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# Analysis on the Circumference Wall Temperature in a Long Horizontal Pipe with Thermal Stratification

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ABSTRACT

The One-dimensional fin model is used to analyze the angular wall temperature variation of long horizontal lines, where stratification could result in top-to-bottom differences in wall temperatures. The top and bottom sections are treated separately and coupled by boundary conditions. The thermal stratification analysis is focused on the effects of the heat transfer rates at the pipe surface. The results show that the heat transfer rate at the pipe surface is the controlling parameter which reduce significantly the temperature difference in pipe circumferential direction. The one-dimensional fin modelling analysis results are verified by comparison with the operating PWR test data. The circumferential temperatures of pipe calculated by one-dimensional fin modelling agree well with the PWR plant test data..

## 1. INTRODUCTION

Stratified flow is a condition in which hotter fluid flows over a colder region of fluid. The fluid is divided into two distinct layers with the denser fluid occupying the lowest position. The stability of the flow stratification is dependent on the differences in temperature and velocity of the flow. The Richardson number is a measure of the stability of a stratified flow. It is a dimensionless parameter representing the ratio of buoyancy force to inertia force. If the Richardson number exceeds unity the stratification may be stable. If the Richardson number is less than unity, the stratification is unstable and may become unstratified. Greater temperature differences and lower velocities result in more stable stratification conditions. Differences in the fluid temperatures and heat transfer coefficients will lead to temperature gradients in the pipe wall in both the circumferential and radial directions. Industry experience indicates that flow stratification occurs primarily during hot standby conditions, heatup and cooldown conditions. The effect of the stratification and the resulting circumferential temperature gradient is that the pipe is subject to high alternating thermal stresses. Thermal stratification in light water nuclear reactor(LWR) piping systems has recently been considered the subject of safety concerns(US NRC Bulletins No. 88-08[1] and 88-11, 1988[2]). The phenomenon of thermal stratification plays an important role in the piping integrity, because the significant thermal stresses induce the failure and the unexpected motions in piping lines of PWR plant. Bejan, A. and Tien.(1978)[6] have performed analytically model the thermal-hydraulic aspects of stratified flow. Hong, S. W. (1977)[7] has studied numerically natural circulation in horizontal pipe. Shiralkar and Tien(1981)[5] performed the high Rayleigh number convection in shallow enclosures with different end temperatures. Recently, Park and Youm(1995)[3] performed the unsteady 2-D numerical analysis for thermal stratification in a surge line of PWR plant. Lubin and Kim(1994)[4] predicted wall temperatures in long horizontal lines with different end temperatures undergoing natural convection at very high Rayleigh number. In this study, the analytical equation which can calculate the circumferential wall temperatures of long horizontal pipe with thermal stratification is developed by using the one-dimensional fin model. A comparison is made between the analysis results the operating PWR test data.

## 2. MODEL FORMULATION

The surge line connects the pressurizer and the hot leg of the reactor coolant system of PWR(Fig. 1). The line runs from a pressurizer bottom nozzle are oriented in a vertical up position on the hot leg of the RCS. Typical lines have one or more long horizontal sections between vertical rises. Surge line is 12 inch nominal diameter and schedule of 160. In normal operation, top to bottom temperatures difference of horizontal surge line is caused by either an insurge or an outsurge from pressurizer. These fluid surges are the result of a number of plant operations, such as pressurizer heaters, spray flow, reactor coolant pump operation, and the mismatch of charging and letdown flow. The surge line is insulated to prevent heat loss from the surge line to containment during all operating mode. Therefore, the ambient heat transfer coefficient($h_\infty$) of the surge line is about 1.0 Btu/hr $ft^2$ °F. Also, the material of surge line is the SA312TP347 stainless steel. The pressurizer temperature, hot leg temperature, Richardson number and surge line flowrate are listed in Table 1.

Table 1 : Analysis Conditions

| | Hot Temp (F) | PZR Temp (F) | SL Flow (gpm) | Outside Temp(F) | Ri No | Re No |
|---|---|---|---|---|---|---|
| Heatup | 441.68 | 124 | 22.4 | 80 | 595 | 6200 |

$Ri = \Delta\rho \, g \, D / \rho \, V^2$

where : $\rho$ = density

$g$ = acceleration of gravity

$D$ = diameter of pipe

$V$ = fluid velocity

Considering the thermal resistances of the inside heat transfer, the wall, and the insulation, the wall temperature is almost uniform in the radial direction. Therefore, one-dimensional model is proposed to predict the variation of wall temperature, $T_w$, in angular direction. The one-dimensional fin model are applied to the stratified flow. As shown in Fig. 2, the stratified flow model consists of hot fluid(pressurizer temperature) in the upper portion of the pipe and cold fluid(hot leg temperature) in the lower half with an interface between two layer. It is assumed that flow occurred only in the upper portion of the pipe during an outsurge. Conversely, flow is assumed to occur only in the lower portion of the pipe during an insurge. The pipe is modeled as a rolled up fin(Fig.2), with distance along the fin, $x = R\theta$. The major assumptions are made to solve this fin model as follows;

(a) Heat conduction in the axial and radial direction is negligible; $dT_w/dx = dT_w/dr = 0$.

(b) The hot and cold fluid can be represented by two homogeneous layers, the top at $T_2$ and the lower at $T_1$, which are assumed to be separated by a flat interface at an interface locations, $\theta = \theta_I$. In this model, $\theta_I$ is $\pi/2$ since the top and bottom flows are similar.

(c) The heat transfer coefficients between both layers and the inner wall are uniform within each layer, and the temperature distribution in the wall is assumed symmetric.

The energy balance is made for heat transfer by convection from the fluid to the wall and the wall to the ambient and conduction along the wall :

$$k_s\, a \frac{d^2T_w}{dx^2} - h_i\,(T_i - T_w) + h_\infty(T_w - T_\infty) = 0. \qquad (1)$$

where i is the lower(i=1) or upper(i=2) layer and $T_i$ is average inside fluid temperature. The heat loss plus gain terms can be expanded and equated to an equivalent ambient temperature $T_{\infty i}$ ;

$$h_i\,(T_i - T_w) - h_\infty(\,T_w - T_\infty) = (h_i + h_\infty)(T_{\infty i} - T_w)$$

$$T_{\infty i} = \frac{(h_i\,T_i + h_\infty T_\infty)}{(\,h_i + h_\infty)} \qquad (2)$$

The equation can be put into standard equation form as;

$$k_s\, a \frac{d^2T_w}{dx^2} - N_i\,(T_w - T_{\infty i}\,) = 0. \qquad (3)$$

where,

$$N_i = \frac{(h_i + h_\infty)}{k_s\, a}$$

Equation(3) is solved by standard means subject to boundary conditions of no heat loss at the ends( $\theta = 0^\circ$, $180^\circ$) and that temperature and heat flux should be continuous at the interface location, $\theta_I$ ;

$$T_1 = T_2\,,$$

$$(dT_1/dx)_{L1} = (dT_2/dx)_{L2} \text{ (Fig. 2).} \qquad (4)$$

1) Lower layer; $0 < x_1 < R\theta_I$ ;

Therefore,

$$T_{w1} = \frac{T_{\infty 1} + (T_{\infty 1} - T_{\infty 2})\, N_2 \sinh(N_2 L_2) \cosh(N_1 x_1)}{N_2 \cosh(N_1 L_1) \sinh(N_2 L_2) + N_1 \cosh(N_2 L_2) \sinh(N_1 L_1)} \qquad (5)$$

2) Upper layer; $R\theta_I < x_2 < R(\pi - \theta_i)$ ;

Therefore,

$$T_{w2} = \frac{T_{\infty 2} + (T_{\infty 2} - T_{\infty 1})\, N_1 \sinh(N_1 L_1) \cosh(N_2 x_2)}{N_2 \cosh(N_1 L_1) \sinh(N_2 L_2) + N_1 \cosh(N_2 L_2) \sinh(N_1 L_1)} \tag{6}$$

This solution indicates that the top to bottom temperature differences approach the difference between the hot and cold fluid temperature with increasing values of heat transfer coefficients.

## 4. ANALYSIS

The upper($h_1$) and lower($h_2$) internal convection fluid heat transfer coefficients is calculated to obtain an outside wall temperature distribution. Two cases are selected for the analysis. Case 1 is a comparison between one-dimensional fin model and test data. Case 2 is an analysis for the variation of heat transfer rates of the pipe surface. The analysis results are compared with the measured values of an operating PWR plant. This comparison is given in Fig. 3. The analysis result shows good consistency with the plant measured data..

### 4-1 Calculation of fluid heat transfer coefficients

An outsurge(or insurge) flow is assumed to occur in the upper(or lower) region of the pipe model. The Reynolds number(Re) is calculated based on the fluid flowing in the pipe and the fluid being stationary.
If Reynolds number(Re) is greater than 2300, it is turbulent flow. Therefore, the Colburn equation is used to calculate the Nusselt number.

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.33} \tag{7}$$

From the definition of the Nusselt number the average convection heat transfer for the appropriate half of the pipe is as follows:

$$Nu = h_{ave.}\ D_h / k_s \tag{8}$$

where, $D_h$ is hydraulic diameter( = 0.611D), $k_s$ is wall thermal conductivity

$$h_{ave.} = Nu\ k_s\ /\ D_h$$

If Reynolds number(Re) is less than 2300, it is laminar flow. Therefore, for laminar flow,

$$Nu = 3.66 \tag{9}$$

### 4-2 Calculation of outside wall temperature distribution

To determine the pipe wall temperature distribution for the stratified flow model, one-dimensional fin model is developed. This model is described in section 3. The value of fluid heat transfer coefficients calculated in the section 4-1 are put into the Eq.(5) through(6) in order to calculate the pipe wall temperature distribution. The calculated wall temperatures are compared with the measured temperatures of an operating PWR plant.

## 5. RESULTS AND DISCUSSION

In order to identify the thermal stratification in the surge line, the horizontal surge line circumferential temperatures are measured during the PWR heatup testing. The thermocouples are located at five positions in the surge line piping circumferential direction. The $0^{\circ}$ location is the top location of horizontal pipe run. The measured temperatures are compared with the calculated temperature in order to verify developed one-dimensional fin modelling.

### 5-1 Case 1: A comparison between one-dimensional fin model and test data

This case is pressurizer outsurge. This outsurge is caused by the fact that letdown flow from the Reactor Coolant System greater than charging flow into the RCS. This causes pressurizer level to decrease which results in flow from the pressurizer through the surge line into the hot leg. The observed outsurge increases the pipe metal temperatures in the upper portion of the surge line horizontal pipe as compared to the lower portion of the pipe. The maximum temperature difference between the analytical fin model and testing data is about 50°F at the upper portion of pipe($0^{\circ}$ through $45^{\circ}$) which is a difference of about 15%. However, at the other portion except the upper portion of pipe, the analysis results are shown to agree well with the plant measured data..

### 5-2 Case 2: An analysis of the variation of heat transfer rates of the pipe surface.

The main purpose of analysis is to identify major parameters of the thermal stratification of the flow in a horizontal pipeline. This will suggest a feasible design change to prevent thermal stratification, and reduce the maximum temperature difference from top to bottom of pipe in the surge line of the PWR plant. Therefore, in this study, the analyses on the thermal stratification is focused on the effects of the ambient heat transfer coefficient at the pipe surface. The analysis is performed the variation of the ambient heat transfer coefficient of pipe. The selected ambient heat transfer coefficients are 10 Btu/hr $ft^2$ °F, 50 Btu/hr $ft^2$ °F, and 100 Btu/hr $ft^2$ °F, respectively. The angular wall temperature difference( from top to bottom) is plotted for the several heat transfer rates of the pipe wall in Fig. 4. It is known that the wall temperature difference between top and bottom of pipe reduces significantly because of the mixing effect by natural convection as the heat transfer coefficient increases. However, the stratification phenomenon still exists in the pipe line. The flat profile in the top portion is a result of the large convective heat transfer coefficient in that section. Likewise, the sharp temperature gradient in the bottom portion of the pipe is a result of the small convective heat transfer coefficient.

## 6. CONCLUSIONS

The results show that the ambient heat transfer coefficient at the pipe surface is the controlling parameter which is affected by the insulation design and environmental conditions of the surge line in the PWR plant. The circumferential wall temperatures calculated by the developed one-dimensional fin model are compared with the measured temperatures of a operating PWR plant. Considering the variation of the ambient heat transfer coefficient at the pipe surface, the temperatures difference from top to bottom of pipe line reduce significantly because of the mixing effect by natural convection as the heat transfer coefficient increases.

## NOMENCLATURE

a : thickness of pipe
h : heat transfer coefficient
$k_s$ : thermal conductivity
Nu : Nusselt number
Pr : Prandtl number

Re : Reynolds number
Ri : Richardson number
T : temperature
θ : angle

Subscripts
1 : upper portion
2 : lower portion
c : cold
h : hydraulic
f : fluid regions
I : interface
i : upper or lower portion
s : solid regions, surface regions
w : pipe wall
∞ : environment

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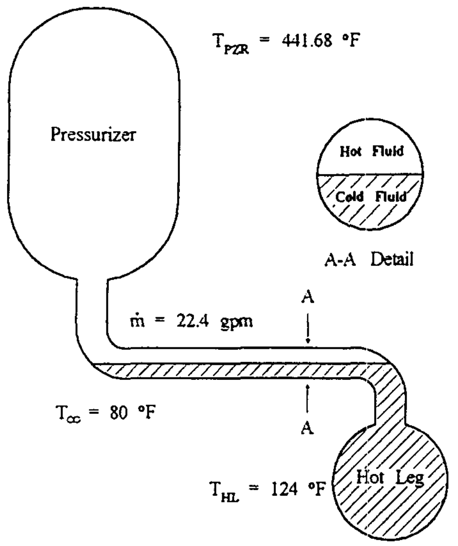


Fig. 1 Conditions of the Analysis Model.

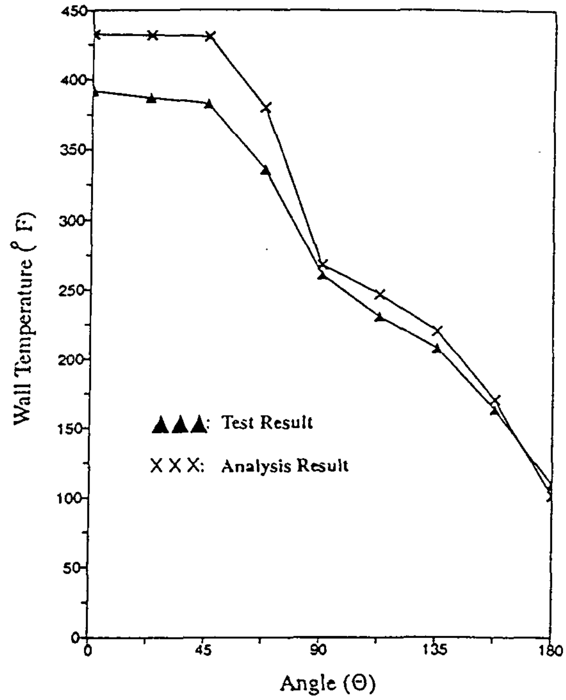


Fig. 3 Comparison with plant test data

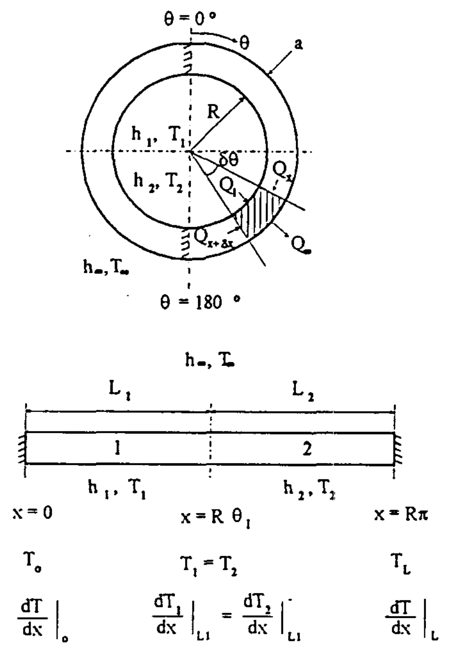


Fig. 2 The Schematic Diagram of One-Dimensional Fin Model.

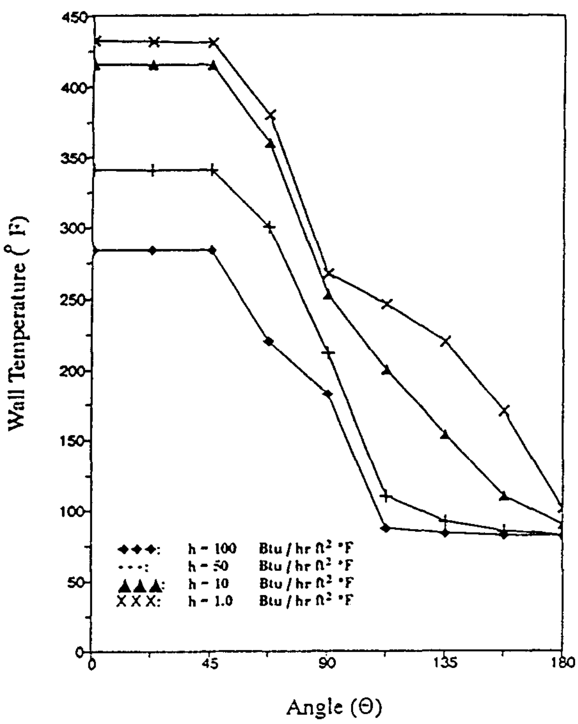


Fig. 4 The wall temperature for the variation of the heat transfer coefficient of the wall surface