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# NUMERICAL ANALYSIS FOR UNSTEADY THERMAL STRATIFIED FLOW WITH HEAT TRACING IN A HORIZONTAL CIRCULAR CYLINDER

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## ABSTRACT

A method to mitigate the thermal stratification flow of a horizontal pipe line is proposed by heating external bottom of the pipe with electrical heat tracing. Unsteady two dimensional model has been used to numerically investigate an effect of the external heating to the thermally stratified flow. The dimensionless governing equations are solved by using the control volume formulation and SIMPLE algorithm. Temperature distribution, streamline profile and Nusselt numbers of fluids and pipe walls with time are analyzed in case of externally heating condition. The numerical result of this study shows that the maximum dimensionless temperature difference between the hot and the cold sections of pipe inner wall is 0.424 at dimensionless time 1,500 and the thermal stratification phenomena is disappeared at about dimensionless time 9,000. This result means that external heat tracing can mitigate the thermal stratification phenomena by lessening $\Delta T_{ma}$ about 0.1 and shortening the dimensionless time about 132 in comparison with no external heat tracing.

## 1. INTRODUCTION

Thermal stratification phenomenon is reported to occur in the piping systems of nuclear power plants. It is usually observed in the pipe lines of the safety related systems and has been identified as a source of fatigue damage[1,2]. Temperature difference between the upper and lower fluids inside the thermally stratified pipe line results bending stresses to nuclear piping systems, such as horizontal section of the pressurizer surge line during insurge or outsurge operations. Thermal stratification of the surge line is caused during the heatup and cooldown operations for the largest temperature difference between the pressurizer and the hot leg of PWR primary system[3].

The propensity for stratification of a fluid in a horizontal pipe can be correlated to its Richardson number Ri which is the ratio of the gravitational buoyant force to inertia force acting on the fluid. If the number is greater than unity, the thermal stratification can be expected to occur. The buoyant force caused by the density difference between the hot and the cold fluid is proportional to the temperature difference. Therefore, the thermal stratification is likely to occur in low flow velocity and the high temperature difference[4].

In this study, a method to mitigate the thermal stratification of a horizontal pipe line is proposed by heating external bottom of the pipe with electrical heat tracing. Unsteady two dimensional model has been used to numerically investigate an effect of the external heating to the thermal stratification flow. The dimensionless governing equations are solved by using the control volume formulation and SIMPLE algorithm. Temperature distributions, streamline profiles and Nusselt numbers of the fluid and pipe walls with time are analyzed for the case of electrical heat tracing of the pressurizer surge line while outsurging of the heatup operation.

## 2. MODEL FORMULATION

During plant heatup and cooldown operations the temperature difference between the pressurizer and the hot leg could be 167℃, in which case the effects of the thermal stratification must be taken into account.

The selected analysis model of this study, outsurging heatup operation is shown in FIG. 1(a). Symmetric configuration allows a half of the pipe cross section to be considered as shown in FIG. 1(b). External heating extends $3/4\pi<\theta<\pi$ and set the heat flux of the external heating as $10kW/m^2$ based on the commercial specification of the electrical heat tracing.

The major assumptions to solve the governing equations are as follows; (a) the flow of hot and cold fluid is 2-dimensional, (b) the thermal and hydrodynamic conditions are symmetrical with respect to the vertical center line, (c) the properties of fluid and solid except density in the body force term are treated as constant, (d) the compressibility effects, viscous dissipation and radiation heat transfer of fluids are neglected, (e) the thickness of the interface layer of the hot and cold fluid is neglected, and (f) the interface level comes down from the top to the center of the pipe for 35.3 seconds(dimensionless time, t=88.25) as the hot fluid with constant low velocity flows into the upper region of the stagnant cold fluid.

The dimensionless governing equations, the continuity, the momentum, and the energy equations for the unsteady 2-dimensional flow, are derived as

$$\frac{1}{r}\frac{\partial}{\partial r}(rv)+\frac{1}{r}\frac{\partial}{\partial \theta}(u) = 0 \tag{1}$$

$$\frac{\partial u}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}(ruv)+\frac{1}{r}\frac{\partial}{\partial \theta}(u^2) = -\frac{1}{r}\frac{\partial P}{\partial \theta}+C_1\left\{\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{1}{r}\frac{\partial u}{\partial \theta}\right) -\frac{u}{r^2}+\frac{2}{r^2}\frac{\partial v}{\partial \theta}\right\}-\frac{uv}{r}-\frac{Gr}{Re^2}T\sin\theta \tag{2}$$

$$\frac{\partial v}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}(rv^2)+\frac{1}{r}\frac{\partial}{\partial \theta}(uv) = -\frac{\partial P}{\partial r}+C_1\left\{\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{1}{r}\frac{\partial v}{\partial \theta}\right) -\frac{v}{r^2}-\frac{2}{r^2}\frac{\partial u}{\partial \theta}\right\}+\frac{u^2}{r}+\frac{Gr}{Re^2}T\cos 6 \tag{3}$$

$$\frac{\partial T}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}(rvT)+\frac{1}{r}\frac{\partial}{\partial \theta}(uT) = C_2\left\{\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial \theta}\left(\frac{1}{r}\frac{\partial T}{\partial \theta}\right)\right\} \tag{4}$$

Dimensionless variables are defined as follows;

$$r=\frac{r^*}{r_i^*},\quad a=\frac{r_0^*-r_i^*}{r_i^*},\quad v=\frac{v^*}{U_o},\quad u=\frac{u^*}{U_0},\quad t=\frac{t^*U_0}{r_i^*},\quad \alpha_s=\frac{k_s}{\rho_s C_{p_s}},\quad \alpha_f=\frac{k_f}{\rho_f C_{p_f}},\quad Pr=\frac{C_p\mu}{k},\quad Re=\frac{U_0 r_i^*}{\nu},$$

$$Gr=\frac{g\beta r_i^{*3}(T^*_{hot}-T^*_{cold})}{\nu^2},\ Bi=\frac{h(r_0^*-r_i^*)}{k_s},\ Biq = \frac{\dot{q}\cdot(r^*_0-r^*_i)}{k_s\cdot(T^*_w-T^*_0)},\ T=\frac{T^*-T^*_{cold}}{T^*_{hot}-T^*_{cold}},\ P=\frac{P'}{\rho_o U_0^2},\ P'=P^*+\rho_o g r\cos\theta \tag{5}$$

where, $U_o$ is the constant velocity in the axial direction and asterisks represent the dimensional values. Although the flow is assumed to be two dimensional, the axial velocity is used to define dimensionless parameter, $Ri(=Gr/Re^2)$. Also, $C_1$ and $C_2$ in the equation (2), (3) and (4) represent the diffusion coefficients in the momentum equations and the energy equation respectively, which state

$$C_1=\begin{cases}\frac{1}{Re} & at\ fluid\\ \infty & at\ solid\end{cases},\quad C_2=\begin{cases}\frac{1}{PrRe} & at\ fluid\\ \frac{\alpha_s/\alpha_f}{PrRe} & at\ solid\end{cases} \tag{6}$$

The initial and boundary conditions are as follows;

(a) $t=0 : 0 \le r < 1+a\ ;\quad u=v=0$ and $T=0$ (7)

(b) $t>0 :\quad 0 \le r \le 1+a$ and $\theta=0$ or $\pi$ ;  $u=\frac{\partial v}{\partial \theta}=\frac{\partial T}{\partial \theta}=0$ (8)

$0 \le \theta \le \pi$ and $1 \le r \le 1+a$ ;  $u=v=0$ (9)

$0 \le \theta \le \pi$ and $r = 1 + a$ ; $$\frac{\partial T}{\partial r} = -\frac{Bi(T_W - T_o)}{a} \tag{10}$$

(c) $0<t<88.25$ $\frac{3}{4}\pi \le \theta \le \pi$ and $r = 1 + a$ ; $$\frac{\partial T}{\partial r} = -\frac{(B_i + B_{iq})(T_W - T_o)}{a} \tag{11}$$

## 3. NUMERICAL ANALYSIS

The governing equations have been solved by the finite volume calculation procedure including SIMPLE algorithm, the power law scheme, and TDMA[5]. The preliminary tests for the number of grids and the time step, $\Delta t$, have been carried out and then an optimal grid system( $r \times \theta$=52x32) as FIG. 2 and an optimal time step, 0.1 second(t=0.25), are determined for this model. A grid distribution in the $\theta$-direction is increased in accordance with the angle and the $r$-direction is divided into three regions. In order to improve convergence, the under relaxation factors of velocity, pressure and temperature are applied 0.15, 0.8 and 0.8 respectively. The converged solutions are obtained when the error of energy balance is less then 0.1%. The analysis results are compared with the experimental data of a HDR-TEMR(T33.19) Test[6] to verify the program as shown in FIG. 3. The general agreement between this analysis and measured values is good except small discrepancies near the interface. These discrepancies are probably due to the following facts; (a) analysis model dose not include the interface fluctuation, (b) the thermal mixing between the hot and the fluid cold is not considered during outsurging time of the hot water to the pipe line.

## 4. RESULTS AND DISCUSSION

Thermal stratification is expected in the given conditions of FIG. 1(a) because *Ri* is calculated 1.35. In order to analyze the variation of streamline and temperature profiles due to the thermal stratification, the material properties for the surge line of a domestic PWR plant are used[7,8].

### 4.1 Flow and Temperature Distribution

The dimensionless distributions of streamlines and isotherms are given in FIG. 4. The values in bracket denote [maximum value (interval) minimum value] of isothermal distribution.

FIG. 4 (a), (b), and (c) show distributions of the streamlines and isotherms at dimensionless time t=25, 88.25 and 150 respectively. At these stages, isotherms are concentrated near the interface layer. It can be explained that the heat is only transfered by heat conduction from the hot fluid to the cold. At this moments temperature of the pipe wall lower section is increased by the external heating while in these time intervals. However, the heat from the external heating is not transfered to the cold fluid yet. The potential of streamlines in the hot region is greater than that in the cold region.

At dimensionless time t=750, 1,500 and 2,250, the distributions of the streamlines and isotherms are shown in FIG. 4(d), (e), and (f). The profiles of isotherms near the interface leads to a typical temperature distribution of the thermal stratification by the external heat input from bottom and the active natural convection between the hot fluid and the cold. As a result, the temperature differences between the inner and the outer surfaces of pipe decrease and the flow distributions become stable. The maximum dimensionless temperature difference between the hot and the cold sections of pipe inner wall is 0.424 at dimensionless time 1,500.

At dimensionless time t=9,000, 12,000 and 24,000, the distributions of the streamlines and isotherms are shown in FIG. 4(g), (h), and (i). As the heat transfer progresses, the temperature difference between two fluids is decreased and the streamlines are getting more stabilized. The thermal stratification phenomena is disappeared at about dimensionless time 9,000.

FIG. 5 shows the temperature differences in the fluids and the pipe inner and outer walls with time. The dimensionless temperature difference of the pipe outer surface rapidly fluctuates during first dimensionless time 2,500 and then decrease slowly with the same temperature difference as the inner wall. The maximum dimensionless temperature difference, $\Delta T_{max}$, of pipe inner wall is 0.424 at dimensionless time 1,500 under given condition. If it is compared with the case of no external heat tracing, external heat tracing is one of effective method to mitigate the thermal stratification by lessening $\Delta T_{max}$ about 0.1 and shortening the dimensionless time about 132.

## 4.2 Heat Transfer

To discuss the heat transfer with time, the local and mean Nusselt number along the inner surface of pipe, $Nu$ and $\overline{Nu}$, are defined as follows ;

$$Nu = \frac{h r_i^*}{k} = -\left.\frac{dT}{dr}\right|_{r=r_i}, \quad \overline{Nu} = \frac{\overline{h} r_i^*}{k} = \frac{1}{\pi}\int_0^{\pi} Nu \, d\theta \tag{12}$$

The local heat transfer rates along the inner surface of the pipe at dimensionless time t=88.25, 150, 750, 1,500 and 24,000 are presented in FIG. 6. Since the temperature difference between the hot fluid and the upper pipe inner wall is greater than the lower, the local heat transfer rates at the upper inner wall are calculated large. However, it rapidly decreases near the interface since the pipe wall in this region is affected by the cold fluid. About t=1,500, the local heat transfer rates are very small and nearly zero at t=24,000. Negative $Nu$'s are shown at t=150 and 750 because of the reversed direction of heat transfer from the pipe outside to the fluid inside by affecting the external heating.

FIG. 7 illustrates the mean heat transfer rates with time. Until about t=88.25 when temperature difference between the fluid and the inner wall of pipe is high, the heat transfer rate increases dramatically, and then decreases exponentially resulting almost no heat transfer at about t=2,250.

# 5. CLOSURE

The thermo-hydraulic transients of the thermal stratification have been numerically investigated in the horizontal surge line pipe and externally heated by electrical heat tracing. To this end, the unsteady 2-dimensional numerical analysis model has been developed. The temperature profiles in the fluid and pipe wall region, the difference of temperature between the upper and the lower section of the pipe, and the heat transfer rates under the given conditions are calculated with time. It was shown that, although, the complex flows in the upper region and the temperature difference between the inner and the outer pipe wall occurs during initial time interval when the interface level is varied, the temperature difference in the pipe wall is very small. In addition the thermal stratification is shown to decay out by the thermal mixture between hot and cold fluid eventually.

The numerical result of this study shows that the maximum dimensionless temperature difference between the hot and the cold sections of pipe inner wall is 0.424 at dimensionless time 1,500 and the thermal stratification phenomena is disappeared at about dimensionless time 9,000. This result means that external heat tracing can mitigate the thermal stratification phenomena by lessening $\triangle T_{max}$ about 0.1 and shortening the dimensionless time about 132 in comparison with no external heat tracing.

# 6. REFERENCES


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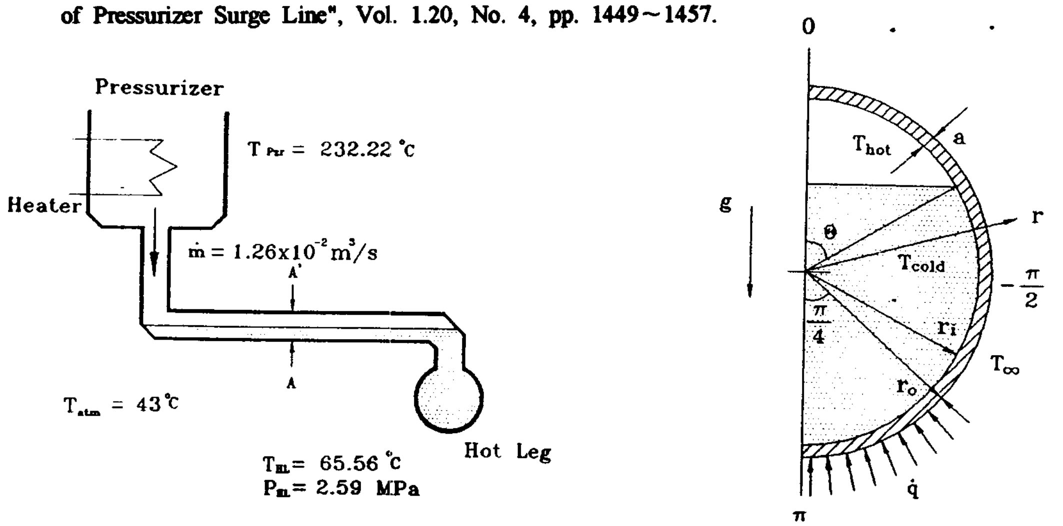
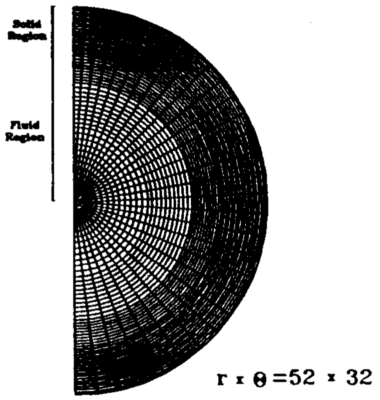



(a) CONDITIONS OF THE ANALYSIS MODEL (b) SCHEMATIC DIAGRAM OF THE CALCULATION DOMAIN
FIG. 1 THE HEATUP CONDITIONS AND CALCULATION DOMAIN



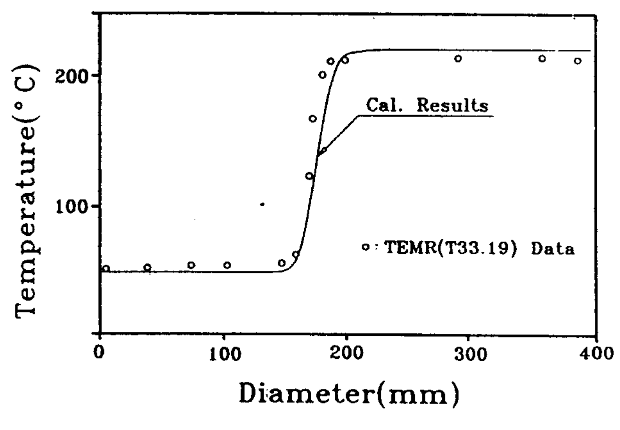




FIG. 2 GRID SYSTEM OF CALCULATION DOMAIN FIG. 3 THE COMPARISON BETWEEN OUR RESULTS AND A TEMR EXPERIMENT

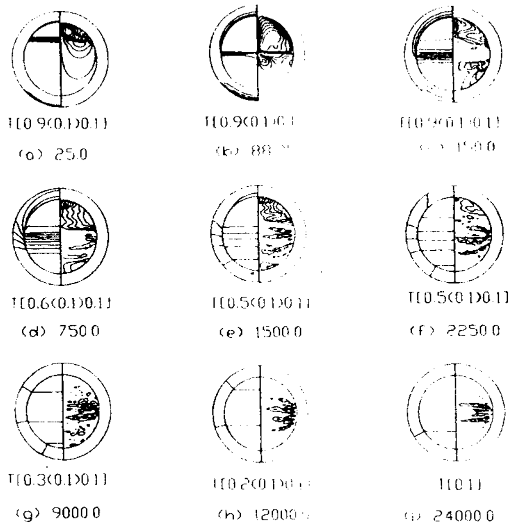



FIG. 4 THE DISTRIBUTION OF ISOTHERMS(LEFT) AND STREAMLINES(RIGHT)

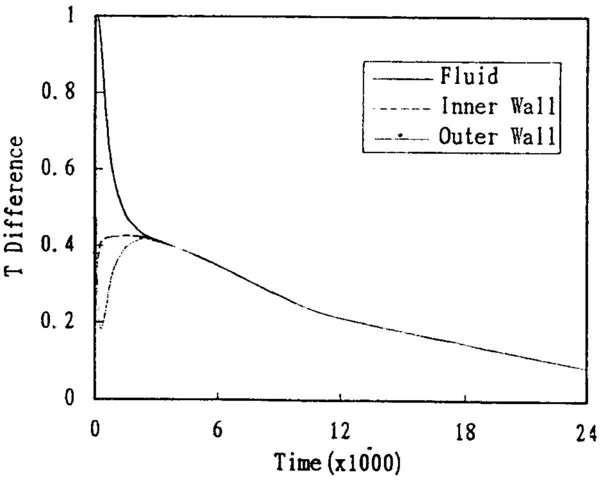



FIG. 5 THE TEMPERATURE DIFFERENCE OF FLUID AND PIPE INNER AND OUTER WALLS

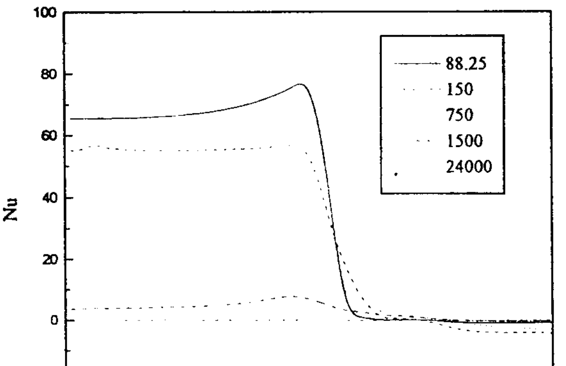



FIG. 6 THE LOCAL NUSSELT NUMBER ALONG THE PIPE INNER WALL

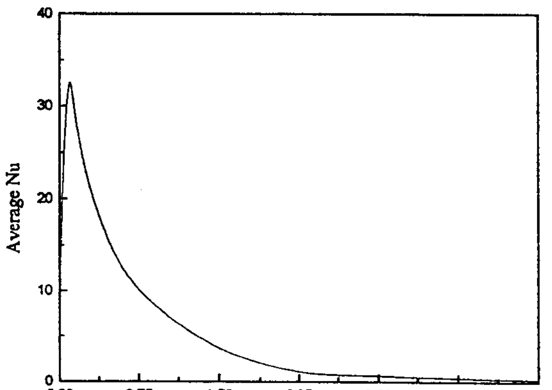



FIG. 7 THE MEAN NUSSELT NUMBER WITH TIME