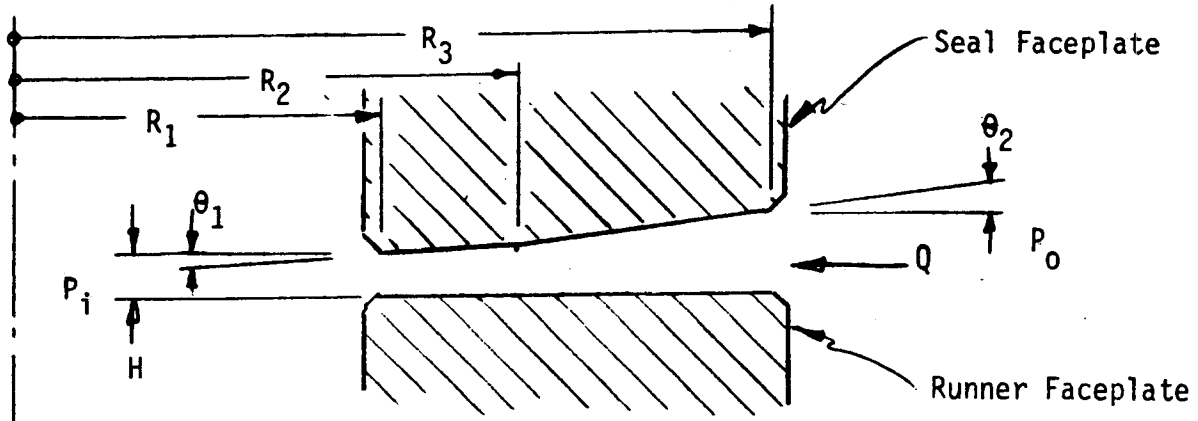
D.2 Exit loss factor (K_i)

Under similar consideration, the exit loss factor is assumed at one-half the maximum value (1.0) given for a sudden expansion.

$$K_i = \frac{1}{2} \times 1.0$$

$$K_i = 0.5$$

Maximum loss factors are from "Flow of Fluids through Valves, Fittings, and Pipes," by Crane Co.

2.0 APPLICATION OF GENERAL SOLUTION TO NO. 1 PRIMARY SEAL

Pressure loss through seal faces

$$\Delta p = \frac{w}{g} K_o \left\{ \frac{W/w}{2\pi R_3 \left[H + (R_2 - R_1)\theta_1 + (R_3 - R_2)\theta_2 \right]} \right\}^2$$

$$+ \frac{w}{g} \left(\frac{W/w}{2\pi} \right)^2 \sum_{r=R_3}^{r=R_2} \left\{ \frac{(\theta_2 + .25f)}{r^2 \left[H + (r - R_2)\theta_2 + (R_2 - R_1)\theta_1 \right]^3} \right.$$

$$\left. + \frac{1}{r^3 \left[H + (r - R_2)\theta_2 + (R_2 - R_1)\theta_1 \right]^3} \right\} \Delta r$$

$$+ \frac{w}{g} \left(\frac{W/w}{2\pi} \right)^2 \sum_{r=R_2}^{r=R_1} \left\{ \frac{(\theta_1 + .25f)}{r^2 \left[H + (r - R_1)\theta_1 \right]^3} + \frac{1}{r^3 \left[H + (r - R_1)\theta_1 \right]^3} \right\} \Delta r$$

$$+ \frac{w}{g} K_i \left(\frac{W/w}{2\pi R_1 H} \right)^2$$

The above difference equation is summed, using a digital computer and Δr increments of 0.01 inch. At each step the local Reynolds number and absolute pressure are evaluated to determine the appropriate friction factor.

3.0 SEAL FORCE BALANCE
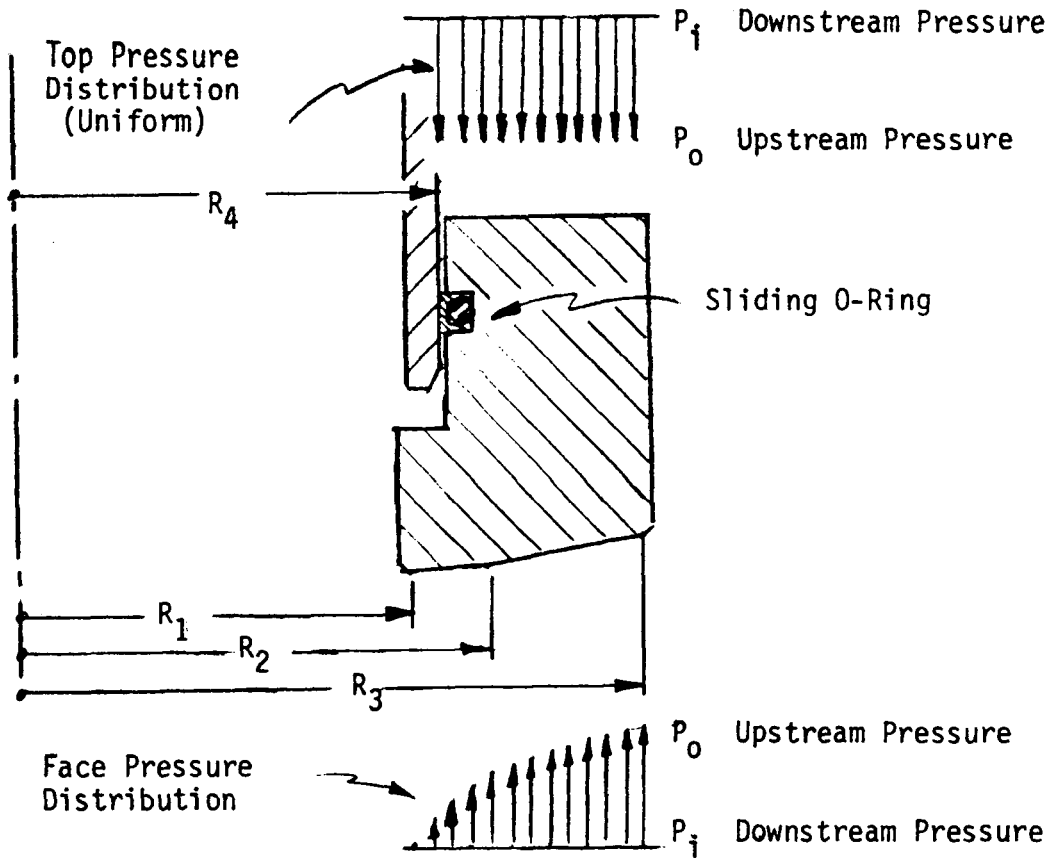
The axial position of the seal (and thus the gap) is stabilized when the force resulting from the face pressure distribution is equal to the force resulting from the top pressure distribution (weight neglected):

$$(P_o - P_i)\pi(R_3^2 - R_4^2) = 2\pi \sum_{r=R_3}^{r=R_1} r\,(p - P_i)\Delta r$$

The previous equation is used to generate a pressure function, $p = p(r)$. The downstream pressure, P_i, is used as the zero reference pressure, i.e., the lowest pressure seen by the seal. The right side of the above equation sums the pressure forces acting on the seal face. Designating this sum as F_f and evaluating the area, $\pi(R_3^2 - R_4^2)$, from Westinghouse drawings, the above equation may be rewritten as:

$$\frac{F_f}{(P_o - P_i)} = 39.728$$

which is used as the force balance criterion. Seal performance
(pressure drop, temperature difference, and face force/pressure
difference ratio) is calculated for various inputs (flow,
pressure, temperature) and operation assumptions (minimum gap
and face angle). Calculations are performed using logic as shown
in the flow diagram in Figure A-2.

Operating conditions which match the force balance criterion
on the previous page are further evaluated against structural
data, generated to predict seal face rotation (angle between
seal faces) resulting from temperature and pressure differences.

Figure A-1, shows performance predictions
for operating conditions which meet the axial force balance
criterion, assuming single-phase flow with 130^OF water. Injection
water at 130^OF is supplied to the seal assembly during normal
pump operation. This parametric data was calculated to provide
initial information on seal characteristics, to be used as guidance
for subsequent detail analyses.

FIGURE A-1: PARAMETRIC PERFORMANCE CALCULATED FOR
BALANCED AXIAL FORCES
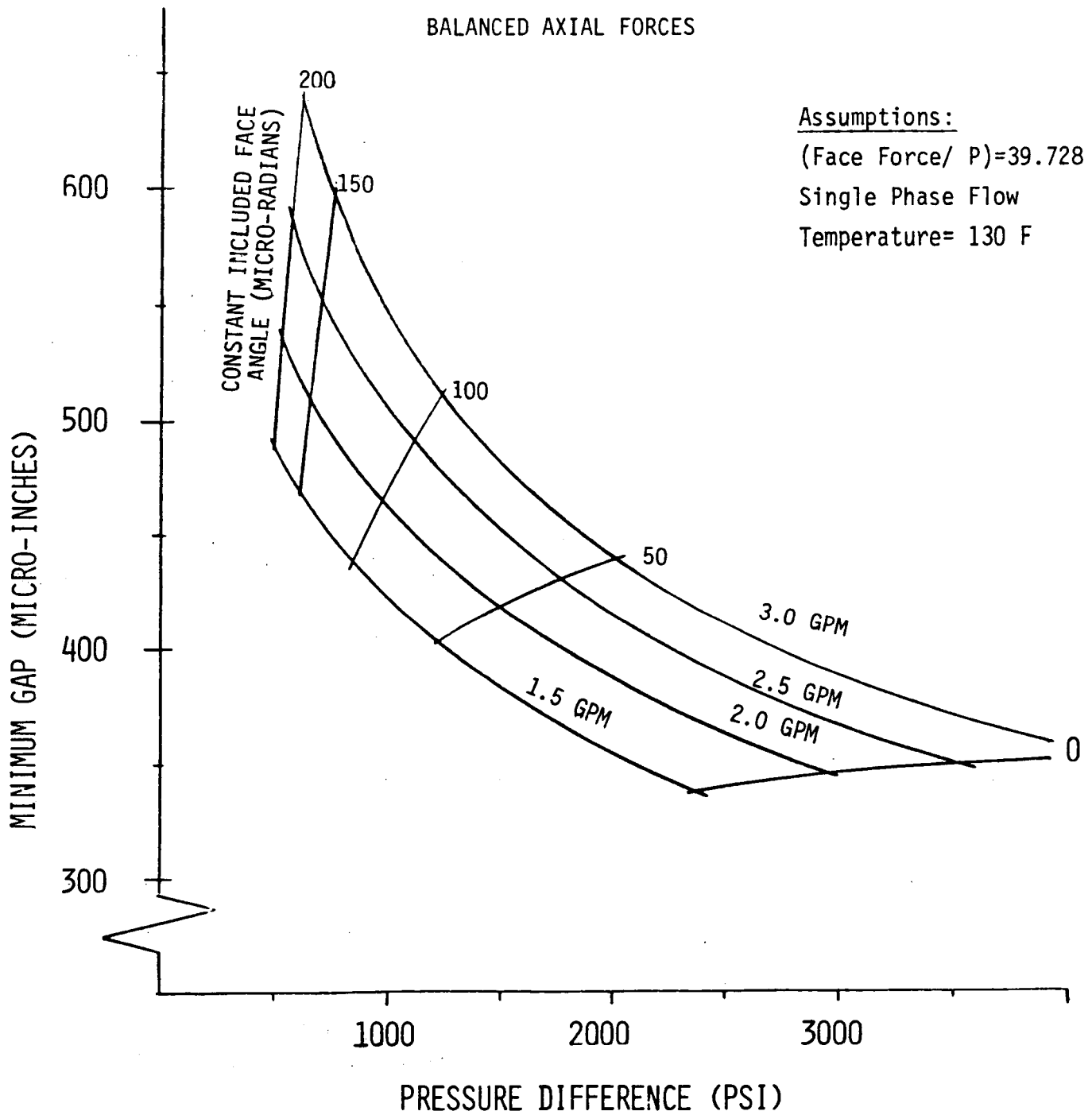
FIGURE A-2

FLOW DIAGRAM FOR SEAL PERFORMANCE CALCULATION

IN — Fluid Data (temp, pres) → Fluid Properties (w,u,vapor pres) → AND

Oper. Assumptions (W,H,θ_1) — IN

IN — Seal Geometry (R_1,R_2,R_3,R_4) → AND ← OR ← Increment Operating Assumptions (W,H,θ_1)

AND → Inlet Pres. Loss → OR → Local Pressure → Reynolds No. & Friction Factor → OR → Pressure Gradient → Decrease Radius → Position

Change to 2 Phase Formula Include Temp. In Output

Off Inner Edge

Position — Inner Portion / Outer Portion → OR → Local Pressure → Below Vapor Pressure

Change Face Angle, θ

Yes / No

Sum Pressure Diff.

OUT — Printout Local Press. ← Option ←

Sum Face Force

OUT — Printout Local Face Force ← Option ←

Outlet Pressure Loss → Outlet Pressure (and Temp.)

Printout Results (P_i,T_i, P,F_f,P_o,T_0, F_f/P) — OUT

54

APPENDIX B

FLOW MODEL

The flow model calculates the pressure drop due to single and two
phase flow through the constant area flow paths, such as the
labyrinth seal, cartridge seal internals, and the leakoff lines.
The two phase flow correlations used are based on the Dukler
constant slip model.

1. DUKLER TWO PHASE FLOW CONSTANT SLIP MODEL

The Dukler constant slip, two phase flow model assumes that the
ratio of the phase velocities to the average velocity is constant
across a cross section. The model does not involve the use of
flow regimes, that is the pattern of two phase flow such as
bubble, slug, and mist.

1.1 Terms and Definitions

A   area (sq ft)

AC   acceleration loss factor

c   function of the derivative of steam density with respect to
        pressure which is used to calculate the acceleration loss
        factor

d   diameter (ft)

e   surface roughness factor (ft)

f   two phase friction factor

fo   single phase friction factor

g   gravitational constant (lbm ft/lbf sec sec)

GG   superficial mass flux of the vapor phase (lbm/sq ft)

GL   superficial mass flux of the liquid phase (lbm/sq ft)

GT   superficial mass flux of the two phase fluid (lbm/sq ft)

K   parameter used in calculating inplace liquid holdup, a
        function of delta

P   pressure (lbf/sq ft)

Re   Reynolds number

Rg   in place vapor holdup, equal to (1-Rl)

Rl   in place liquid holdup calculated using Hughmark's
         correlation

si   parameter used to calculate two phase friction factor,
         function of flowing volume liquid holdup

tau   partial derivative of pressure with respect to length due to
         friction

V   two phase velocity (ft/sec)

Vsg   superficial velocity of the vapor phase (ft/sec)

Vsl   superficial velocity of the liquid phase (ft/sec)

W   mass flowrate (lbm/sec)

X   mass fraction of vapor

α correction factor for two phase friction factor

δ parameter used to calculate in place liquid
         holdup, a function of Reynolds and Froudes
         numbers.

λ flowing volume holdup of the liquid

μ two phase viscosity (lbm/sec ft)

μg vapor phase viscosity (lbm/sec ft)

μl liquid phase viscosity (lbm/sec ft)

ρ two phase density (lbm/cu ft)

ρg vapor phase density (lbm/cu ft)

ρl liquid phase density (lbm/cu ft)

ρns two phase density calculated assuming homogeneous two phase
         flow (lbm/cu ft)

1.2 Basic Equations

$$GL = (1-X)*W/A$$

$$GG = X*W/A$$

$$GT = GL + GG$$

$$Vsl = GL/\rho l$$

$$Vsg = GG/\rho g$$

$$V = Vsl + Vsg$$

$$\lambda = Vsl/V$$

$$\mu = \mu l * \lambda + \mu g * (1-\lambda).$$

$$\rho ns = \rho l * \quad + \rho g * (1-\lambda)$$

$$\rho = \frac{\rho l * \lambda}{R1} + \frac{\rho g * (1-\lambda)}{(1-R1)}$$

$$Re = (\rho * GT * d)/(\mu * \rho ns)$$

1.3 Two Phase Friction Factor

$$f = \alpha * fo$$

$$\alpha = 1 - \log(\lambda/si)$$

$$si = 1.281 + .478 * \log\lambda + .444 * (\log\lambda)^2 + .094 * (\log\lambda)^3 + .00843 * (\log\lambda)^4$$

$$fo = .25 * \left[\log(e/(3.7 * d) + 2.51/(Re * fo^{.5})) \right]^{-2}$$

$$tau = f * \rho * GT /(2 * g * d * \rho ns)$$

1.4 Acceleration Loss Factor

$$AC = \frac{GL * Vsl/R1 + (GG * Vsg/Rg) * (c-R1/Rg)}{P * g}$$

1.5 Hughmark's Correlation for In Place Liquid Holdup

$$R1 = 1-(1-\lambda) * K$$

$$\delta < 10$$
$$K = .1637 - .3104 * \delta + .3525 * \delta^2 - .001366 * \delta^3$$

$$\delta > 10$$
$$K = -.75545 - .003585 * \delta + .00001436 * \delta^2$$

$$\delta = \frac{.642 * V^{.5} * GT^{.1667} * d^{.0427}}{Vsl^{.25} * (R1 * (\mu l - \mu g) + \mu g)^{.1667}}$$

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13. ABSTRACT (200 words or less)

An independent analysis was performed to determine seal leakage rates for the Westinghouse Reactor Coolant Pump (RCP) during a postulated station blackout resulting from loss of ac electric power. The analysis confirmed Westinghouse calculations on RCP seal performance for the three conditions investigated: (1) all three seals function, (2) No. 1 seal fails open while Nos. 2 and 3 seals function, and (3) all three seals fail open.

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