NUCLEAR FEEDWATER HEATER PERFORMANCE INDICATORS

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Abstract

The nuclear industry's practice of accurately measuring feedwater flow affords an opportunity for the thermal performance engineer to monitor the thermal performance of feedwater heaters. This paper will discuss how the thermal performance engineer may use this and other station instrumentation to calculate meaningful feedwater heater performance indicators. The paper will illustrate how one may calculate an apparent feedwater heater fouling based on the measured overall heat transfer rate and an analysis of the shell-side and tube-side convection heat transfer coefficients. The paper will provide a case study of how these techniques have been applied to one of Ontario Power Generation's Nuclear Plants.

Feedwater Heater Design Conditions

The nuclear generating station feedwater heater design information for Feedwater Heater Number 2 (HX2) is shown in Table A-1, Appendix A. A review of the feedwater heater design information reveals that the resistances to heat transfer listed on the heat exchanger manufacturer's data sheets are referenced to the outside tube diameter. Therefore, the inverse of the sum of the resistances is equal to the calculated heat transfer coefficient without correction for the different surface areas. That is,

$$U = \frac{1}{r_{sc} + r_{sf} + r_w + r_{tf} + r_{tc}} = \frac{Q}{A_S\ EMTD}$$

Analysis Approach

Figure B-1, Appendix B, shows the analysis model for a typical feedwater heater with an integral subcooling zone but without a desuperheating zone. Condensate or feedwater enters the feedwater heater through the inlet channel and passes through the tubes which pass first through the subcooling zone and then through the condensing zone before exiting the feedwater heater through the outlet channel. Extraction steam and cascading drains enter the feedwater heater on the shell side into the condensing zone where the extraction steam and flashing drains are condensed in an isothermal process. The condensate is then further cooled as it passes through the shell side of the subcooling zone of the feedwater heater before exiting the feedwater heater through the drain outlet. Since the tube-side temperature exiting the subcooling zone cannot be measured, the physical properties of the fluid (specific heat, thermal conductivity, and viscosity) on the tube side are based on the average tube-side temperature. The tube-side performance of the feedwater heaters is based on the effectiveness method as defined by

$$Q = C_t (t_o - t_i) = C_t P (T_{sat} - t_i)$$

$$P = \frac{t_o - t_i}{T_{sat} - t_i}$$

The effectiveness as defined here is referenced to the cold stream (tube side). Algorithms for the effectiveness of the subcooling and condensing sections of the feedwater heaters are as follows (see Reference 1, p. 904):

$$P_1 = \frac{1 - e^{[-NTU_1 (1 - R_1)]}}{1 - R_1 \, e^{[-NTU_1 (1 - R_1)]}}$$

$$P_2 = 1 - e^{-NTU_2}$$

where the subcooling section is assumed to be a pure counter-flow heat exchanger and the condensing section is isothermal. The variables are

$$NTU_1 = \frac{U_1 \; A_{s,1}}{C_t}$$

$$NTU_2 = \frac{U_2 \; A_{s,2}}{C_t}$$

and

$$R_1 = \frac{C_t}{C_{s,1}}$$

$$C_t = m_t \, c_{p-t}$$

Since the temperature of the shell-side drains in the subcooling zone is very close to that of the tube-side fluid, the properties of the shell-side flow are assumed equal to those of the tube side. Therefore,

$$C_{s,1} = m_{d,o}\, c_{p-t}$$

where

$$m_{d,o} = m_e + m_{d,i}$$

If the overall coefficients of heat transfer for the subcooling and condensing zones are known, then the heat transferred in the drain cooling section may be calculated from

$$Q_1 = C_t\ P_1\ (T_{sat} - t_1)$$

and the tube-side temperature leaving the drain cooler is

$$t_2 = t_1 + \frac{Q_1}{C_t}$$

Similarly, the heat transferred in the condensing section, Q_2, is

$$Q_2 = C_t\ P_2\ (T_{sat} - t_2)$$

and the outlet temperature is

$$t_3 = t_2 + \frac{Q_2}{C_t}$$

and the terminal temperature difference is

$$TTD_{calc} = T_{sat} - t_3$$

The shell-side outlet temperature may be calculated from

$$T_o = T_{sat} - \frac{Q_1}{C_{s,1}}$$

And the drain cooler outlet approach is

$$DOA_{calc} = T_o - t_1$$

A check on the methods of feedwater heater analysis is to confirm the following equation:

$$Q = Q_1 + Q_2$$

$$C_t (t_3 - t_1) = C_t\, P_1 (T_{sat} - t_1) + C_t\, P_2 (T_{sat} - t_2)$$

Where the values for P_1 and P_2 are calculated as shown above.

Verification of Analysis Approach at Baseline Conditions

If the proposed method of analysis is valid, the procedure should yield results that agree fairly well with the vendor data at baseline or design conditions. The following analysis illustrates the process of verification against baseline data for HX2. (See Table A-1.) For the average tube-side temperature of 74.9 ^oC, the tube-side properties are as follows:

$$c_{p-t} = 4.192 \left(\frac{kJ}{kg\ ^oC} \right)$$

$$k_t = 6.668\, x 10^{-4} \left(\frac{kJ}{s\, m\ ^oC} \right)$$

$$\mu_t = 3.774\, x 10^{-4} \left(\frac{kg}{m\, s} \right)$$

Since the tube-side flow rate is 291.967 (kg/s), the tube-side thermal capacity rate is, therefore,

$$C_t = m_t\, c_{p-t} = 1224 \left(\frac{kJ}{s\ ^oC} \right)$$

The total shell-side flow rate consists of 11.308 (kg/s) in extraction steam and 10.898 (kg/s) in cascading drains. So

$$m_{d,o} = m_e + m_{d,i} = 22.206 (kg/s)$$

Therefore, the shell-side capacity rate through the subcooling zone is

$$C_{s,1} = m_{d,o}\, c_{p-t} = 93 \left(\frac{kJ}{s\ ^oC} \right)$$

And the capacitance ratio is

$$R_1 = \frac{C_t}{C_{s,1}} = 13.15$$

For the baseline conditions, the overall heat transfer coefficient for the subcooling zone is known from

$$U_1 = \frac{1}{r_{sc,1} + r_{sf,1} + r_w + r_{tf} + r_{tc,1}} = 2{,}125\left(\frac{W}{m^2\ {}^oC}\right)$$

Therefore,

$$NTU_1 = \frac{U_1\ A_{s,1}}{C_t} = 0.123$$

and

$$P_1 = \frac{1 - e^{[-NTU_{t,1}(1-R_1)]}}{1 - R_1\ e^{[-NTU_{t,1}(1-R_1)]}} = 0.06$$

Therefore, the calculated subcooling zone heat transfer rate is

$$Q_1 = C_t P_1 (T_{sat} - t_i) = 1{,}785 \left(kJ/s\right)$$

The value shown in Table A-1 is 1,746 (kJ/s). This small discrepancy in part is due to the fact that the subcooling section is not a pure counter-flow heat exchanger as $EMTD_1$ is not equal to $LMTD_1$, which is the basis for the for algorithm for P_1. The corresponding subcooling zone outlet temperature is

$$t_2 = t_i + \frac{Q_1}{C_t} = 65.27\ {}^oC$$

Since the value shown in Table A-1 is 65.24 ^oC the approximation, nevertheless, yields good agreement.

Similarly, for the condensing section,

$$U_2 = \frac{1}{r_{sc,2} + r_{sf,2} + r_w + r_{tf} + r_{tc,2}} = 3{,}364\left(\frac{W}{m^2\ {}^oC}\right)$$

$$NTU_2 = \frac{U_2\ A_{s,2}}{C_t} = 2.38$$

$$P_2 = 1 - e^{-NTU_2} = 0.908$$

$$Q_2 = C_t P_2 (T_{sat} - t_2) = 25{,}382 \left(kJ/s\right)$$

The value shown in Table A-1 is 25,409 (kJ/s). The corresponding condensing zone outlet temperature is

$$t_3 = t_2 + \frac{Q_2}{C_t} = 86.01\ ^oC$$

which agrees with the value in Table A-1. Therefore, the total calculated heat transfer is

$$Q = Q_1 + Q_2 = 27{,}166\left(kJ/_s\right)$$

whereas the value in Table A-1 is 27,155 (kJ/s). The calculated TTD is

$$TTD_{calc} = T_{sat} - T_{t,o} = 2.1\ ^oC$$

which agrees with the value in Table A-1.

The shell-side outlet temperature may be calculated from

$$T_o = T_{sat} - \frac{Q_1}{C_{s,1}} = 68.94\ ^oC$$

and the drain cooler outlet approach is

$$DOA_{calc} = T_o - t_1 = 5.1\ ^oC$$

These values are at variance with the shell-side outlet temperature and the drain cooler approach of 69.37 ^{o}C and 5.6 ^{o}C, respectively.

Analysis of Coefficients of Heat Transfer

The analysis described above illustrates that if the overall coefficient of heat transfer is known, the thermal performance of the feedwater heater may be characterized reasonably well by the effectiveness method. However, in order to use this approach at other than baseline conditions, the coefficients of heat transfer for the tube-side and shell-side convection films must be calculated.

A method of calculating the overall coefficient of heat transfer for the subcooling zone and the condensing zone may be calculated by developing an algorthm for the individual resistances to heat transfer for the tube-side and shell-side convection films, since tube-side and shell-side fouling resistances and the tube wall resistance are assumed to be constant.

The tube-side convection film coefficient for both the subcooling and condensing zones may be determined as follows:

First, calculate the tube-side flow area, A_t.

$$d_i = d_o - 2T_W$$

$$A_t = \frac{\pi\, d_i^2}{4} N$$

Then calculate the tube-side mass flux, G_t.

$$G_t = \frac{m_t}{A_t}$$

Calculate the tube-side Reynolds number, Re_t and the Prandtl number, Pr_t .

$$\mathrm{Re}_t = \frac{G_t\, d_i}{\mu_t}$$

$$\mathrm{Pr}_t = \frac{\mu_t\, c_{p-t}}{k_t}$$

The Petukhov and Kirillov correlation is used to calculate the tube-side Nusselt number. (See Reference 1, p. 469.) First calculate the Fanning friction number

$$f = (1.58 \ln \mathrm{Re}_t - 3.28)^{-2}$$

then the Nusselt number.

$$Nu = \frac{(f/2)\, \mathrm{Re}_t\, \mathrm{Pr}_t}{1.07 + 12.7\,(f/2)^{1/2}\,(\mathrm{Pr}_t^{1/3} - 1)}$$

Finally, calculate the tube-side convection film coefficient.

$$h_t = Nu_t \frac{k_t}{d_i}$$

The tube-side convection film resistance is the inverse of the convection film coefficient referenced to the shell-side area.

$$r_{tc} = (d_o/d_i)\frac{1}{h_t}$$

The procedure for implicitly calculating the shell-side convection film coefficient for the subcooling zones of the feedwater heaters is complex and requires extensive knowledge of the internal design of the heat exchanger. (See Reference 1, Appendix N-6.) However, the value that must be used to arrive at the overall coefficient of heat transfer specified by the heat exchanger manufacturer may be determined by the back-calculation method, and the corresponding resistance to heat transfer may be compared with that specified in the heat exchanger data sheet as follows:

$$h_{sc,1} = \frac{1}{1/U_1 - [\, r_{tc} + r_{tf} + r_w + r_{sf} \,]}$$

$$r_{sc,1} = \frac{1}{h_{sc,1}}$$

This procedure amounts to a check on the adequacy of the algorithm for predicting the tube-side convection film coefficient as all of the other variables are taken from the heat exchanger data sheet.

For the condensing zone, the shell-side convection film coefficient may be computed directly from the following equation from the Bleed Heater Manufacturers Association expressed in British units (See Reference 1, p.800):

$$h_{sc,2} = \frac{1}{0.06834(T_{sat} - 0.2\,LMTD_2)^{-0.8912}}$$

Expressed in SI units, this equation becomes

$$h_{sc,2} = \frac{5.67826}{0.06834(1.8T_{sat} - 0.36\,LMTD_2 + 32)^{-0.8912}}$$

where

$$LMTD_2 = \frac{t_3 - t_2}{Ln\left(T_{sat} - t_2 \big/ T_{sat} - t_3 \right)}$$

However, if the resulting shell-side convection film coefficient exceeds 2,500 Btu/(hr ft^2 ^{o}F) or 14,196 W/(m^2 ^{o}C), this value is to be used. The corresponding resistance to heat transfer is

$$r_{sc,2} = \frac{1}{h_{sc,2}}$$

This analysis approach may be tested by comparing the resistances calculated for baseline conditions against those listed in the heat exchanger data sheet for HX2. (See Table A-1.) Tube-side convection film resistance is calculated for both the subcooling and condensing zones as follows:

$$d_i = d_o - 2T_W = 0.0173\,(m)$$

$$A_t = \frac{\pi\, d_i^2}{4} N = 0.1709\,(m^2)$$

$$G_t = \frac{m_t}{A_t} = 1{,}720\left(\frac{kg}{m^2\, s}\right)$$

$$\mathrm{Re}_t = \frac{G_t\, d_i}{\mu_t} = 78{,}600$$

$$\mathrm{Pr}_t = \frac{\mu_t\, c_{p-t}}{k_t} = 2.37$$

$$f = (1.58 \ln \mathrm{Re}_t - 3.28)^{-2} = 0.00474$$

$$Nu = \frac{(f/2)\, \mathrm{Re}_t\, \mathrm{Pr}_t}{1.07 + 12.7\,(f/2)^{1/2}\,(\mathrm{Pr}_t^{1/3} - 1)} = 284$$

$$h_t = Nu_t \frac{k_t}{d_i} = 10{,}982\left(\frac{W}{m^2\, {}^oC}\right)$$

$$r_{tc} = (d_o/d_i)\frac{1}{h_t} = 0.000100\left(\frac{m^2\, {}^oC}{W}\right)$$

This calculated film resistance at reference conditions is applicable to both the subcooling zone and the condensing zone of HX2. This calculated value may be compared with the values of 0.000106 ($m^2\,{}^oC/W$) and 0.000114 ($m^2\,{}^oC/W$) for the subcooling zone and the condensing zones, respectively, for HX2 in Table A-1.

The shell-side convection film resistance is calculated as follows for the subcooling zone at baseline conditions for HX2:

$$h_{sc,1} = \frac{1}{1/U_1 - [r_{tc} + r_{tf} + r_w + r_{s.1}]} = 4{,}170\left(\frac{W}{m^2\, {}^oC}\right)$$

$$r_{sc,1} = \frac{1}{h_{sc,1}} = 0.000240 \left(\frac{m^2 \ {}^oC}{W} \right)$$

This calculated value may be compared with the value of 0.000226 (m^2 $^oC/W$) for HX2 in Table A-1.

The shell-side convection film resistance is calculated as follows for the condensing zone at baseline conditions for HX2:

$$LMTD_2 = \frac{t_3 - t_2}{Ln\left({T_{sat} - t_2}\Big/{T_{sat} - t_3} \right)} = 8.72\ {}^oC$$

$$h_{sc,2} = \frac{1}{0.06834(1.8T_{sat} - 0.36\, LMTD_2 + 32)^{0.8912}} = 8{,}814 \left(\frac{W}{m^2\ {}^oC} \right)$$

$$r_{sc,2} = \frac{1}{h_{sc,2}} = 0.000113 \left(\frac{m^2\ {}^oC}{W} \right)$$

This calculated value is the same as the value in Table A-1.

ANALYSIS OF FEEDWATER HEATER TEST RESULTS

Apparent Fouling Resistance

The approach in evaluating feedwater heater test results is to determine an "apparent" fouling resistance that will yield the measured total heat transfer. This "apparent" fouling resistance may be due to fouling on the tube side or the shell side of the tube or may be due to some condition other than fouling that may result in poor heat transfer. Possible problems with feedwater heater performance in addition to tube-side or shell-side fouling include air binding, by-passing around the pass partition plate in the channel head, plugged tubes, and improper level control. The "apparent" fouling resistance is expressed as a fouling ratio, FR, which is a multiple of the sum of the design fouling resistances on the tube and shell side such that

$$U = \frac{1}{r_{sc} + r_w + r_{tc} + FR\,(r_{sf} + r_{tf})_{design}}$$

For a given test condition, there is a fouling ratio which would yield a value for the overall heat transfer coefficient when inserted in the following equation that would result in

$$Q_{test} = Q_{calc.}$$

where

$$Q_{test} = C_t\,(t_3 - t_1)$$
$$Q_{calc} = Q_1 + Q_2 = C_t\,P_1\,(T_{sat} - t_1) + C_t\,P_2\,(T_{sat} - t_2)$$

Since P_1 and P_2 are functions of U_1 and U_2 as discussed above, one may select a value for FR which when incorporated into the equation for U above with the appropriate expression for r_{sc} and r_{tc} will satisfy the requirement that the calculated heat transfer rate must equal the measured value. Since for feedwater heaters with an integral subcooling zone the temperature leaving the subcooling zone cannot be measured, the value of FR must be determined by trial and error. The accuracy of the proposed approach may be evaluated by computing the required fouling ratio at baseline conditions where the fouling ratio should be 1.0.

Determining the Fouling Ratio at Test Conditions

To illustrate the procedures discussed in the previous sections, consider the results of a test, in which the values listed in Table A-2 were measured or calculated for HX2.

For the average tube-side temperature of 74.3 °C, the tube-side properties are as follows:

$$c_{p-t} = 4.191\left(\frac{kJ}{kg\ {}^{o}C}\right)$$

$$k_t = 6.663\,x10^{-4}\left(\frac{kJ}{s\,m\ {}^{o}C}\right)$$

$$\mu_t = 3.806\,x10^{-4}\left(\frac{kg}{m\,s}\right)$$

Since the tube-side flow rate is 297.9 (kg/s), the tube-side thermal capacity rate is, therefore,

$$C_t = m_t\,c_{p-t} = 1249\left(\frac{kJ}{s\ {}^{o}C}\right)$$

The total shell-side flow rate consists of 11.308 (kg/s) in extraction steam and 10.898 (kg/s) in cascading drains. So

$$m_{d,o} = m_e + m_{d,i} = 23.738\,(kg/s)$$

Therefore, the shell-side capacity rate through the subcooling zone is

$$C_{s,1} = m_{d,o}\,c_{p-t} = 99\left(\frac{kJ}{s\ {}^{o}C}\right)$$

And the capacitance ratio is

$$R_1 = \frac{C_t}{C_{s,1}} = 12.55$$

The tubeside convection film resistance is calculated as follows for both the subcooling and condensing zones:

$$G_t = \frac{m_t}{A_t} = 1{,}750\left(\frac{kg}{m^2\ s}\right)$$

$$\mathrm{Re}_t = \frac{G_t\ d_i}{\mu_t} = 79{,}500$$

$$\mathrm{Pr}_t = \frac{\mu_t\ c_{p-t}}{k_t} = 2.39$$

$$f = (1.58\ \ln \mathrm{Re}_t - 3.28)^{-2} = 0.00473$$

$$h_t = Nu_t\ \frac{k_t}{d_i} = 11{,}133\left(\frac{W}{m^2\ {}^oC}\right)$$

$$r_{tc} = ({}^{d_o}\!/\!_{d_i})\frac{1}{h_t} = 0.000099\left(\frac{m^2\ {}^oC}{W}\right)$$

The shell-side convection film resistance for the subcooling zone at test conditions is calculated by relating it to the value back-calculated value at baseline conditions. From the Delaware method described in Reference 1, p. 733,

$$h_{sc,1} = J_T\ h_{sc,1,ideal}$$

Therefore,

$$J_T = (\frac{h_{sc,1}}{h_{sc,1,ideal}})_{design} = (\frac{h_{sc,1}}{h_{sc,1,ideal}})_{test}$$

and

$$(h_{sc,1})_{test} = (\frac{h_{sc,1,test}}{h_{sc,1,design}})_{ideal}\ h_{sc,1,design}$$

where

$$h_{sc,1} = Nu_{s,1}\,\frac{k_{s,1}}{d_o}$$

From Taborek, a simple correlation for the Nusselt number on the shell side is (See p.734 of Reference 1.)

$$h_{sc,1} = 0.2\ \mathrm{Re}_{s,1}^{0.6}\ \mathrm{Pr}_{s,1}^{1/3}\ (\frac{k_{s,1}}{d_o})$$

Substituting,

$$Nu_{s,1} = 0.2\ \mathrm{Re}_{s,1}^{0.6}\ \mathrm{Pr}_{s,1}^{1/3}$$

where

$$\mathrm{Re}_{s,1} = \frac{G_{s,1}\ d_o}{\mu_t}$$

$$\mathrm{Pr}_{s,1} = \frac{\mu_{s,1}\ c_{p-t}}{k_t}$$

Substituting, and canceling out constants,

$$(h_{sc,1})_{test} = \frac{[({}^{m_{d,o}}/_{\mu_t})^{0.6}\ ({}^{\mu_t\ c_{p,t}}/_{k_t})^{1/3}\ k_t]_{test}}{[({}^{m_{d,o}}/_{\mu_t})^{0.6}\ ({}^{\mu_t\ c_{p.t}}/_{k_t})^{1/3}\ k_t]_{design}}\ (h_{sc,1})_{design}$$

$$(h_{sc,1})_{test} = \left(\frac{m_{d,o,test}}{m_{d,o,design}}\right)^{0.6} \left(\frac{\mu_{t,design}}{\mu_{t,test}}\right)^{0.267} \left(\frac{k_{t,test}}{k_{t,design}}\right)^{1/3}\ (h_{sc,1})_{design}$$

Therefore, the shell-side convection film coefficient in the subcooling zone under test conditions is computed by correcting the design value for the differences in mass flow rate, viscosity, and thermal conductivity between the design and test values. Therefore, substituting the appropriate values from Table A-1,

$$(h_{sc,1})_{test} = \left(\frac{m_{d,o,test}}{m_{d,o,design}}\right)^{0.6} \left(\frac{\mu_{t,design}}{\mu_{t,test}}\right)^{0.267} \left(\frac{k_{t,test}}{k_{t,design}}\right)^{1/3}\ (h_{sc,1})_{design} = 4{,}231\left(\frac{W}{m^2\ {}^oC}\right)$$

The shell-side convection film resistance is, therefore,

$$r_{sc,1} = \frac{1}{h_{sc,1}} = 0.000231\left(\frac{m^2\ {}^oC}{W}\right)$$

Since the subcooling zone outlet temperature is unknown, $LMTD_2$ is unknown, so the equation presented previously for computing the condensing zone shell-side convection film coefficient is not applicable. However, Thomas (Reference 1) suggests the following equation in this event:

$$h_{sc,2} = \frac{1}{0.06834(T_{sat} - 5.0)^{-0.8912}} \left(\frac{Btu}{hr\ ft^2\ {}^oF} \right)$$

This equation expressed in SI units becomes

$$h_{sc,2} = \frac{5.67826}{0.06834(1.8T_{sat} + 27)^{-0.8912}} = 8{,}677 \left(\frac{W}{m^2\ {}^oC} \right)$$

and

$$r_{sc,2} = \frac{1}{h_{sc,2}} = 0.000115 \left(\frac{m^2\ {}^oC}{W} \right)$$

The fouling ratio required to yield the heat transfer rate measured during the test is determined by trial and error to be FR=0.526 as follows:

$$U_1 = \frac{1}{r_{sc,1} + r_w + r_{tc} + FR(r_{sf} + r_{tf})_{design}} = 2{,}543 \left(\frac{W}{m^2\ {}^oC} \right)$$

$$NTU_1 = \frac{U_1\ A_{s,1}}{C_t} = 0.144$$

$$P_1 = \frac{1 - e^{[-NTU_{t,1}(1-R_1)]}}{1 - R_1\ e^{[-NTU_{t,1}(1-R_1)]}} = 0.0656$$

$$Q_1 = C_t P_1 (T_{sat} - t_1) = 2{,}032 \left(kJ/s \right)$$

$$t_2 = t_1 + \frac{Q_1}{C_t} = 64.15\ {}^oC$$

$$U_2 = \frac{1}{r_{sc,2} + r_w + r_{tc} + FR(r_{sf} + r_{tf})_{design}} = 4{,}112 \left(\frac{W}{m^2\ {}^oC} \right)$$

$$NTU_2 = \frac{U_2\ A_{s,2}}{C_t} = 2.856$$

$$P_2 = 1 - e^{-NTU_2} \quad = 0.942$$

$$Q_2 = C_t P_2 (T_{sat} - t_2) = 27{,}274 \left({kJ}/{s} \right)$$

Therefore,

$$Q_{test} = C_t \, (t_3 - t_1) = 29{,}306 \left({kJ}/{s} \right)$$
$$Q_{calc} = Q_1 + Q_2 = C_t \, P_1 \, (T_{sat} - t_1) + C_t \, P_2 \, (T_{sat} - t_2) = 29{,}306 \left({kJ}/{s} \right)$$

These results suggest that during this test, HX2 exhibited only 52.6% of the design fouling on the tube side and shell side of the feedwater heater.

CONCLUSIONS AND RECOMMENDATIONS

A feedwater heater analysis approach is presented based on the effectiveness method that has been validated against the feedwater heater design or baseline conditions from the feedwater heater manufacturer presented in Appendix A. Excellent agreement exists between the heat transfer rates (duties) and the tube-side outlet temperatures calculated for the subcooling and condensing zones. However, discrepancies of as much as 0.5 ^{o}C exist between the calculated shell-side outlet temperatures and those provided by the manufacturer.

Algorithms for the for the individual resistances to heat transfer for the tube-side and shell-side convection films and the overall coefficient of heat transfer for the subcooling zone and the condensing zone are presented. These algorithms permit the user to predict the performance of the feedwater heaters at other than baseline conditions and to evaluate the performance that the feedwater heaters exhibited during thermal performance tests. Agreement between the resistances to heat transfer due to the tube-side convection film calculated by the Petukhov-Kirillov method and that provided by the manufacturer varies by 14% for HX2. For the subcooling zone, the resistance due to the shell-side convection film at baseline conditions is back-calculated from the manufacturer's overall heat transfer coefficient, so variations that exist on the tube side are also reflected on the shell side to force the calculated overall heat transfer coefficient to agree with the manufacturer's data. For the condensing zone, the resistance due to the shell-side convection film is calculated based on the procedure proposed by the BHMA, and agreement with the values shown by the manufacturer is excellent.

The user may evaluate the results of thermal performance tests by calculating the predicted performance of a feedwater heater at test conditions and calculating the "apparent" fouling resistance required to achieve the measured heat transfer rate. This "apparent" fouling resistance may be expressed as a fouling ratio between the "apparent" fouling resistance and the design fouling resistance.

REFERENCES

1. Thomas, L.C., *Heat Transfer-Professional Version*, second edition, Capstone Publishing Company, Tulsa, Oklahoma, 1999.

SYMBOLS

Symbol	Units	Description
A	m^2	Surface area
A_t	m^2	Tube-side flow area
C	$kJ/(s\ ^oC)$	Capacity rate
c_p	$kJ/(kg\ ^oK)$	Specific heat
d_I	m	Tube inside diameter
d_o	m	Tube outside diameter
DOA_{calc}	^oC	Calculated drain outlet approach
EMTD	^oC	Effective mean temperature difference
f		fanning friction factor
G	$kg/(\ m^2\ s)$	Mass flux
h	$W/(m^2\ ^oC)$	Heat transfer coefficient
h_{sc}	$W/(m^2\ ^oC)$	Shell-side convection film heat transfer coefficient
$h_{sc,ideal}$	$W/(m^2\ ^oC)$	Ideal shell-side convection film heat transfer coefficient
$h_{sc,test}$	$W/(m^2\ ^oC)$	Test shell-side convection film heat transfer coefficient
J_T		Total correction factor for shell-side convection film heat transfer coefficient
k	$W/(s\ m\ ^oC)$	Thermal conductivity
LMTD	^oC	Log mean temperature difference
m	kg/s	Mass flow rate
$m_{d,I}$	kg/s	Mass flow rate, drains in
$m_{d,o}$	kg/s	Mass flow rate, drains out
m_e	kg/s	Mass flow rate, extraction steam
N		Number of tubes
NTU		Number of transfer units
Nu		Nusselt number
P		Effectiveness
Pr		Prandtl number
Q	kJ/s	Heat transfer rate
$Q_{calc.}$	kJ/s	Calculated heat transfer rate
Q_{test}	kJ/s	Test heat transfer rate
R		Capacitance ratio
Re		Reynolds number
r_{sc}	$(m^2\ ^oC)/W$	Shell-side condensate film resistance
r_{sf}	$(m^2\ ^oC)/W$	Shell-side fouling resistance
r_{tc}	$(m^2\ ^oC)/W$	Tube-side condensate film resistance
r_{tf}	$(m^2\ ^oC)/W$	Tube-side fouling resistance
r_w	$(m^2\ ^oC)/W$	Tube wall resistance

t_1	^oC	Tube-side temperature entering subcooling zone
t_2	^oC	Tube-side temperature exiting subcooling zone
t_3	^oC	Tube-side temperature exiting condensing zone
t_I	^oC	Tube-side temperature entering
t_o	^oC	Tube-side temperature exiting
T_o	^oC	Shell-side temperature exiting
T_{sat}	^oC	Shell-side saturation temperature
TTD_{calc}	^oC	Calculated terminal temperature difference
T_w	m	Tube wall thickness
U	$W/(m^2\ ^oC)$	Overall heat transfer coefficient
μ	kg/(m s)	Dynamic viscosity
υ	m^2/s	Kinematic viscosity

Subscripts

1	Subcooling zone (zone 1)
2	Condensing zone (zone 2)
s	Shell side
t	Tube side

Appendix A: FEEDWATER HEATER DESIGN CONDITIONS

Table A-1
Feedwater Heater Design Conditions

Design Parameter	Value	Units
Tubeside Flowrate	291.967	Kg/s
Tubeside Inlet Temperature	63.81	^OC
Tubeside Inlet Enthalpy	266.86	KJ/kg
Temp. of Tubeside Leaving Sub. Zone	65.24	^OC
Enthalpy of Tubeside Leaving Sub. Zone	272.84	KJ/kg
Tubeside Outlet Temperature	86.01	^OC
Tubeside Outlet Enthalpy	359.99	KJ/kg
Shellside Inlet Steam Flow	11.308	Kg/s
Shellside Pressure	65.50	Kpaa
Shellside Saturation Temperature	88.12	^OC
Shellside Inlet Steam enthalpy	2,604.33	KJ/kg
Shellside Inlet Drains Flow	10.898	Kg/s
Shellside Inlet Drains Temperature	91.56	^OC
Shellside Inlet Drains Enthalpy	383.51	KJ/kg
Shellside Outlet Temperature	69.37	^OC
Shellside Outlet Enthalpy	290.12	KJ/kg
Terminal Temperature Difference	2.11	^OC
Drain Cooler Approach	5.56	^OC

Total Condensing Surface Area	871	m^2
Effective Condensing Surface Area	867	m^2
Total Subcooling Surface Area	75	m^2
Effective Subcooling Surface Area	71	m^2
Total Surface Area	946	m^2
Total Effective Surface Area	938	m^2
Condensing Heat Transfer Rate	25,409	KJ/s
Subcooling Heat Transfer Rate	1,746	KJ/s
Total Heat Transfer Rate	27,155	KJ/s
Condensing LMTD	8.72	^{O}C
Condensing EMTD	8.72	^{O}C
Subcooling LMTD	12.24	^{O}C
Subcooling EMTD	11.56	^{O}C
Condensing Heat Trans. Coeff., Clean	3,871	$W/(m^2\ ^{O}C)$
Condensing Heat Trans. Coeff., Service	3,364	$W/(m^2\ ^{O}C)$
Subcooling Heat Trans. Coeff., Clean	2,640	$W/(m^2\ ^{O}C)$
Subcooling Heat Trans. Coeff., Service	2,125	$W/(m^2\ ^{O}C)$
Condensing Tubeside Fouling Resistance	0.000039	$(m^2\ ^{O}C)/\ W$
Condensing Tubeside Film Resistance	0.000106	$(m^2\ ^{O}C)/\ W$
Condensing Tube wall Resistance	0.000039	$(m^2\ ^{O}C)/\ W$
Condensing Shellside Film Resistance	0.000113	$(m^2\ ^{O}C)/\ W$
Condensing Shellside Fouling Resistance	0	$(m^2\ ^{O}C)/\ W$
Subcooling Tubeside Fouling Resistance	0.000039	$(m^2\ ^{O}C)/\ W$
Subcooling Tubeside Film Resistance	0.000114	$(m^2\ ^{O}C)/\ W$
Subcooling Tube wall Resistance	0.000039	$(m^2\ ^{O}C)/\ W$
Subcooling Shellside Film Resistance	0.000226	$(m^2\ ^{O}C)/\ W$
Subcooling Shellside Fouling Resistance	0.000053	$(m^2\ ^{O}C)/\ W$
Number of Tubes per HX	726	
Eff. Straight Length of Tubes Cond. Sec.	17.96	m
Eff. Straight Length of Tubes Sub. Sec.	0.75	m
Number of Baffles in Condensing Sec.	8	
Number of Baffles in Subcooling Section	2	
Tube Type	U bend	
Tube Outside Diameter	1.095	cm
Tube Gauge	20	
Tube Pitch	2.38	cm
Shell Outside Diameter	1.600	m
Tube Wall Thickness	0.09525	cm
Shell Thickness	1.270	cm
Tube Material	439 SS	

Table A-2

Test Results for HX2

Design Parameter	Value	Units
Tubeside Flowrate	297.900	kg/s
Tubeside Inlet Temperature	62.53	^{o}C
Tubeside Outlet Temperature	86.00	^{o}C
Shellside Inlet Steam Flow	12.211	kg/s
Shellside Pressure	63.29	kPaa
Shellside Saturation Temperature	87.33	^{o}C
Shellside Inlet Drains Flow	11.527	kg/s
Shellside Inlet Drains Temperature	93.36	^{o}C
Shellside Outlet Temperature	70.33	^{o}C
Terminal Temperature Difference	1.33	^{o}C
Drain Cooler Approach	7.81	^{o}C
Total Duty	29,324	kJ/s

Appendix B: FEEDWATER HEATER ANALYSIS MODEL

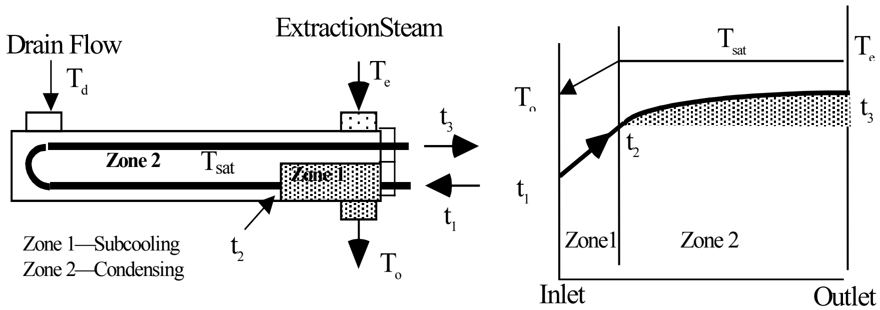

Figure B-1