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A Study on Development of a Plugging Margin Evaluation Method Taking Into Account the Fouling of Shell-and-Tube Heat Exchangers

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As the operating time of heat exchangers progresses, fouling caused by water-borne deposits and the number of plugged tubes increase and thermal performance decreases. Both fouling and tube plugging are known to interfere with normal flow characteristics and to reduce thermal efficiencies of heat exchangers. The heat exchangers of Korean nuclear power plants have been analyzed in terms of heat transfer rate and overall heat transfer coefficient as a means of heat exchanger management. Except for fouling resulting from the operation of heat exchangers, all the tubes of heat exchangers have been replaced when the number of plugged tubes exceeded the plugging criteria based on design performance sheet. This paper describes a plugging margin evaluation method taking into account the fouling of shell-and-tube heat exchangers. The method can evaluate thermal performance, estimate future fouling variation, and consider current fouling level in the calculation of plugging margin. To identify the effectiveness of the developed method, fouling and plugging margin evaluations were performed at a component cooling heat exchanger in a Korean nuclear power plant.

Key Words : Heat Exchanger, Plugging Margin, Fouling, Design Margin, Manufacture Margin, Component Cooling Water Heat Exchanger

1. Introduction

The main functions of heat exchangers are to provide heat transfer from one fluid to another in addition to offering leak tightness between the tube-side fluid and the shell-side fluid. Heat exchangers in nuclear power plants support or perform a specific function for shutting down a reactor to the safe shutdown condition, for maintaining the safe shutdown condition, or for mitigating the consequences of an accident. As the operating time of heat exchangers progresses, deterioration of heat transfer function may occur due to fouling or blockage of heat exchanger tubes. Fouling or blockage typically occurs in component cooling water heat exchangers exposed to raw water.

In-service performance testing in Korean nuclear power plants has been performed in order to scrutinize the thermal performance of heat exchangers with regard to heat transfer rate and overall heat transfer coefficient based on the ASME OM-S/G-Part 2 (ASME, 1994). While the thermal characteristics of heat exchangers can be identified by the testing, it is hard to predict fouling trend and tube plugging margin reflecting current

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fouling level and to device an appropriate management plan.

The purpose of this paper is to introduce a tube plugging margin evaluation method taking into account the fouling resulting from the operation of heat exchangers, which is composed of fouling evaluation and plugging margin evaluation. The applicability of the plugging margin evaluation method was reviewed by its application to a component cooling water heat exchanger.

2. Evaluation Theory and Methodology

2.1 Fouling evaluation method

Fouling should be significantly considered for the subcomponents of heat exchangers being exposed to borated reactor coolant water or raw water. If heat exchangers undergo wet-and-dry cycle repeatedly, precipitation of boron is vulnerable to occur. In addition, fouling may occur on the internal or external surfaces of subcomponents exposed to raw water due to airborne contamination and condensation of water vapor. Fouling interferes with normal flow characteristics and can degrade thermal efficiencies of the heat exchangers (Conklin, 1995).

To evaluate the tube plugging margin associated with fouling of shell-and-tube heat exchangers, fouling level resulting from operation should be preferentially identified. The fouling evaluation method suggested in this paper can be applied to any heat exchanger with the exception of coil tube heat exchangers. The basic data for the fouling evaluation are as follows :

- Process fluid inlet temperature (T_1), ℃
- Process fluid outlet temperature (T_2), ℃
- Cooling fluid inlet temperature (t_1), ℃
- Cooling fluid outlet temperature (t_2), ℃
- Process fluid flow rate (W_p), kg/sec
- Cooling fluid flow rate (W_c), kg/sec

After obtaining only five of the six parameters, the heat transfer rates for process and cooling fluids are calculated from Eqs. (1) and (2),

$$Q_p = W_p C_{P_p} (T_1 - T_2) \tag{1}$$

$$Q_c = W_c C_{P_c} (T_1 - T_2) \tag{2}$$

where

Q_p =heat transfer rate for process fluid, MW
Q_c =heat transfer rate for cooling fluid, MW

The logarithmic mean temperature difference ($LMTD$) is calculated from Eq. (3) for parallel flow and Eq. (4) for true counter flow.

$$LMTD = \frac{(T_1 - t_1) - (T_2 - t_2)}{\ln\left(\dfrac{T_1 - t_1}{T_2 - t_2}\right)} \tag{3}$$

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left(\dfrac{T_1 - t_2}{T_2 - t_1}\right)} \tag{4}$$

The $LMTD$ is adjusted by Eq. (5) for deviations from true counter flow.

$$MTD = LMTD \cdot F \tag{5}$$

where the MTD is mean temperature difference, and F is the $LMTD$ correction factor. F is calculated from Eq. (6) which equals 1 for true counter flow and parallel flow (Norris and Robert, 1991).

$$F = \frac{B}{R-1} \cdot \frac{\ln\left(\dfrac{1-X}{1-RX}\right)}{\ln\left(\dfrac{2-X(R+1-B)}{2-X(R+1+B)}\right)} \tag{6}$$

where B, X, R, and P are calculated from Eqs. (7), (8), (9), and (10).

$$B = \sqrt{1+R^2} \tag{7}$$

$$X = \frac{1 - \left(\dfrac{1-RP}{1-P}\right)^{1/n}}{R - \left(\dfrac{1-RP}{1-P}\right)^{1/n}} \tag{8}$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} \tag{9}$$

$$P = \frac{t_2 - t_1}{T_1 - t_1} \tag{10}$$

where n is the number of shell-side flow passes. The overall heat transfer coefficient using the adjusted mean temperature difference is calculated from

$$U = \frac{Q}{A_o \cdot MTD} \tag{11}$$

where

U = overall heat transfer coefficient based on outside surface area, W/m²-℃

A_o = total effective surface area based on outside surface area, m²

In case the types of heat exchangers are split-flow, divided-flow, and cross-flow, the overall heat transfer coefficient is calculated from Eq. (12) based on the number of transfer unit (NTU).

$$U = \frac{NTU \cdot W_c C_{Pc}}{A_o} \tag{12}$$

where NTU is a function of R from Eq. (9) and P from Eq. (10). NTU is calculated from Eq. (13) for $R=0$ and $R=\infty$ and from Eq. (14) for $R \neq 0$ and $R \neq \infty$.

$$NTU = \ln\left(\frac{1}{1-P}\right) \tag{13}$$

$$NTU = \frac{1}{\sqrt{R^2+1}} \cdot \ln\left(\frac{2-P(R+1-\sqrt{R^2+1})}{2-P(R+1+\sqrt{R^2+1})}\right) \tag{14}$$

The total fouling resistance (r_t) based on outside surface area using the values calculated above is calculated from Eq. (15) to assume the fouling level for heat exchangers.

$$U = \frac{1}{r_t + \frac{1}{h_o}\frac{1}{E_f} + r_w + \frac{1}{h_i}\frac{A_o}{A_i}} \tag{15}$$

where

h_o = outside film coefficient based on outside surface area, W/m²-℃

h_i = inside film coefficient based on inside surface area, W/m²-℃

r_t = total fouling resistance based on outside surface area, m²-℃/W (refer to Eq. (16))

r_w = tube wall resistance based on outside surface area, m²-℃/W

A_i = inside effective surface area based on inside surface area, m²

E_f = weighted fin efficiency (dimensionless, equal to 1 for non-finned tubes, less than 1 for finned tubes)

The total fouling resistance based on outside surface area can be calculated from

$$r_t = r_o \frac{1}{E_f} + r_i \frac{A_o}{A_i} \tag{16}$$

where

r_o = outside fouling resistance based on outside surface area, m²-℃/W

r_i = inside fouling resistance based on inside surface area, m²-℃/W

Inside film coefficient based on inside surface area can be calculated from Eq. (17) for $Re > 10{,}000$, Eq. (18) for $Re < 2{,}100$, and Eq. (19) for $2{,}100 < Re < 10{,}000$ (Perry et al., 1984).

$$h_i = 0.023 \frac{k_t}{d_i} Re^{0.8} Pr^{1/3} \left(\frac{\mu_{bt}}{\mu_{wt}}\right)^{0.14} \tag{17}$$

$$h_i = 1.86 \frac{k_t}{d_i} Re^{1/3} Pr^{1/3} \left(\frac{d_i}{L}\right)^{1/3} \left(\frac{\mu_{bt}}{\mu_{wt}}\right)^{0.14} \tag{18}$$

$$h_i = 0.116 \frac{k_t}{d_i} (Re^{2/3} - 125) Pr^{1/3} \left(1 + \left(\frac{d_i}{L}\right)^{2/3}\right) \left(\frac{\mu_{bt}}{\mu_{wt}}\right)^{0.14} \tag{19}$$

where

k_t = bulk thermal conductivity of the tube side fluid, W/m-℃

d_i = inside diameter of tube, m

μ_{bt} = bulk absolute viscosity of the tube side fluid, Pa-sec

μ_{wt} = absolute viscosity of the tube side fluid at the tube wall temperature, Pa-sec

L = total length of tube, m

ρ_t = density of the tube side fluid, kg/m³

V = velocity inside tube based on flow rate, m/sec

Reynolds number and Prandtl number of the tube side fluid can be calculated from Eqs. (20) and (21).

$$Re = \frac{\rho_t V d_i}{\mu_{bt}} \tag{20}$$

$$Pr = \frac{C_{Pt} \mu_{bt}}{k_t} \tag{21}$$

Outside film coefficient (Perry et al., 1984) based on outside surface area (h_o) can be calculated from

$$h_o = h_k J_c J_l J_b J_r \tag{22}$$

where

h_k =heat transfer coefficient of the shell side fluid, W/m^2-℃
J_c =correction factor for baffle configuration effects
J_l =correction factor for baffle leakage effects
J_b =correction factor for bypass flow
J_r =correction factor for adverse temperature gradient at low Reynolds numbers

Reynolds number of the shell side fluid can be calculated from Eq. (23), which is used for calculation of the outside film coefficient.

$$Re_s=\frac{d_o W_s}{\mu_{bs} S_m} \tag{23}$$

where

W_s =weight flow rate of the shell side fluid, kg/sec
μ_{bs} =viscosity at bulk temperature of the shell side fluid, Pa-sec
S_m =cross flow area at or near centerline for one cross-flow section, m^2

S_m is calculated from Eq. (24) for rotated and in-line square tube layouts and from Eq. (25) for triangular tube layouts.

$$S_m=l_s\left[D_s-D_{otl}+\frac{D_{otl}-d_o}{p_n}(p'-d_o)\right] \tag{24}$$

$$S_m=l_s\left[D_s-D_{otl}+\frac{D_{otl}-d_o}{p'}(p'-d_o)\right] \tag{25}$$

where

l_s =baffle spacing, m
D_s =inside diameter of heat exchanger shell, m
D_{otl}=outside diameter of tube bundle, m
d_o =outside diameter of tube, m
p' =center-to-center spacing of tubes (tube pitch), m
p_n =tube pitch normal to flow, m

The heat transfer coefficient of the shell side fluid can be calculated from Eq. (26), where the correction factor j_k is shown (Perry, Green, and Maloney, 1984) to be:

$$h_k=j_k C_{Ps}\frac{W_s}{S_m}\left(\frac{k_s}{C_{Ps}\mu_{bs}}\right)^{2/3}\left(\frac{\mu_{bs}}{\mu_{ws}}\right)^{0.14} \tag{26}$$

where

C_{Ps} =specific heat of the shell side fluid, J/kg-℃
μ_{ws} =absolute viscosity at tube wall temperature of the shell side fluid, Pa-sec

The fraction of total tubes in cross flow F_c can be calculated from Eq. (27), which is used for the calculation of the correction factor for baffle configuration effects J_c.

$$F_c=\frac{1}{\pi}\left[\pi+2\frac{D_s-2l_c}{D_{otl}}\cdot\sin\left(\cos^{-1}\frac{D_s-2l_c}{D_{otl}}\right)-2\cos^{-1}\frac{D_s-2l_c}{D_{otl}}\right] \tag{27}$$

where l_c is baffle cut. The correction factor for baffle leakage effects J_l can be calculated based on the tube-to-baffle leakage area for one baffle S_{tb}, the shell-to-baffle leakage area for one baffle S_{sb}, the cross flow area at or near centerline for one cross flow section S_m. S_{tb} and S_{sb} can be calculated from Eqs. (28) and (29), respectively.

$$S_{tb}=bD_oN_T(1+F_C) \tag{28}$$

$$S_{sb}=\frac{D_s\delta_{sb}}{2}\left[\pi-\cos^{-1}\left(1-\frac{2l_c}{D_s}\right)\right] \tag{29}$$

where

b =6.223×10^{-4} for SI unit and 1.701×10^{-4} for U.S. customary
$\cos^{-1}(1\text{-}2l_c/D_s)$ =between 0 and $\pi/2$ in radians
δ_{sb} =diametral shell-to-baffle clearance
N_T =total number of tubes

The correction factor for bypass flow J_b can be calculated on the basis of number of sealing strips N_{ss} and number of tubes in one cross flow section N_C. The fraction of cross flow area available for bypass flow F_{bp} can be calculated from

$$F_{bp}=\frac{(D_s-D_{otl})\,l_s}{S_m} \tag{30}$$

If the Reynolds number of the shell side fluid is less than 100, the correction factor for adverse temperature gradient at low Reynolds numbers (j_r) should be considered in the calculation of the outside film coefficient. If the Reynolds number exceeds 100, j_r equals to 1.

The tube wall resistance r_w can be calculated from Eq. (31) for bare tubes and Eq. (32) for integrally finned tubes (Richard, 1988).

$$r_w = \frac{d_o}{2k} \ln\left(\frac{d_o}{d_o - 2t}\right) \tag{31}$$

$$r_w = \frac{t}{k} \cdot \frac{d_o + 2N_f w_f (d_o + w)}{d_o - t} \tag{32}$$

where

N_f =number of fins per meter
w_f =fin height, m
t =tube wall thickness, m

Finally, the total fouling resistance based on outside surface area r_t can be calculated from

$$r_t = \frac{1}{U} - \frac{1}{h_o} \frac{1}{E_f} - r_w - \frac{1}{h_i} \frac{A_o}{A_i} \tag{33}$$

Because the total fouling resistance generally increases as the operating time progresses, the thermal performance variations of heat exchangers may be predicted by periodic tests and fouling evaluations.

2.2 Plugging margin evaluation method

To evaluate plugging margin of heat exchangers, system design margin and manufacture margin should be calculated on the basis of the design materials. The plugging margin can be calculated from Eq. (34) represented as the sum of the system design margin (α_s) and the manufacture margin (α_m) (KOPEC, 2003).

$$\alpha = \alpha_s + \alpha_m \tag{34}$$

To calculate the system design margin of heat exchangers, the operating modes considered in design should be identified. The operating modes considered in the design of the closed cooling water system are plant start-up, normal operation, shutdown, refueling, recirculation, safety injection, etc. The system design margin based on the operating mode used in heat exchanger design can be calculated from

$$\alpha_s = 100 \cdot \left(\frac{(UA_o)_d}{(UA_o)_s} - 1\right) \tag{35}$$

where the term UA_o is the heat exchanger performance capability. Both $(UA_o)_d$ considering in actual design and $(UA_o)_s$ covering design conditions can be calculated from Eq. (11). The manufacture margin can be calculated from

$$\alpha_m = 100 \cdot \left(\frac{A_{o,m}}{A_{o,r}} - 1\right) \tag{36}$$

where

$A_{o,m}$=actual manufacture area, m^2
$A_{o,r}$=required area in calculation of heat exchanger thermal performance, m^2

The plugging margin of Eq. (34) can be converted into the number of spare tubes ($N_{p,d}$) from

$$N_{p,d} = \frac{A_{o,m}\alpha}{A_{o,l}} \tag{37}$$

where

$A_{o,l}$ =outside surface area of a tube, m^2

The fouling level resulting from the operation of heat exchangers is not covered in the plugging margin calculated from Eq. (34). In order to calculate the adjusted plugging margin taking into account the fouling level, the fouling factor (r_a) corresponding to a plugged tube should be calculated from

$$r_a = \frac{r_{t,d}}{N_{p,d}} \tag{38}$$

where

$r_{t,d}$ =fouling factor reflected in design

The adjusted plugging margin taking into account the current fouling level and the number of plugged tubes (N_{plug}) can be calculated from Eqs. (39) for $r_t \leq r_a \cdot N_{plug}$ and (40) for $r_t > r_a \cdot N_{plug}$.

$$N_{p,New} = N_{p,d} \tag{39}$$

$$N_{p,New} = N_{p,d} - \frac{r_t - r_a N_{plug}}{r_a} \tag{40}$$

3. Application of Plugging Margin Evaluation Method Considering Fouling

Fouling level and plugging margin for a component cooling water heat exchanger in a Korean nuclear power plant were evaluated using the method developed in this study. The component cooling water heat exchanger is one shell and one tube pass counter-current heat exchanger. The shell side is supplied with component cooling water from the discharge of the CCW pumps and the tube side is supplied with sea water from the NSCWS. The number of plugged tubes of the heat exchanger was 232 in June 2003. Figure 1 shows the schematic diagram for the component cooling water system around the component cooling water heat exchangers. The design data and the measured average data in June 2003 for the calculation of the fouling level and plugging margin are shown in Table 1.

The fouling evaluation results using the measured data from 1990 to 2003 are shown in Fig. 2. The calculated fouling factors of the component cooling heat exchanger have changed due to continued cleaning and plugging. An example using the measured data on June 12, 2003, among the fouling evaluation results is presented in Table 2. The component cooling water heat exchanger meets the acceptance criteria in terms of overall heat transfer coefficient, performance capability, and fouling factor described in the ASME OM-S/G part 2.

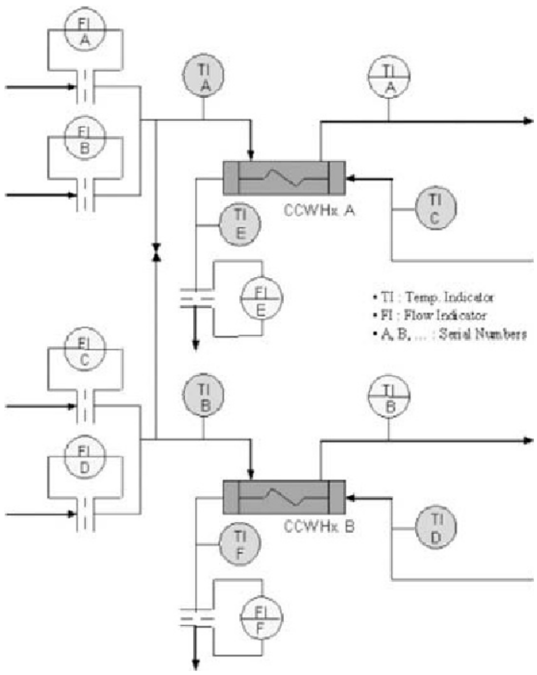

Fig. 1 Schematic diagram for CCW system

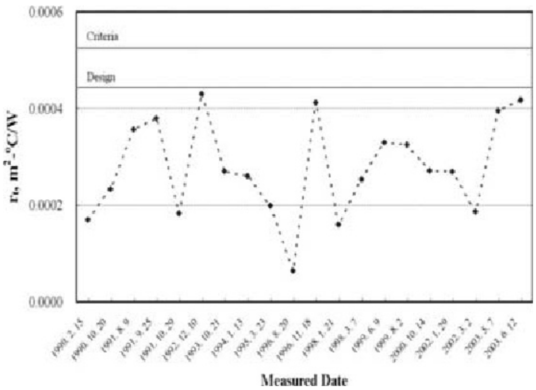

Fig. 2 Fouling evaluation results

Table 1 Input data for evaluation

	Process Fluid			Cooling Fluid		
	Inlet Temp., ℃	Outlet Temp., ℃	Flow Rate, kg/hr	Inlet Temp., ℃	Outlet Temp., ℃	Flow Rate, kg/hr
Design	41.00	35.00	2.95×10^6	32.00	37.39	3.49×10^6
2003. 6	26.20	22.60	2.52×10^6	21.50	23.60	4.48×10^6

Table 2 Fouling evaluation results

Items	Units	Design	Criteria	2003. 6
r_t	m^2 ℃/W	0.000443	≤0.000526	0.000418
U	W/m^2 ℃	1, 335.61	≥1, 202.05	1, 374.43
U*	–	–	≥0.9	0.97

U* : Specific Performance Capability (UA/U_oA_o)

Plugging margin evaluation results using the measured data as of June 12, 2003 are presented in Table 3. The operating mode reflected in design of the component cooling water heat exchanger is hot shutdown. The calculated plugging margin was 5.73% based on the design performance sheet, which is the sum of the system design margin 4.971% and the manufacture margin 0.762%. Thus the number of spare tubes based on the plugging margin was turned out to be 232.

The recalculated number of the spare tubes is 232, being the same as that of the design plugging margin because the calculated fouling factor is less than $r_a \cdot N_{plug}$. Figure 3 shows the adjusted plugging criteria expressed as the number of the spare tubes with the number of plugged tube to date. Since the number of the current plugged tubes, 232, reached the adjusted plugging criteria, appropriate actions such as heat exchanger replacement or all tube replacement should be taken in the component cooling water heat exchanger.

Table 3 Plugging margin evaluation results

Items		Units	Values
Total number of tubes		–	4,012
Number of plugged tubes (percentage)		–	232 (5.8%)
Heat transfer rate in design		MW	20.51
Heat transfer rate at hot shutdown		MW	19.97
Design margin	$(UA)_d$	MW/℃	6.222
	$(UA)_s$	MW/℃	5.928
	α_s	%	4.971①
Manufacture margin	$A_{o,m}$	m^2	4,694
	$A_{o,r}$	m^2	4,659
	α_m	%	0.762②
Plugging margin of tubes α_s(①)$+\alpha_m$(②)		%	5.73
Number of tubes by plugging margin		–	232
Fouling factor in design		m^2 ℃/W	0.000443
Fouling factor in June 2003		m^2 ℃/W	0.000418
Fouling factor by plugged tubes ($r_a N_{plug}$)		m^2 ℃/W	0.000443
Number of tubes by new plugging margin		–	232

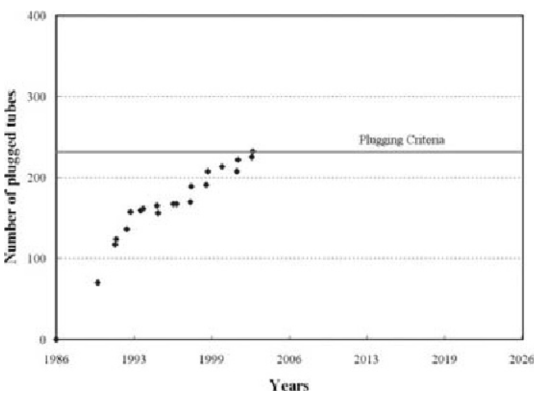

Fig. 3 Plugging margin evaluation results

4. Conclusions

Tube plugging margin has been used as the main criteria for the determination of the operating lifetime of heat exchangers. Except for the fouling resulting from the operation of heat exchangers, however, the system design margin and the manufacture margin have only been considered in the determination of the tube plugging margin. In order to reflect fouling level resulting from operation, new plugging margin evaluation method taking into account the fouling of shell-and-tube heat exchangers is suggested in this study, which evaluates the thermal performance for heat exchangers, estimates the future fouling variations, and considers the current fouling level in the calculation of plugging margin. The method used in the new plugging margin evaluation method involves determining the system design margin and the manufacture margin in design, and then determining the adjusted plugging margin based on the fouling level of heat exchangers.

A sample calculation for a component cooling water heat exchanger in a Korean nuclear power plant was performed using the new method. From the evaluation result, it was identified that appropriate actions such as heat exchanger replacement or all tube replacement should be taken.

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