

Prediction of heat transfer in the kraft recovery boiler superheater region

Mashiro Kawaji, X. H. Shen, Honghi Tran, S. Esaki, and C. Dees

ABSTRACT: *Accurate prediction of tube surface temperature is important for determining whether accelerated superheater corrosion will occur in kraft recovery boilers. A heat transfer model, which combines an iterative numerical solution scheme with a more realistic flue-gas flow field obtained from a CFD code FLUENT, has been developed to predict the tube, steam, and flue-gas temperatures in the superheater region. Results of tests performed for a recovery boiler show that temperatures predicted by the model are in good agreement with actual temperatures measured using thermocouples. The presence of a large flue-gas recirculation zone above the bullnose is shown to have an adverse effect on the heat transfer in the superheater region.*

KEYWORDS: *Energy transfer, furnaces, heat transfer, heating equipment, models, recovery furnaces, superheaters, temperature.*

Severe corrosion of superheater tubes has been experienced in a number of kraft recovery boilers (1, 2). The corrosion typically occurs at the bottom tube bends of superheater platens, where the tube surface temperature exceeds the first melting point of the fireside deposits. There are many parameters that can influence the tube surface temperature, including steam tempera-

ture and flow rate, deposit thickness and characteristics, and the rates of convective and radiative heat transfer from the flue gas and furnace cavity. These parameters are, in turn, strongly affected by the superheater design and boiler operation.

Several models have been developed which can be used to obtain the temperature distributions in the superheater region (3, 4). These mod-

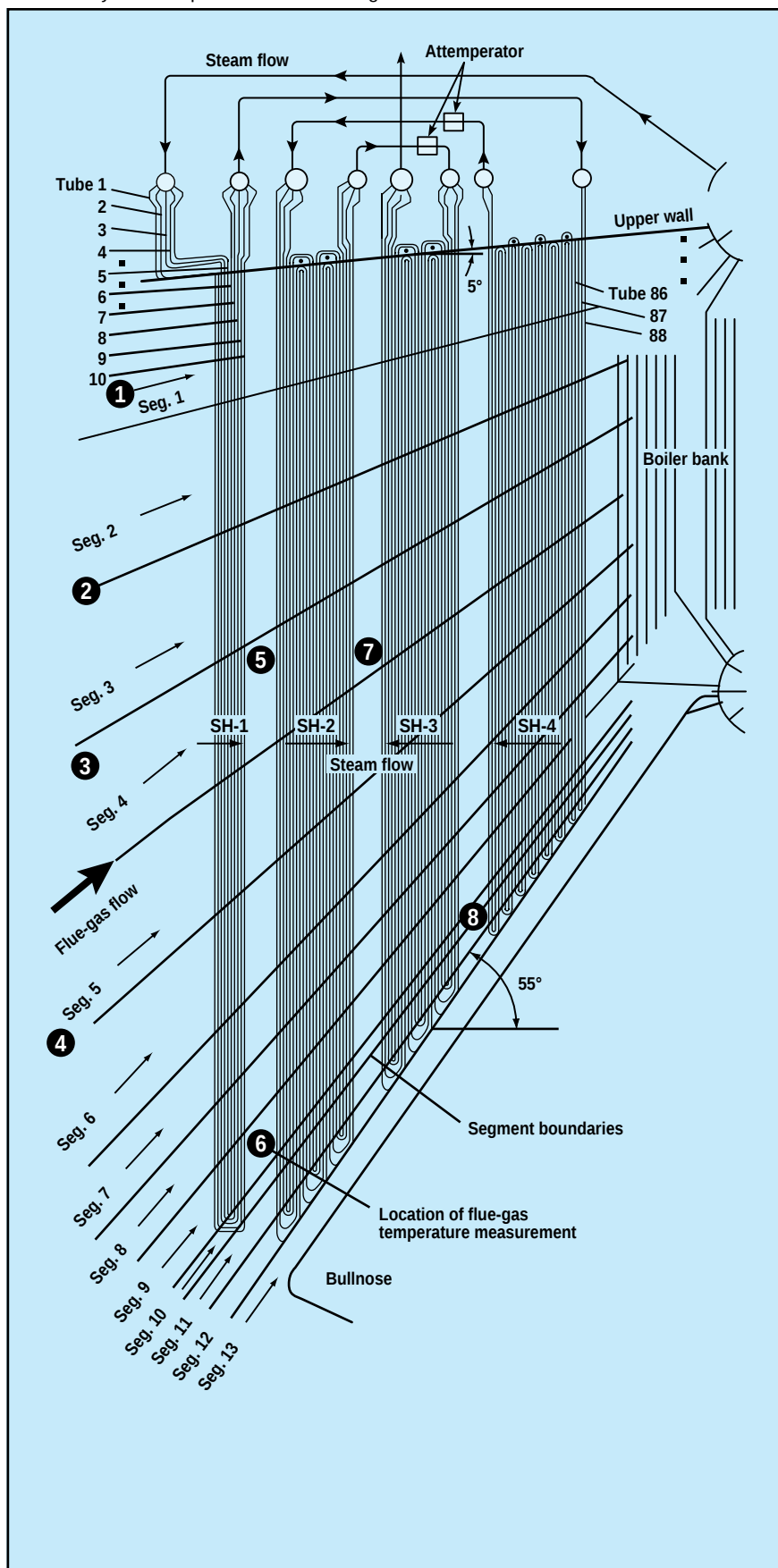
els, however, either cover the entire boiler and place more emphasis on the combustion chamber than on the superheaters (3) or require the flue-gas flow field and temperature distribution as input data to the program (4).

Our objective, therefore, is to develop a heat transfer model that can be used to predict the temperatures of the tube surface, steam, and flue gas at various locations in the superheater region of a recovery boiler, as a function of superheater design and boiler operation. The first version of the model, which also serves as the base model, has been described in detail by Esaki *et al.* (5). It was designed to accommodate variations in the superheater design and geometry, tube dimensions, steam flow direction, physical properties of the deposit, flue gas, and steam at various locations in the superheater region. Validation of the base model was carried out through a field study in a recovery boiler at Willamette Industries in Hawesville, KY. The results showed that although the model was successful in predicting steam temperatures, there was a significant over-prediction of the flue-gas temperatures in the middle region of the superheater platens above the bullnose. The major weakness of the base model was that it did not take into account the presence of the recirculation zone above the bullnose, which greatly influences the heat transfer (6).

This paper describes the main features of the base model and the im-

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1. Recovery boiler superheater tube arrangement at Willamette mill



provements made to include the effect of the recirculation zone on heat transfer.

Heat transfer model

Base model

In the base model, all the tubes in the central superheater platens of a recovery boiler were divided into many small segments, as shown in **Fig. 1**. The mass and heat balance equations for both steam and flue gas were solved for each tube segment. The flue gas was assumed to have a unidirectional flow parallel to the segment, and steam could flow either parallel or counter to the direction of the flue-gas flow, so an iterative solution scheme had to be used to solve the mass and heat balance equations. This scheme was found to have good convergence characteristics and to be flexible enough to accommodate a fairly complicated superheater tube layout with attemperators between the tube platens (5).

In a given computational cell, a tube segment is covered by a deposit layer, and all the heat transfer caused by both convection and radiation in both the inside and outside of a tube was modelled. On the outside, the convective heat transfer coefficient was calculated assuming a forced flow of flue gas across a tube bank of certain pitch-to-diameter ratios, and the radiative heat transfer between flue gas and deposit surface was calculated according to the surrounding flue-gas volume. The gas absorptivity and emissivity were estimated using the mean beam length and partial pressures of main gas components: CO_2 , H_2O , N_2 , and O_2 . The composition of the flue gas was estimated from the composition of the as-fired black liquor and the average excess O_2 at which the boiler was operated. Details of the calculations can be found elsewhere (5).

Improved model

The major feature of the improved model is the incorporation of a more realistic flue-gas flow field (**Fig. 2**) into the base model. The flue-gas flow field in this case is a two-dimensional flow field integrated from a three-dimensional flow field obtained using a CFD code FLUENT (7). The distributions of the steam temperature and overall heat transfer coefficient in the individual superheater platens obtained from the base model predictions are employed as the thermal boundary conditions for FLUENT, as shown in **Fig. 3**. For the conditions under study, the flue-gas flow field as predicted by FLUENT includes a large recirculation zone existing in the region above the bullnose, which is very different from the parallel flow assumed in the base model. Incorporating the flue-gas flow field in the recirculation zone into the base model makes the predictions much more realistic and accurate.

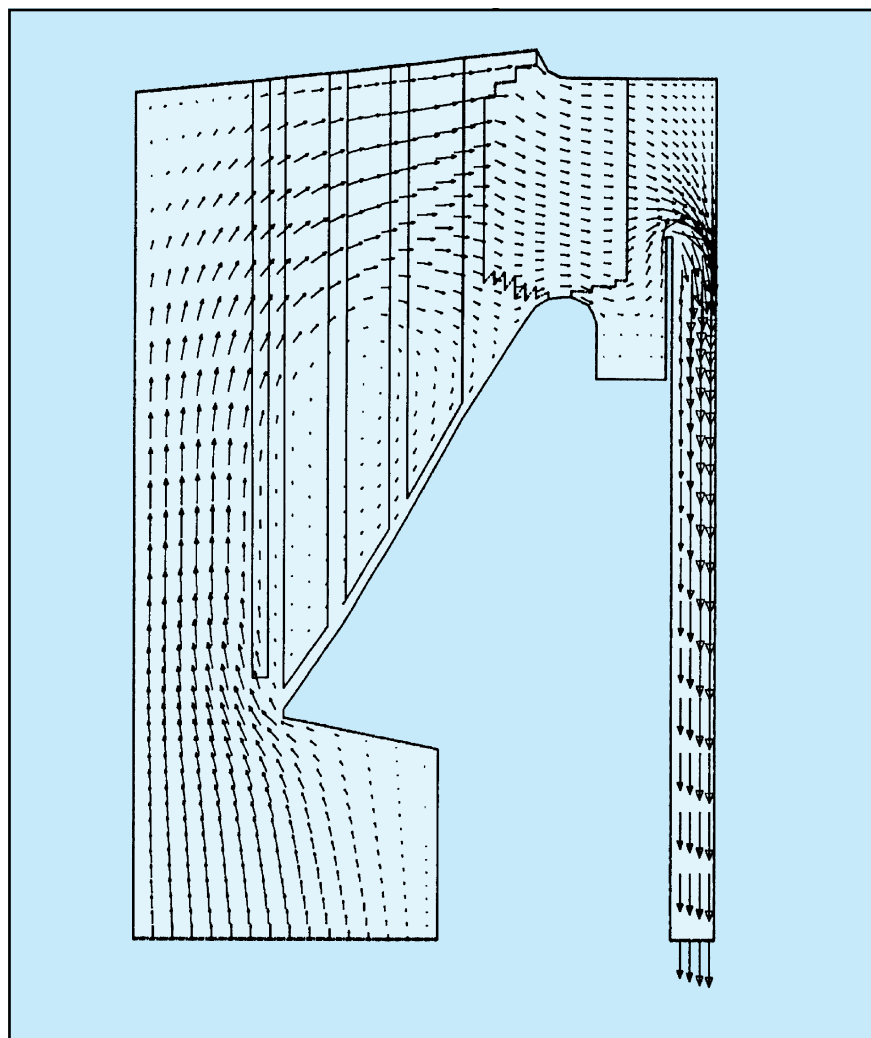
As shown in **Fig. 4**, we define U_{ij} and $U_{i+1,j}$ as the flue-gas velocity components parallel to the segment, and V_{ij} and V_{ij+1} as the vertical velocity components at the boundaries of the cell (i,j). Using the two-dimensional flue-gas velocity components, U and V , the mass flow rates of flue gas crossing the cell boundaries can be calculated as $W_x = \rho U A_x$ and $W_y = \rho V A_y$, respectively. Here, the flue-gas density, ρ , must be evaluated at an appropriate cell boundary temperature, and A_x and A_y are the cross-sectional flow areas. The cell inlet temperature of the flue gas is then given by,

$$T_{in}(i,j) = [W_x(i,j)T_x(i,j) + W_y(i,j)T_y(i,j)] / [W_x(i,j) + W_y(i,j)]$$

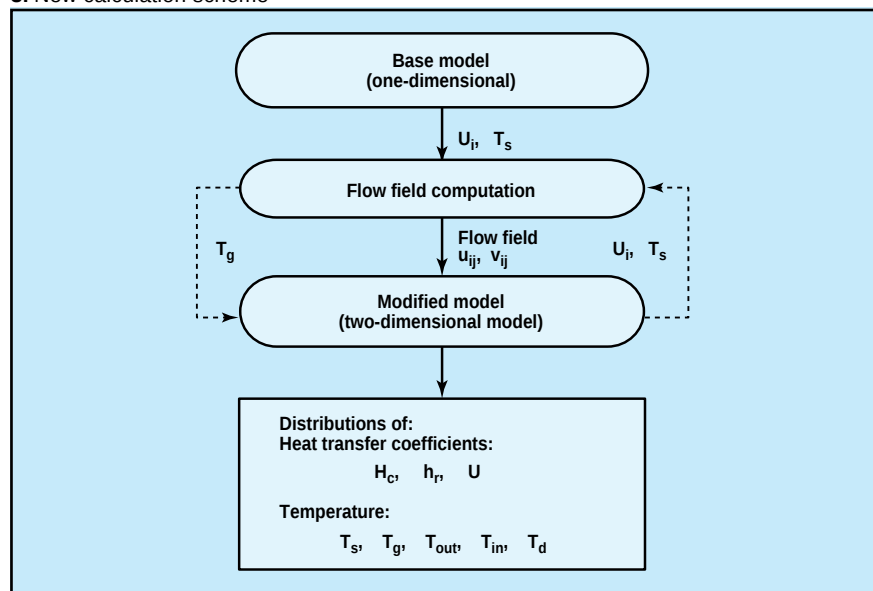
where T_x and T_y are the flue-gas temperatures at the cell inlet boundaries. Since the total inlet and outlet gas flow rates must be equal to conserve mass,

$$W_g = W_x(i,j) + W_y(i,j) = W_x(i+1,j) + W_y(i,j+1)$$

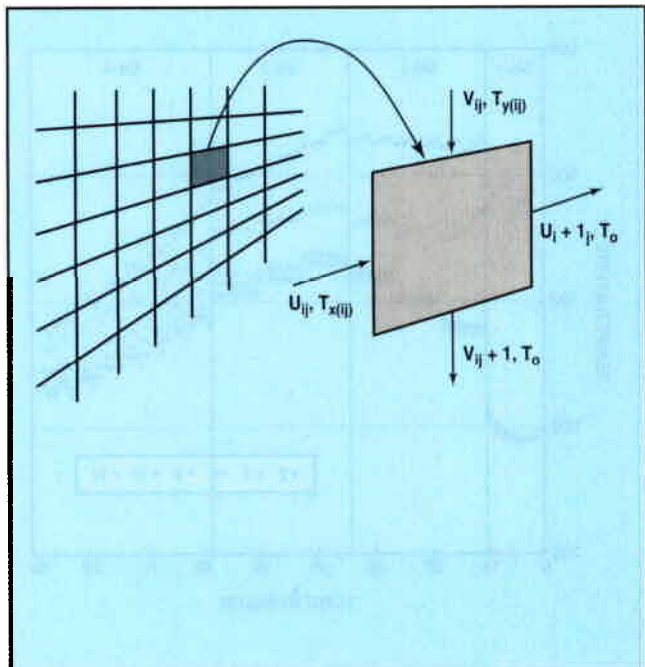
2. Flue-gas flow field predicted by FLUENT



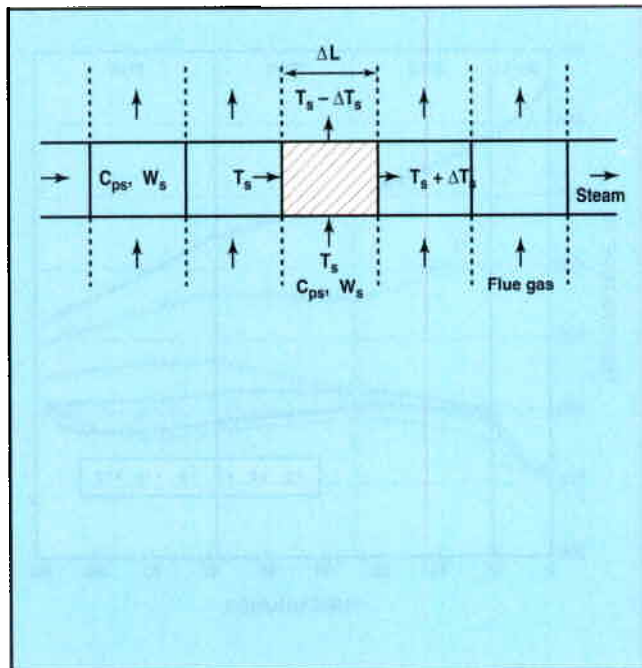
3. New calculation scheme



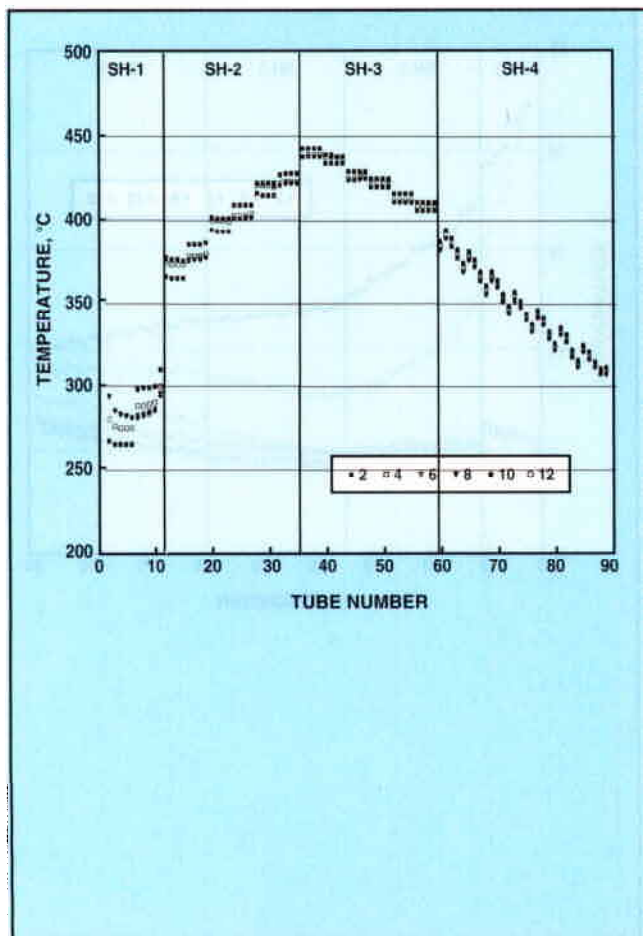
4. Two-dimensional flow of flue gas into and out of a computational cell



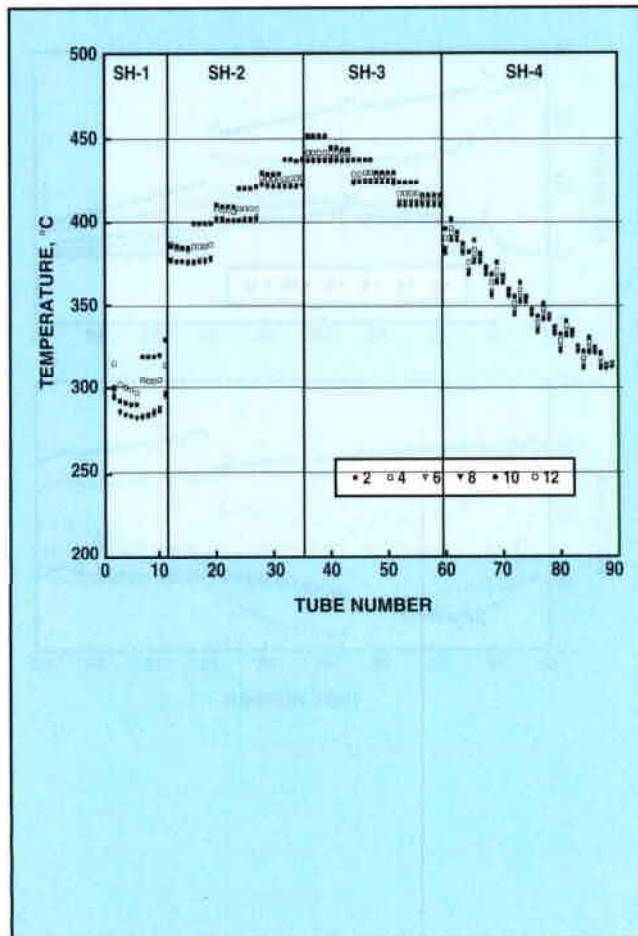
5. Heat balance for a computational cell



6. Predicted superheater steam temperature distribution

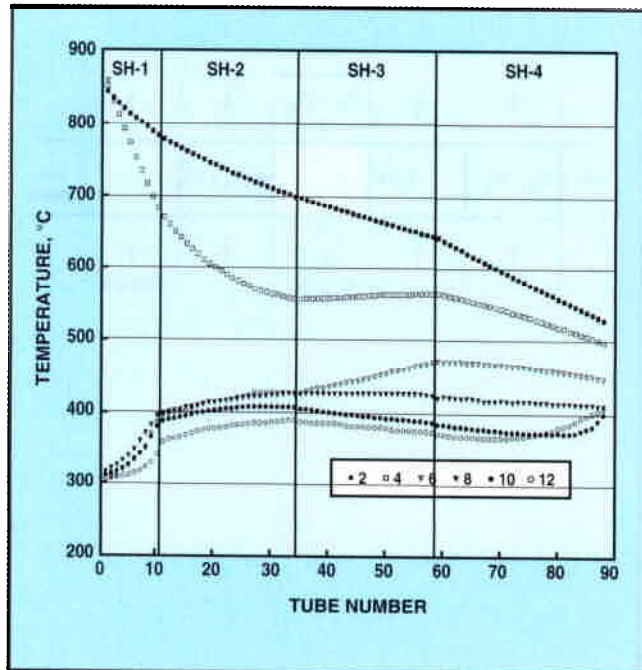


7. Predicted tube outer surface temperature distribution

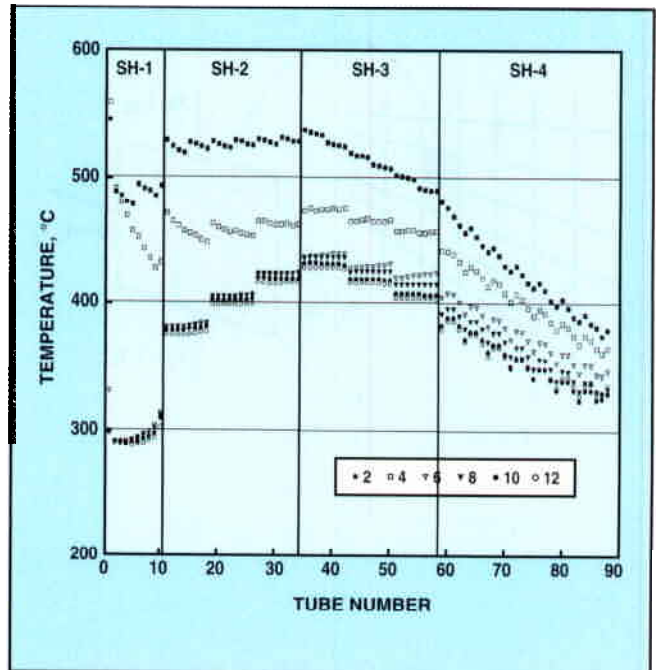


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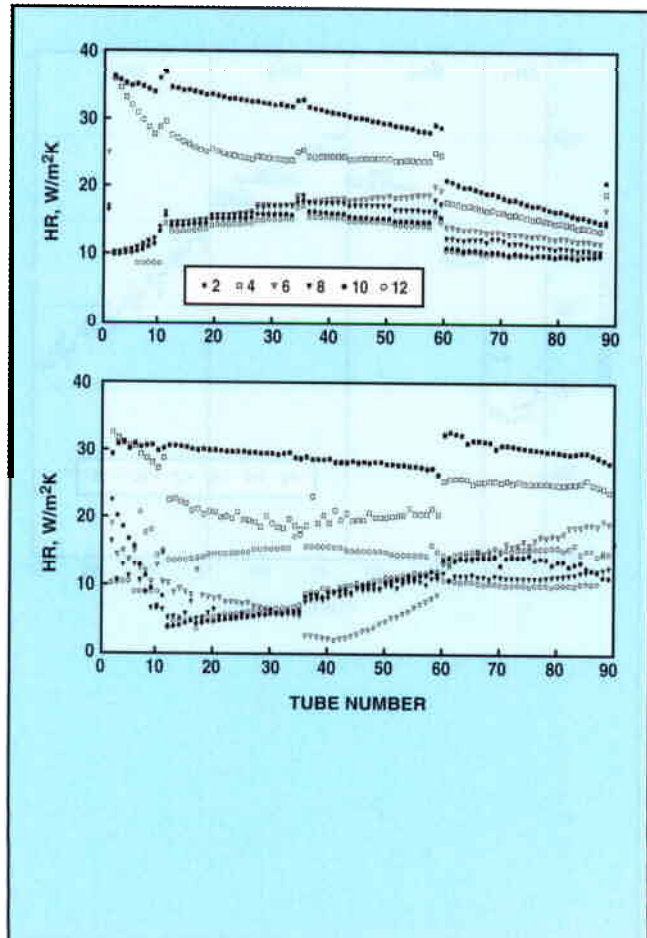
8. Predicted flue-gas temperature distribution



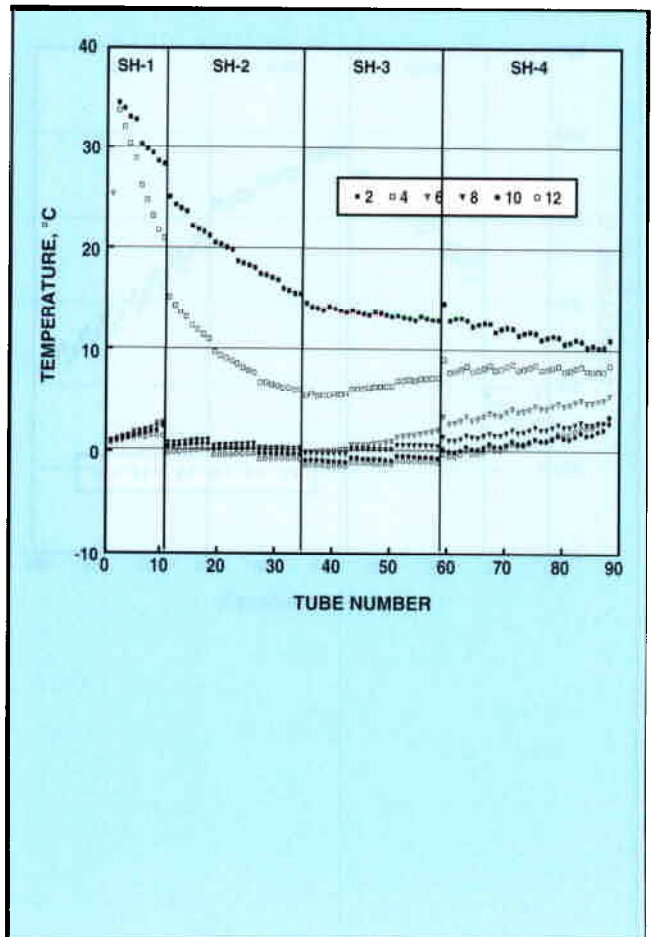
9. Predicted temperatures on the superheater deposit surface



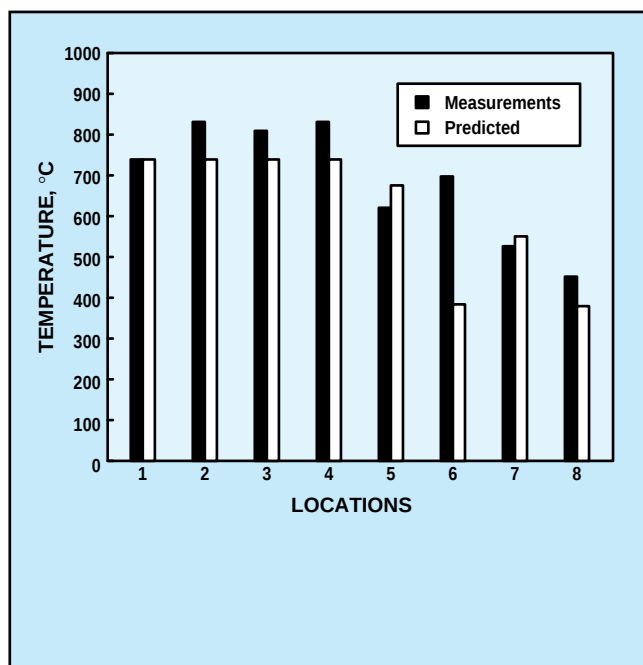
10. Predicted distributions of convective and radiative heat transfer coefficients



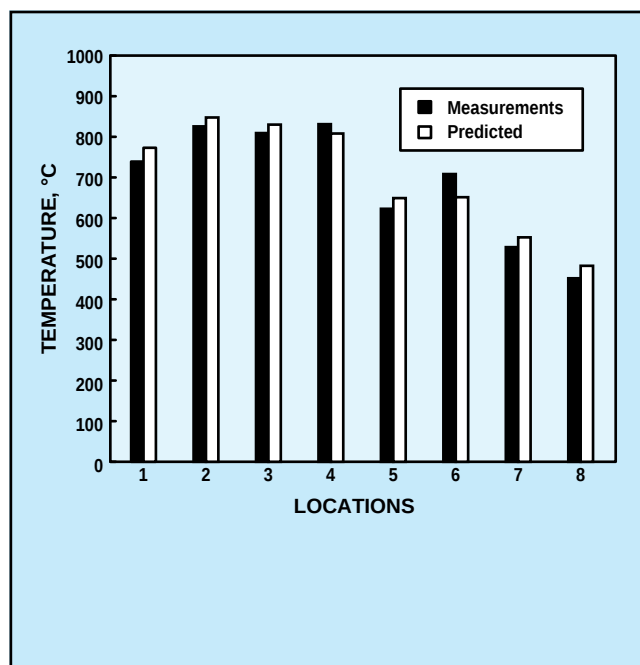
11. Predicted gas-to-steam heat transfer rates



12. Comparison of predicted and measured flue-gas temperatures (see Fig. 1 for measurement locations)



13. Comparison of measured flue-gas temperatures with the FLU-ENT predictions (see Fig. 1 for measurement locations)



The outlet flue-gas temperature, $T_{out}(i,j)$, is then given by a heat balance, as shown in Fig. 5,

$$C_{pg} W_g [T_{out}(i,j) - T_{in}(i,j)] = C_{ps} W_s \Delta T_s$$

where the right-hand side indicates the change in the energy of the superheated steam across the computational cell, and the heat diffusion caused by the turbulence gas mixing on the cell boundaries is ignored. The rate of heat transfer between the flue gas and steam is calculated using an overall heat transfer coefficient computed for each cell by accounting for the convection of the inner and outer flows, thermal radiation from the flue gas to the deposit surface, and heat conduction across the tube wall and deposit layer. The details of the correlations used to compute the overall heat transfer coefficient are given elsewhere (5).

The heat transfer from flue gas to the bullnose at the bottom of the superheater has also been added to the model. For the tube-side heat transfer, saturated boiling of high-pressure water is assumed and Levy's correlation (8) is used,

$$q = 0.00283 P^{4/3} \Delta T_{sat}^3$$

where

q = boiling heat flux, W/m^2

P = saturated water pressure, kPa

ΔT_{sat} = tube wall superheat, $T_{wall} - T_{sat}$, in $^{\circ}C$.

On the outside of a tube in the bullnose, radiative heat transfer is modeled in the same manner as for a superheater tube, but the convective heat transfer coefficient is calculated from a heat transfer correlation for convection on a flat plate of length L .

$$Nu = h_m L / k_g = 0.037 Re^{0.8} Pr^{1/3}$$

where

Re = VL/v_g

h_m = average convective heat transfer coefficient

Pr = Prandtl number of the flue gas

V = the flue-gas velocity component parallel to the bullnose

k_g = thermal conductivity

v_g = kinematic viscosity of the flue gas.

The heat flux from the flue gas to the saturated water is calculated using an overall heat transfer coefficient formulated in terms of the radiative and convective heat transfer coefficients in the same manner as for a superheater tube.

The improved model is solved in an iterative manner by first assuming a uniform flue-gas temperature field and performing the calculations from the first tube on the left to the last tube on the right. The obtained flue-gas and steam temperatures are then used in the second iteration, and so on, until convergence is achieved.

Case description and related assumptions

The present model has been applied to a recovery boiler at the Willamette Hawesville mill. As shown in Fig. 1, the boiler has four superheater tube banks referred to as Superheater No.1 (SH-1) through No. 4 (SH-4). The saturated steam from the boiler bank flows through SH-1, SH-4, SH-2, and then SH-3. The steam flow is parallel to the flue-gas flow in SH-1 and SH-2, while it runs counter to

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the flue-gas flow in SH-3 and SH-4. There are two attemperators to reduce the steam temperature located between SH-4 and SH-2 and between SH-2 and SH-3.

The tube bank SH-1 consists of 22 platens, which are spaced 0.352 m apart across the width of the boiler; SH-2 and SH-3 are set up the same way. Each platen of SH-1 has five tubes, with each tube making two passes before steam is discharged to the outlet header; the platen of either SH-2 or SH-3 has four tubes, and each tube has six passes. SH-4 consists of 43 platens, and the spacing between the two adjacent platens is 0.176 m from tube to tube. Each platen consists of two tubes; one tube makes 14 passes while the other makes 16 passes. All the superheater tubes here have a 44.5 mm o.d. and 4.6 mm wall thickness.

For computational purposes, the central superheater platens are selected, and a tube number is designated for each vertical pass of the superheater platens from the left to the right side of the upper boiler. Thus, the left-most tube pass in SH-1 is called tube No. 1 and is connected to tube No. 10 at the bottom of the platen through the outermost bend, while tube No. 5 is connected to tube No. 6 through the innermost bend, and so on. To incorporate the thermal radiative effects from the furnace cavity, the bottom segment (Segment No. 11 in Fig. 1) of tube No. 1 in SH-1 and that (Segment No. 12) of tube No. 11 in SH-2 are assumed to receive direct radiation from the lower furnace with a temperature of 1200°C. Other major assumptions made in the calculations are:

1. Deposits are formed uniformly on all tubes in the superheater (i.e., same thickness, morphology, and thermal conductivity), and the average deposit thickness is assumed to be 6.0 mm.
2. There is no steam leakage in all the superheater platens across

the boiler from side to side. Within a platen, the steam flow rate is equally distributed to individual tubes.

3. There is no loss in steam pressure due to friction in the tubes; that is, steam inlet and outlet pressures are the same.
4. Radiative heat transfer from flue gas to a superheater tube depends on the volume of the flue gas surrounding the tube; i.e., the radiation from flue gas to the tubes located at the outer edges of each platen is larger than that to the other tubes. The deposits' surface emissivity is taken as 0.7.

The actual inlet and outlet superheated steam temperatures of SH-2 and SH-3, and flue-gas temperatures at eight locations, were measured using thermocouples at various liquor firing loads. The steam flow rate, water flow rate for attemperation, pressure of the boiler bank, air flow rate supplied to the boiler, and O₂ concentration of the outlet flue gas were also measured.

Results and discussion

Steam, tube surface, and flue-gas temperatures

The distributions of the superheated steam temperature and outer-tube surface temperature are shown in **Figs. 6 and 7**, respectively. The predicted results are similar to those of the base model. However, the flue-gas temperature distribution shown in **Fig. 8** is significantly different from that predicted by the base model. The effect of recirculation just above the bullnose on the flue-gas temperature is clearly evident as the recirculating flue gas at the lower elevations (fifth segments and below) flows back from the superheater exit with a temperature close to or lower than that of the superheated steam in the tubes.

As a result of the wide variation in the flue-gas temperature with el-

evation, the deposit surface temperature for a given tube also shows a large variation (**Fig. 9**).

Heat transfer coefficients

Because of the low flue-gas velocities and temperatures in the recirculation zone, both the convective and radiative heat transfer coefficients are substantially reduced, as shown in **Fig. 10**. The overall heat transfer coefficient is also lower in the recirculation zone.

Figure 11 shows the heat transfer rates. Cooling of the superheated steam occurs in some parts of the recirculating zone, where heat is transferred from the steam to the flue gas by both convection and radiation. From a heat transfer point of view, the presence of the recirculating zone is not desirable due to the reduction in the effective heat transfer area of the superheater. For the recovery boiler examined, it appears that even if the lengths of the superheater platens are reduced to approximately half of the present lengths, there may be little change in the superheater performance. The reduction of the platen length, however, could result in substantial savings in capital and maintenance costs of the superheater.

Model validation

As a result of the modifications, the predicted flue-gas temperatures are in better agreement with the field data, as shown in **Fig. 12**. There are no predictions for the measurement at locations 1 through 4, since the measured flue-gas temperatures at those locations are used as part of the boundary conditions in the present model. The low flue-gas temperatures measured at locations 5 through 8 in the recirculation zone are now correctly predicted by the improved model. The only significant discrepancy between the predicted and measured values is at location 6. This is due to the unrealistic assumption made on the heat balance in each computational cell,

which ignores the effect of the violent turbulent mixing between the recirculating gas and the gas directly rising from the lower furnace just above the bullnose on the heat transfer.

However, this shortcoming can be overcome by first using the overall heat transfer coefficient and steam temperature distributions of the superheater platens predicted by the present model in FLUENT to obtain the flue-gas temperature field. The results as shown in **Fig. 13** indicate that more realistic predictions of the flue-gas temperatures are achieved.

For further improvement of the calculation procedure, we suggest that both the flue-gas velocity and temperature distributions predicted by FLUENT be used in the superheater code, and the superheated steam temperature and overall heat transfer coefficient predictions from the superheater code then be used in FLUENT, in an iterative manner. The calculation should ideally converge after a few iterations between these two codes.

Conclusions

An improved heat transfer model has been developed to predict the heat transfer rates in the superheater region of kraft recovery boilers. The model takes into account the flue-gas recirculation and heat loss to the bullnose. Predictions using the model for a recovery boiler at the Willamette Hawesville mill show good agreement with the actual measurements for both the steam and flue-gas temperatures. The analysis

of heat transfer in the superheater region indicates significantly cold flue gas in the recirculation zone can result in negative heat transfer from the flue gas to the steam. Minimizing such recirculation zones by improved configuration of the bullnose is expected to improve the heat transfer performance of the superheater significantly. 

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