

# Initial validation of a mathematical model of cold flow in a recovery boiler

Zia Abdullah, Martha Salcudean, Paul Nowak, Zhuzhi Xiao,  
Michael C. Savage, Craig Markovic, Vic C. Uloth, and Peter H. Thorn

---

**ABSTRACT:** *Air flow in a commercial recovery boiler was modeled mathematically for two sets of cold-flow conditions. Case 1 was for air flow through the primary and secondary ports, while Case 2 was restricted to air flow through the secondary ports only. There was no flow through the tertiary ports. The fully turbulent cold flow resulting from the jet interaction was computed using code developed specifically for modeling furnace air flow. A steady-state solution was obtained for Case 1, but an unsteady solution was obtained for Case 2. The computed air-flow patterns resembled real flow fields measured physically under the same set of conditions.*

**KEYWORDS:** *Air flow, air jets, flow measurement, mathematical models, openings, physical measurement, recovery furnaces, simulation.*

---

Recent developments in computational fluid dynamics promise greatly improved techniques to solve the complex transport phenomena that occur in kraft recovery boilers. These techniques, together with a dramatic reduction in the cost of computing, have enabled researchers to develop powerful mathematical models to simulate the air flow in recovery boilers.

Mathematical models can provide useful information about the flow patterns in a boiler before expensive modifications are made. In principle, all of the relevant details of the flow patterns can be obtained, allowing investigators to evaluate the effects of operating parameters on air flow.

Modeling of flows in recovery boilers is not an easy task. The task is

complicated by the large difference of scales, the presence of three dimensions, gross flow unsteadiness, and the effects of buoyancy and turbulence. These difficulties have been discussed in detail (1) and present considerable challenges to the modeler.

Previous attempts to model recovery boiler air flows (2-7) have used a variety of general-purpose finite-difference codes that were modified for recovery boiler applications. Work is in progress at the University of British Columbia (UBC) to develop a modeling capability specifically tailored to recovery boilers (8). Code is being developed to handle such boiler-specific modeling needs as aerodynamic complexity, convergence of the solution, and a high resolution of the flow. Preliminary three-dimensional computational results show dramatically improved convergence rates, with excellent resolution at the air-port level.

This article is a preliminary report on the collaborative research efforts at UBC to develop a mathematical model of air flows in a recovery boiler. We hope to demonstrate to the industry that mathematical modeling can provide useful information for boiler designers and operators. In this article, we compare physical measurements of cold-flow velocity distributions in a commercial recovery boiler with computed results from a mathemati-

---

Abdullah, research associate, Salcudean, associate v.p. research, Nowak, research associate, Xiao, research engineer, and Savage, research engineer, are associated with the Department of Mechanical Engineering, University of British Columbia, 2324 Main Mall, Vancouver, BC, Canada V6T 1Z4. Markovic, associate technical specialist, and Uloth, senior research engineer, are associated with PAPRICAN, Vancouver, BC, Canada. Thorn, senior process engineer, works for Weyerhaeuser Paper Co., WTC 2H 22, Tacoma, WA 98477.

## Simulated Boiler Flow

cal model. The mathematical models of the air flow yielded velocity distributions that were fairly representative of the real flow field. The results also show that mathematical modeling can provide much more detail of the flow than could possibly be obtained by physical measurements in an actual boiler. We will report future comparisons as the program proceeds.

### Physical modeling

Physical measurements of the cold flow were carried out in a B&W recovery boiler. **Figure 1** is a schematic of the boiler showing the primary, secondary, and tertiary air ports. Air flow in the boiler at the liquor-gun level was measured for two sets of flow conditions:

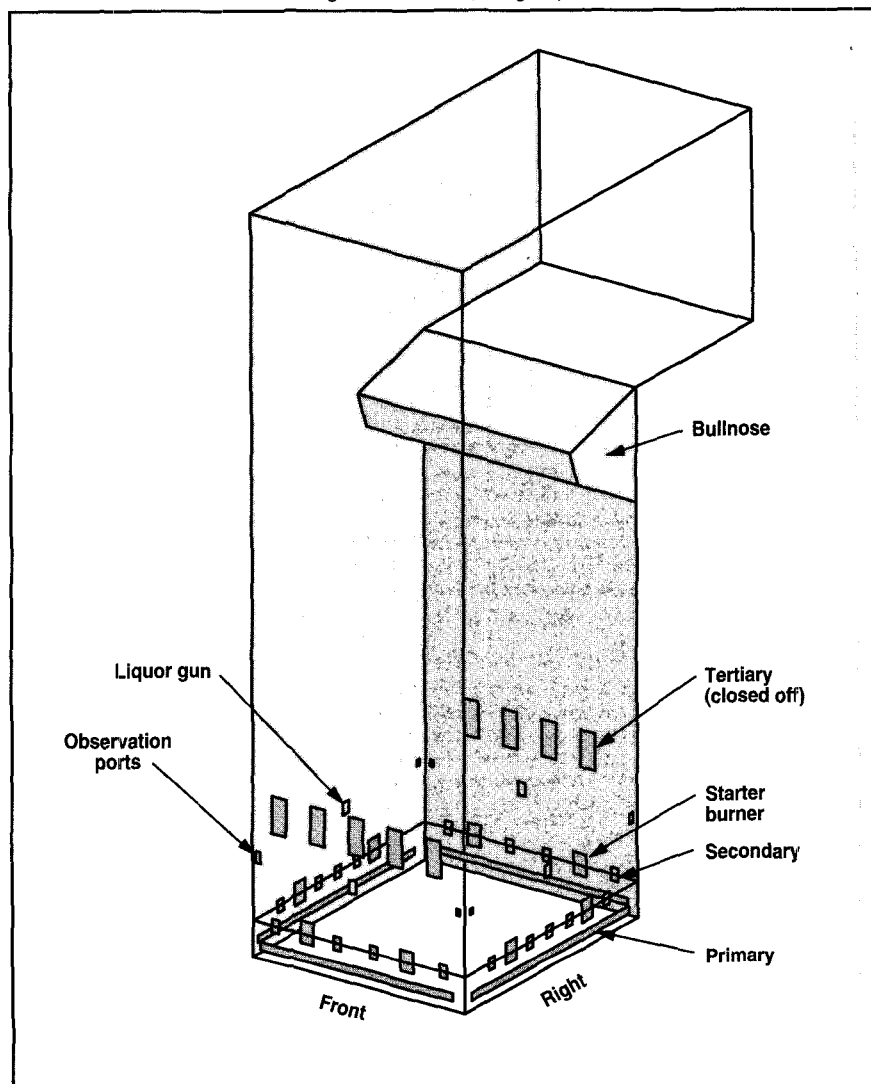
- Case 1—air flow through primary and secondary ports
- Case 2—air flow through secondary ports only.

There was no air flow through the tertiary ports.

Flow velocity was measured using hot-wire anemometers. Unfortunately, the anemometers arrived too late for calibration before the tests. We tested both instruments at the mill site using a cooling fan. Both gave different readings, so we labeled them "Probe 1" and "Probe 2" for future calibration. During the initial testing, we also observed that the measurements were direction dependent. The instruments showed different flows, depending on whether the large or small element was facing the flow. During the trials, the probes were positioned with the small element facing the floor of the furnace.

After the trials, both probes were evaluated in a wind tunnel. The indicated flows from Probe 1 were 30% higher with the large element on the windward side compared with the reverse orientation. Separate calibration curves, accurate to within  $\pm 5\%$ , were developed for both flow direc-

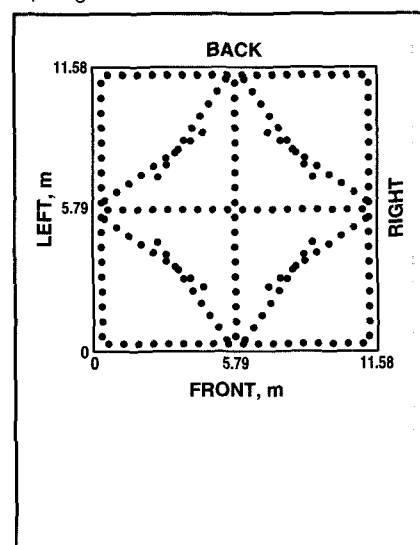
1. Schematic of the boiler showing the air and liquor-gun ports



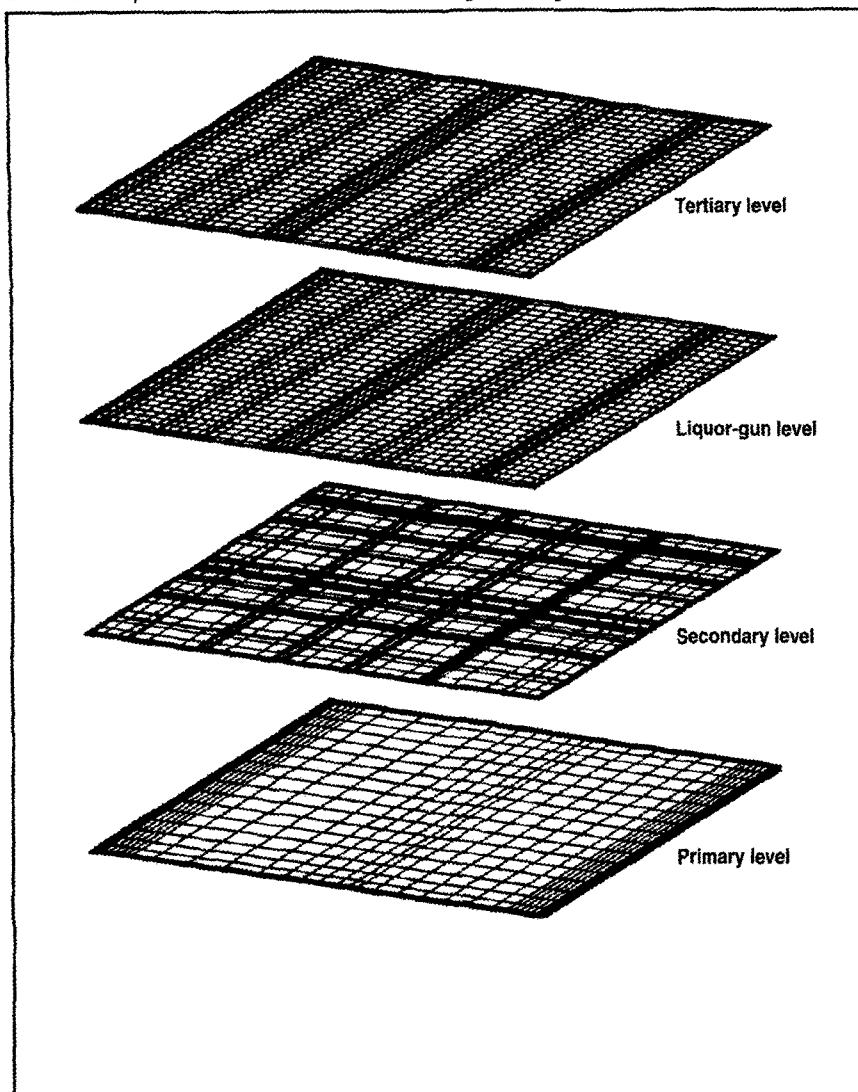
tions. Calibration data showed that measurements with Probe 2 were less accurate and repeatable than those with Probe 1. Overall accuracy of measurement with Probe 2 was within  $\pm 20\%$ . No correction for flow direction was appropriate for Probe 2.

Flow measurements in the boiler were made in a horizontal plane at the liquor-gun level. The hot-wire anemometer probes were attached to extension tubes and inserted through the liquor-gun openings and the observation ports at the liquor-gun level. **Figure 2** maps the measurement locations at the liquor-gun level. The direction of the flow was determined from the orientation of

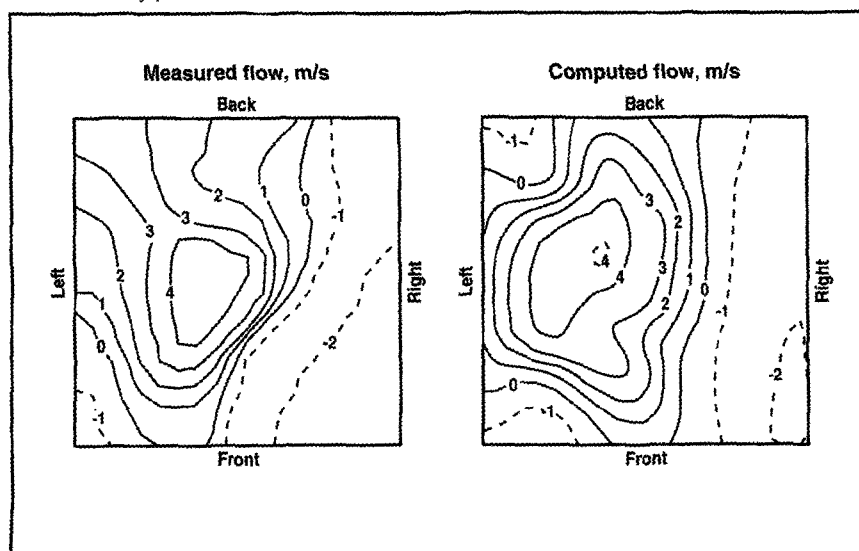
2. Locations of velocity measurement at the liquor-gun level



3. Horizontal planes of the three-dimensional segmented grid



4. Measured and computed air flow at the liquor-gun level. Case 1: inflow through primary and secondary ports.



brightly colored paper ribbons located near the hot-wire probes.

Pressure was measured at the primary and secondary windboxes in an effort to calculate the flow through the individual air ports. This information is necessary to establish the boundary conditions of the mathematical model of the flow in the boiler. To determine the flow distribution across an air-port opening, pitot-tube measurements were carried out for selected air ports where this was possible. Flow velocities in the primary and secondary air ducts were also measured using pitot tubes. This information was helpful in crosschecking the measurements at each air port.

To eliminate outlying data points resulting from the flow unsteadiness or instability, root-mean-square velocities were plotted as a function of position to create velocity profiles across the furnace for the various traverses conducted during both trials. Points that showed significant deviation from the smoothed profile were discarded.

It can be seen from Fig. 2 that velocity could not be measured in some regions at the liquor-gun level. Velocities in these regions were interpolated on the basis of neighboring velocity profiles, and by comparing the total integrated upflow at the liquor-gun level with the measured total air flow in the primary and secondary ducts. Agreement between the total flow through the ducts and the integrated upflow at the liquor-gun level was within  $\pm 30\%$ . We consider this level of agreement to be quite good, considering the sources of errors, i.e., the large scale of the measurement volume, the sparse regions where the measurements were taken, and the flow unsteadiness.

All the physical cold-flow data (boundary conditions and velocity measurements in the furnace cavity) were assembled before proceeding to the mathematical modeling.

# Simulated Boiler Flow

## Mathematical modeling

### Cold flow

Cold flow in recovery boilers can be described mathematically by the differential equations expressing the mass and momentum conservation of the fluid. Turbulence is modeled by two differential transport equations, one for the turbulence kinetic energy,  $\kappa$ , and the other for the turbulence energy-dissipation rate,  $\epsilon$ .

The time-averaged equations describing the conservation of mass and momentum can be written in Cartesian tensor notation (9) as follows.

Mass conservation equation

$$(\partial/\partial x_j)(\rho u_j) = 0 \quad (1)$$

Momentum conservation equation

$$\begin{aligned} (\partial/\partial x_j)(\rho u_j u_i) = & (-\partial P/\partial x_i) \\ & + \{[(\partial/\partial x_j)\mu_{\text{eff}}][(\partial u_i/\partial x_j)(\partial u_j/\partial x_i)]\} + \rho g_i \end{aligned} \quad (2)$$

where

$x$  = coordinate

$u$  = velocity

$\rho$  = density

$P$  = pressure

$g$  = gravitational acceleration

$\mu_{\text{eff}}$  = effective viscosity.

Boussinesq's eddy-viscosity concept is used to substitute for the turbulent stresses. The effective viscosity,  $\mu_{\text{eff}}$ , is given by Eq. 3.

$$\mu_{\text{eff}} = \mu + \mu_t \quad (3)$$

where

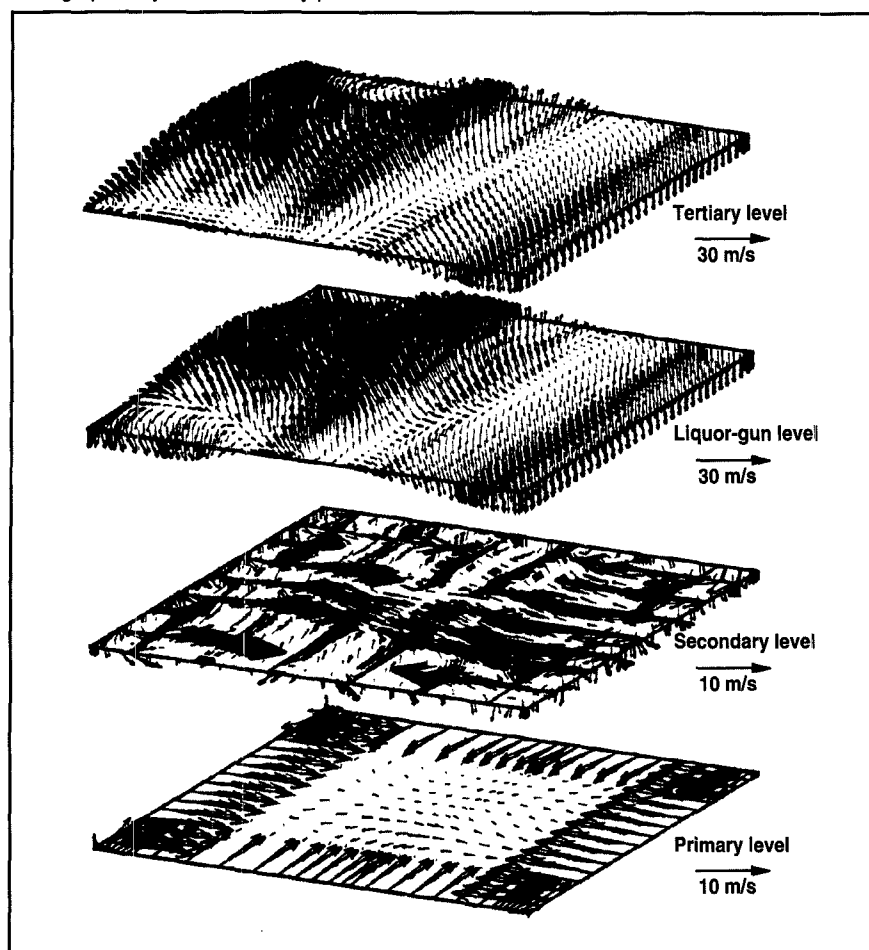
$\mu$  = molecular viscosity

$\mu_t$  = turbulent viscosity.

The two-equation  $\kappa$ - $\epsilon$  turbulence model is employed to characterize the turbulent flow (10).

The governing differential equations are integrated over three-dimensional control volumes. After application of Gauss's divergence theorem, integral equations for the convective and diffusive fluxes across the control volume faces are obtained (11, 12). The fluxes across the con-

5. Three-dimensional velocity distribution at selected horizontal planes. Case 1: inflow through primary and secondary ports.



trol-volume boundaries, in terms of the nodal values of the dependent variable, are obtained using the power-law differentiation scheme (11). The relaxation procedure and solution techniques have been described elsewhere (1, 8). A segmented grid was used to optimize the distribution of the grid nodes in the computational domain.

### Computational domain and boundary conditions

The front, back, left, and right boundaries of the computational domain represent the walls of the boiler. The computational model simulates all of the primary and secondary air ports in the boiler, including the starter-burner ports. The rows of primary air ports at each

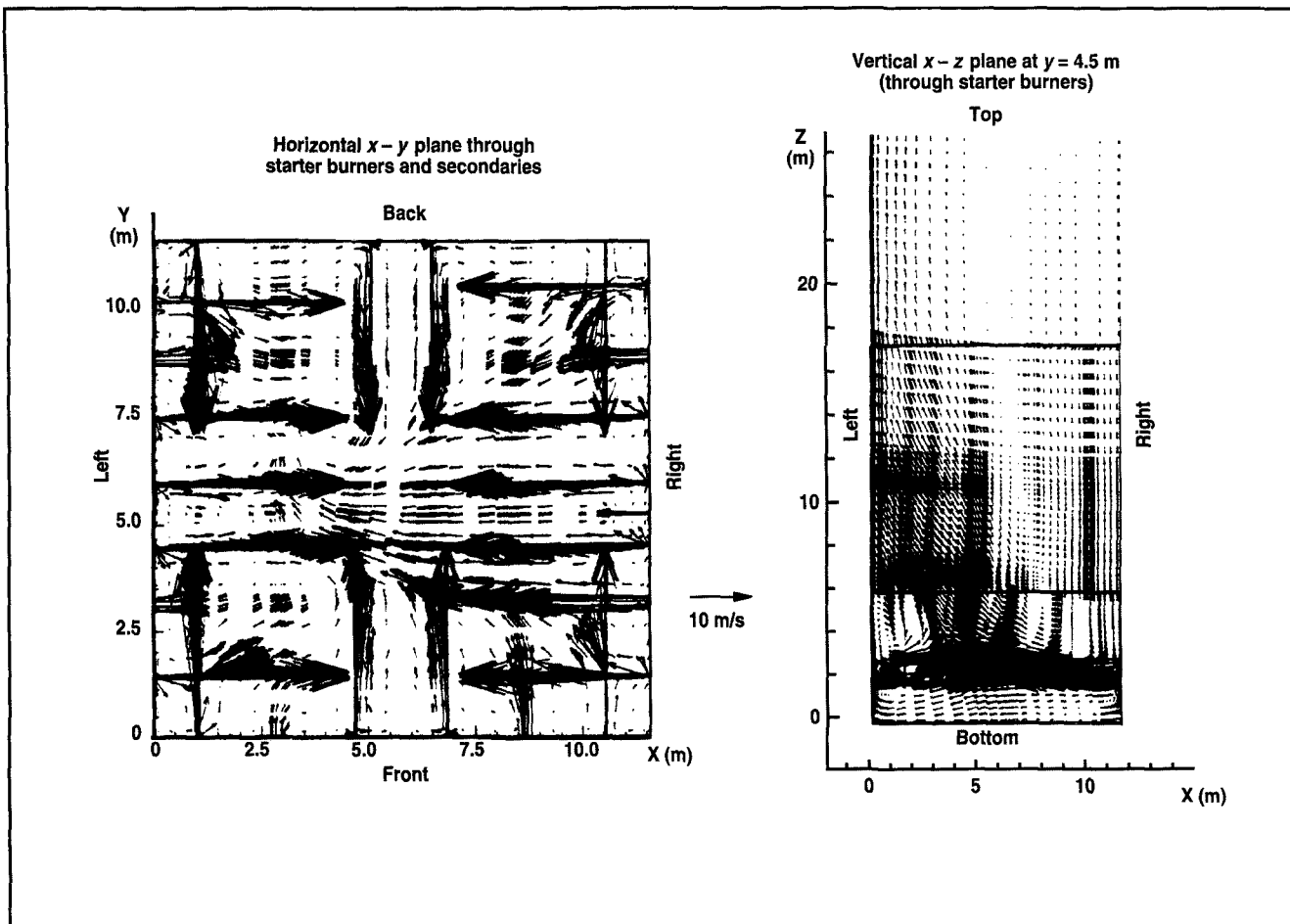
wall are modeled as slots. However, each secondary and starter-burner port is modeled individually. The computational domain extends up to the top of the boiler and includes the bull nose. The superheaters and boiler tube banks are not modeled.

Figure 3 shows some horizontal slices of the three-dimensional grid in the lower regions. The grid at each level is different and addresses the air-port configuration and the requirements for resolution of local flow.

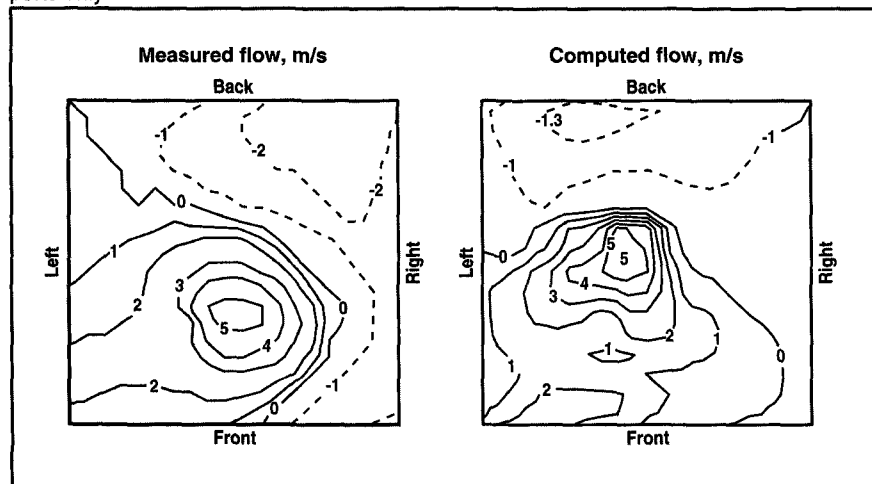
The boundary conditions on the computational domain are specified as follows:

- Inflow: This condition is applied to all the air-port openings. Values of all the variables except pressure are specified. The tur-

6. Flow field in a horizontal plane (secondary level) and in a vertical plane (through starter burners at front of boiler). Case 1: inflow through primary and secondary ports.



7. Measured and computed air flow at the liquor-gun level. Case 2: inflow through secondary ports only.



bulent kinetic energy,  $\kappa$ , is estimated from the inflow velocities, and the turbulence energy dissipation rate,  $\epsilon$ , is estimated from  $\kappa$  and a characteristic length scale.

- **Outflow:** This condition is imposed at the exit of the computational domain, located close to the boiler tube bank. A zero streamwise gradient condition is imposed for all variables.
- **Wall conditions:** This condition is applied at all the walls of the domain. Zero values for the normal and tangential velocities are used, in conjunction with conventional wall function treatment (10) for the turbulence quantities and the wall stresses.

## Results and discussion

### A first step

The cold-flow velocity measurements and the mathematical modeling reported in this paper represent only

## Simulated Boiler Flow

the first step in a series of tests to validate the mathematical model. Additional cold-flow trials with radical variations in combustion-air distribution are planned. Measurements under operating conditions will also be attempted as part of the validation procedure. We hope that the combined series of validation tests will increase industry confidence by demonstrating that the mathematical model predicts qualitatively realistic flow patterns and that it shows the same trends observed in full-scale physical measurements.

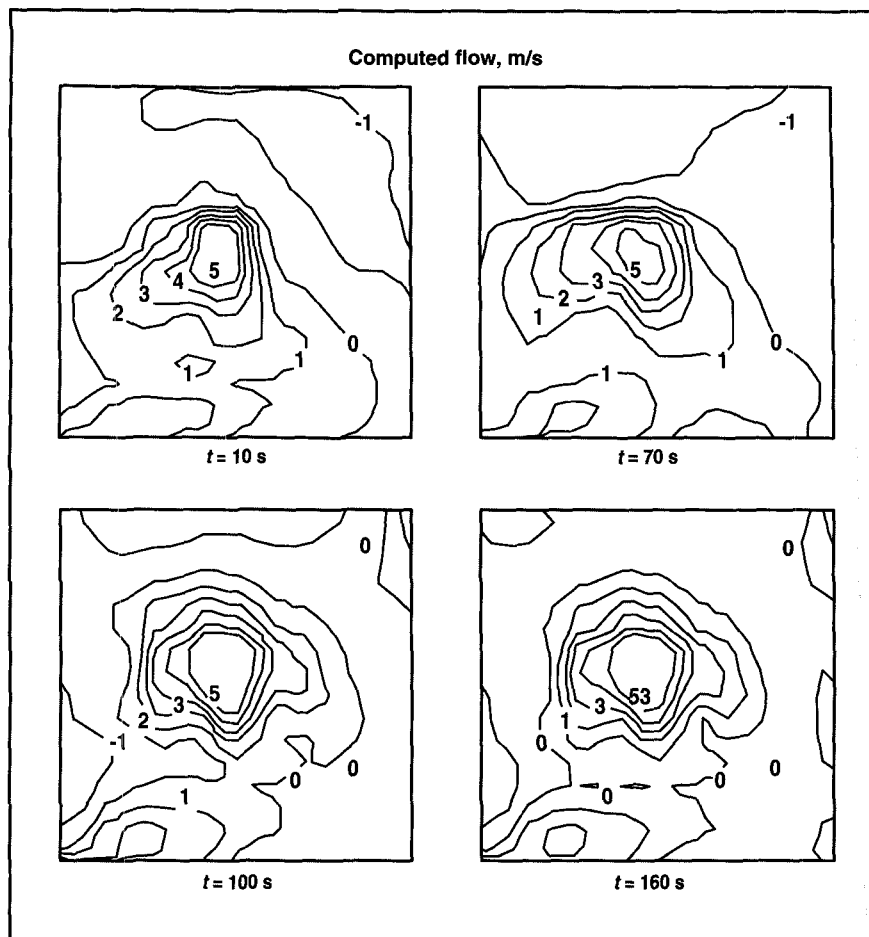
### Sources of error

The physical measurements and the mathematical modeling were carried out for a relatively large B&W recovery boiler (rated capacity 4 million lb of black liquor solids/day; approximately 12 m in cross section). The measurement grid shown in Fig. 2 was quite sparse, with large regions where it was not possible to obtain data. Velocities in these regions were estimated by interpolating between measured data points.

In addition, it was not possible to directly measure the velocity at most of the air ports. Instead, velocity was calculated from windbox pressures, which were obtained by inserting a probe into small access openings in each windbox. These pressure readings varied considerably, since they depend on the orientation of the flow to the pressure-measurement probe.

The velocity distributions across the port could not be obtained for most air ports. The hot-wire probes were accurate to 5% if the flow was reasonably aligned with the probe under the controlled steady conditions of a wind tunnel. However, the flow in the boiler had an unsteady component, which caused the ribbons to flutter and made it difficult to determine the direction of flow in some regions. Blackwell and co-workers (13) have also reported similar high levels of unsteadiness in the flow in recovery boilers.

8. Computed air flow at the liquor-gun level at different times. Case 2: inflow through secondary ports only.



Because of these difficulties, a 30% difference was noted between the total flow calculated from the windbox pressures and the calculated total upward flow integrated at the liquor-gun level. Although the 30% tolerance seems large, it should not be too surprising, considering the sources of error and the previously described problems associated with calibration of the hot-wire anemometers. The 30% variation is also well within the variation in velocity measurements at a given location observed in this and earlier studies (13).

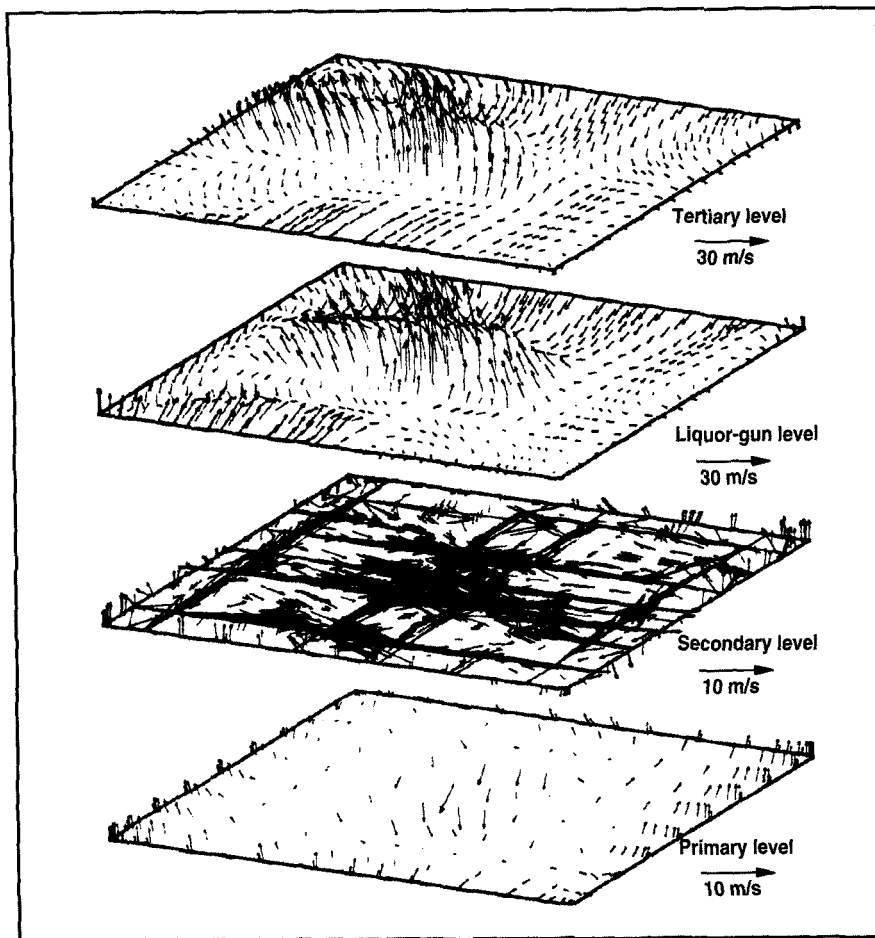
The degree of agreement between the measured and the predicted results thus should not be expected to be quantitative. At best, we should expect a qualitative agreement, as the computations are based on the velocity boundary conditions calcu-

lated from the windbox pressures. With this in mind, we present the measured and the computed results.

### Case 1: Air flow through primary and secondary ports

Figure 4 shows the measured and computed upward velocity distributions at the liquor-gun level for Case 1, with air flow through both the primary and the secondary ports. The measurements indicate a strong upflow core, mostly in the central region, but deflected somewhat toward the rear-left corner. A fairly strong downflow region exists along the right wall. The downflow is more pronounced in the front-right corner. Weak downward flow was also measured in the front-left corner. The downflow was weaker along the rear-right corner. Some fluctuation

9. Three-dimensional velocity distribution at selected horizontal planes. Case 2: inflow through secondary ports only.



of the flow direction was observed at the rear-left corner during the measurements, although the averaged flow measurements indicated a weak upflow in this region, as seen in Fig. 4.

The computed velocity distribution, shown on the right panel of Fig. 4, is very similar to the measured velocity distribution. The upward and downward flow and the zero-velocity regions are at similar locations. The computed results indicate a weak downflow in the rear left corner. However, the zero-velocity contour is also close to this location. In actual physical measurements, one would expect very low velocities (and even reversal in flow direction) in such a region. Indeed, measurements in this region revealed weak flow reversals.

Figure 5 shows computed three-dimensional velocity distributions at four horizontal planes. Air penetration at the primary-air-port level appears to be low. Flow at the secondary level interacts strongly, resulting in both downward and upward flow in the central region of the boiler. The downward flow component is less strong, as the flow through the primary pushes the secondary flow upward. This flow then combines with the upward flow in the central region to form the upward-rising core. This upward-rising core is deflected to the left by the strong flow from the starter-burner air port on the right wall, close to the front of the boiler.

The detailed flow pattern at the level of the secondary air ports is shown in the left panel of Fig. 6. The

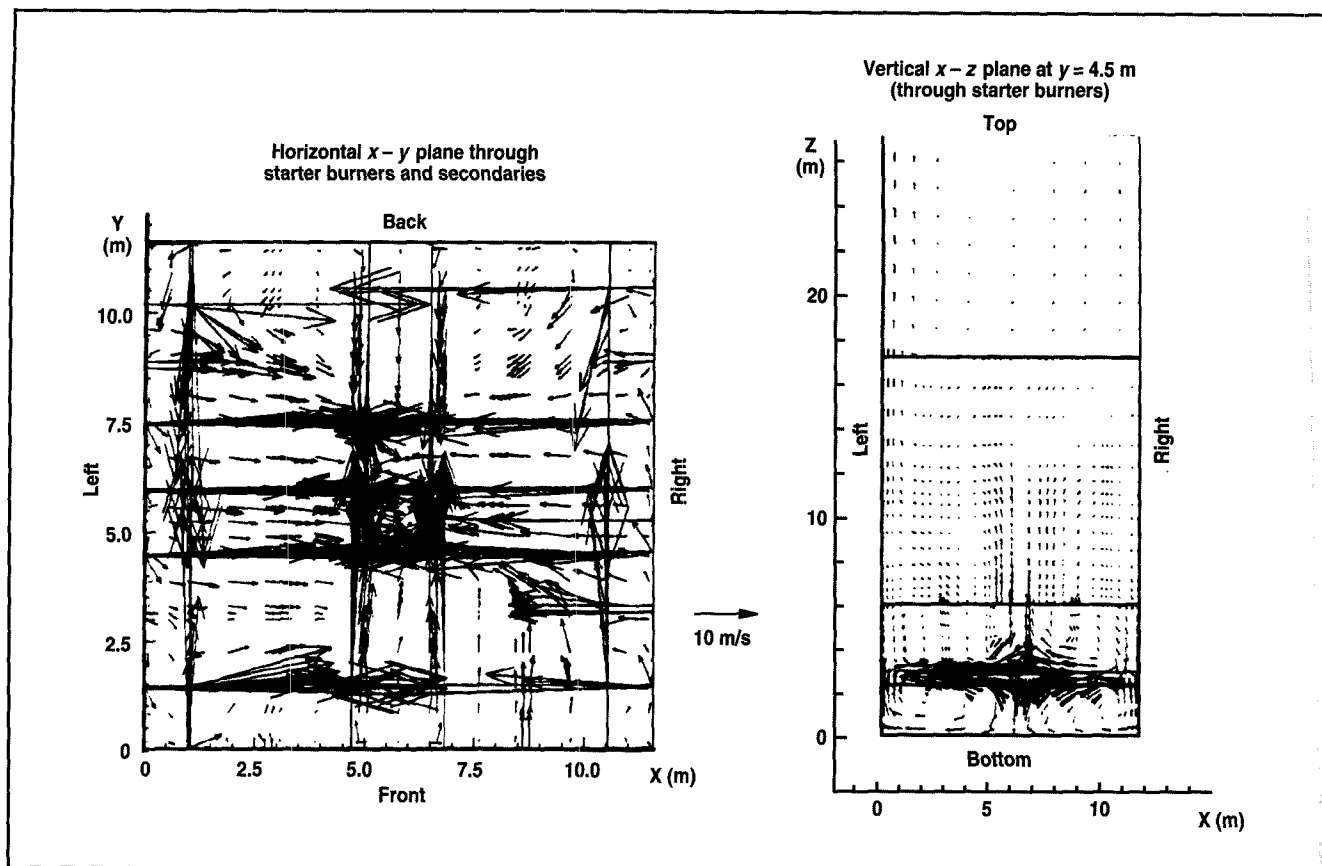
flow pattern in a vertical plane through the starter burners close to the front of the boiler is shown in the right panel of Fig. 6. Note the strong interaction of the air entering from the secondary and starter-burner ports in the left panel. The flow from the starter-burner ports, especially from the front right port, is very strong and penetrates deeply into the domain. This flow then turns upward and forms part of the upward-flowing core. The displacement of the upward flow toward the left wall is because of the deep penetration of the flow from the starter-burner port at the front-left region. Very high velocity gradients exist at the secondary level, with very low velocities in some regions between the jets and very high velocities in others.

#### Case 2: Air flow through secondary ports only

Steady-state computations did not yield a converged solution for Case 2, with air flow through the secondary ports only. A transient algorithm yielded a converged solution that varied with time. Computations were carried out to simulate a real-time period of 170 s, and a converged solution was obtained for each time step. However, no periodicity in the solution was observed during this computation. Considerable unsteadiness was observed in the cold-flow measurements for this case. Apparently, the primary air flow stabilized the flow field considerably. UBC has previously reported on the unsteady interaction of opposed-jet configurations (14, 15).

Figure 7 shows the measured results for one time step for Case 2 and the corresponding converged, computed results at the level of the liquor guns. Since the flow measurements were carried out over a period of several hours, the measured velocity distribution represents a form of an average of the time-dependent flow field. The average vertical velocity is approximately zero in a fairly large region close to the

10. Flow field in a horizontal plane (secondary level) and in a vertical plane (through starter burners at front of boiler). Case 2: inflow through secondary ports only.



rear-left wall. In fact, physical measurements showed considerable flow reversals in this region. The upflow is mainly in the center and in the front-left corner. There is downflow along the right and rear walls.

The computed velocity field for one time step shows many similarities between the measured and the computed fields. The computed field also indicates upflow to be along the front-left corner and downflow along the right and rear walls. The strongest downflow is along the rear wall for the time step shown in Fig. 7. This strong downflow region moves along the rear and side walls as the time is advanced in the transient computations.

Figure 8 shows the computed vertical velocity field at the liquor-gun level at different times. The transient nature of the flow field is apparent. The change of direction of

the flow along the left wall can also be seen as the time advances.

Figure 9 shows the three-dimensional velocity distribution at one time step for Case 2. There is considerable downflow under the secondary jets for this case, as the flow from the primary ports is not present to help deflect the secondary flow upward. The downflow from the secondary ports deflects upward from the floor of the boiler and interacts with the air from the secondary ports. The more central upward-flowing core can be seen for this case.

The detailed flow pattern at the secondary level at the same time step as in Fig. 9 is shown in the left panel of Fig. 10. The right panel shows the flow pattern in a vertical plane through the starter burners close to the front of the boiler. Note the strong interaction of the air entering from the secondary and starter-

burner ports. Since the flow from the primary ports is not present in this case, the flow from the secondaries is deflected downward more strongly. The upward-flowing core is more centrally located with respect to the two side walls than was in Case 1, shown in Fig. 6.

## Conclusions

Cold air flow in a recovery boiler was simulated using code developed specifically for boiler applications. Results were compared with preliminary experimental data from actual full-scale measurements.

The flow measurements and computations were carried out for two cases:

- Case 1—flow through the primary and secondary ports
- Case 2—flow through the secondary ports only.

Flow for Case 1 yielded a stable velocity field, as observed in both the measurements and in the simulations. Flow for Case 2 was unstable and required unsteady computational techniques. An upflow core was observed in both cases, and there was qualitative agreement between the computed and the measured results. Upflow and downflow regions were correctly predicted. The contours for zero upward velocity have a similar shape across most of the cross section, both for the physical measurements and the mathematical model. The peak values of the upflow and downflow velocities were similar for both the measurements and the model.

We encountered several equipment problems during the cold-flow measurement work and are taking steps to improve the quality of future work. The hot-wire anemometers were returned to the factory for recalibration, and work is planned to check the calibration and investigate the directional dependency of the probes. We also are planning to automate the flow-direction measurements using specially designed flow-direction indicators that will be mounted near the hot-wire probes. A computer interface will collect the directional information on-line. Problems with the construction of the extension tubes used with the probes have also been addressed. The new extension tubes will be constructed from lightweight aluminum tubing. This should reduce problems with end droop and make the probes easier to use. 

## Literature cited

1. Salcudean, M., Nowak, P., and Abdullah, Z., *TAPPI 1992 International Chemical Recovery Conference Proceedings*, TAPPI PRESS, Atlanta, p. 197.
2. Grace, T. M., personal communication.
3. Grace, T. M., Walsh, A., Jones, A., et al., *TAPPI 1989 International Chemical Recovery Conference Proceedings*, TAPPI PRESS, Atlanta, p. 1.
4. Karvinen, R., Hyoty, P., and Siiskonen, P., *Tappi J.* 74(12): 171(1991).
5. Frederick, W. J., Noopila, T., and Hupa, M., *Tappi J.* 74(12): 163(1991).
6. Chapman, P. J. and Jones, A. K., *TAPPI 1990 Engineering Conference Proceedings*, TAPPI PRESS, Atlanta, Vol. 1, p. 193.
7. Fiveland, W., *Proceedings of Timberline Colloquium on Recovery Boiler Modeling*, jointly sponsored by B&W and Weyerhaeuser, Mount Hood, OR, April 10-13, 1991.
8. Salcudean, M., Gartshore, I., Abdullah, Z., et al., "Mathematical modelling of black liquor recovery boilers," Report to US DOE, Department of Mechanical Engineering, UBC, September 12, 1991.
9. Reynolds, A. J., *Turbulent Flows in Engineering*, John Wiley & Sons, London, 1974.
10. Launder, B. E. and Spalding, D. B., *Comp. Meth. Appl. Mech. Eng.* 3: 269(1974).
11. Patankar, S. V., *Numerical Heat Transfer and Fluid Flow*, Hemisphere, New York, 1980.
12. Anderson, D. A., Tannehill, J. C., and Pletcher, R. H., *Computational Fluid Mechanics and Heat Transfer*, McGraw-Hill, New York, 1984.
13. Blackwell, B., Oliver, R., and Briscoe, B., *Pulp Paper Can.* 92(5):T121(1991).
14. Quick, J. W., Gartshore, I. S., and Salcudean, M., *TAPPI 1992 Engineering Conference Proceedings*, TAPPI PRESS, Atlanta, p. 123.
15. Abdullah, Z., Nowak, P., Salcudean, M., et al., *TAPPI 1992 Engineering Conference Proceedings*, TAPPI PRESS, Atlanta, p. 103.

The authors acknowledge the financial contribution of NSERC, B.C. Science Council, and U.S. DOE for this work. The contributions of Steve Ketler (UBC), Peter Gorog (Weyerhaeuser Tacoma), Brian Blackwell (Sandwell), and the staff at the Weyerhaeuser mill in Kamloops in obtaining the full-scale measurements are also gratefully acknowledged.

Received for review July 6, 1993.

Accepted Sept. 30, 1993.

Presented at the TAPPI 1993 Engineering Conference.