A CFD STUDY OF AIRFLOW IN A MIXING BOX

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BACKGROUND

One of the most important components of a heating, ventilating and air-conditioning (HVAC) system is an air-handling unit (AHU). This component is the interface between the primary plant and the secondary system. It contains the mixing box, the filters, heating and cooling coils and the fan. Because of its pivotal role as the merging point of two air streams and its widespread use, and because it is complicated enough to afford a challenging application of the techniques to be investigated here, the mixing box was selected as the first component to be investigated.

Mixing effectiveness is a prime concern, since mixing is the main purpose of the component. Poor mixing can negatively affect the relative distribution of ventilation air (EPA, 1991). Another important issue in mixing boxes is the extent of mixing and the distribution of temperatures across the discharge plane of the box. If one of the streams is below freezing temperature and mixing is incomplete, cold air passing across a portion of a water coil may cause a localized freezing condition (Delaney, 1984). Knowledge of this temperature distribution would be helpful to installers of freezestat control sensors. This information would also be helpful in knowing where average temperatures occur when selecting or locating mixed air temperature sensors. Finally, it would also be helpful in designing new versions of mixing boxes to improve mixing.

Robinson (1988, 1999, 2000) has recently published the results of field tests on several mixing boxes. He used earlier work done in the 1960's at the National Bureau of Standards as a source for three different methods for rating mixing effectiveness. These are Range Effectiveness:

$$E_{range} = 1 - \frac{RangeDS}{RangeUS}$$

Where RangeDS = t_{max}-t_{min} downstream and RangeUS = t_{max}-t_{min} upstream.

The second measure was Statistical Effectiveness:

$$E_{range} = 1 - \frac{SDds}{SDus}$$

where SDds = standard deviation downstream and Sdus = standard deviation upstream. The third measure, and the one used in his papers, was Modified Range Effectiveness :

$$E_{RdT} = 1 - \frac{t_{\max} - t_{\min}}{abs\left[t_{ra} - t_{oa}\right]}$$

Where E_{Rdt} = Modified range mixing effectiveness, t_{max} and t_{min} = maximum and minimum temperatures in the discharge air stream, t_{ra} and t_{oa} = temperatures of the return and outside air streams.

Robinson (1998) performed a field study of the mixing box in an air-handling unit installed in a Colorado school. The mixing box had a typical configuration with parallel blade dampers arranged to divert the inlet air streams toward each other. The inlets were located with outside air on the top and return air on the back. He measured the distribution of temperatures across the outlet plane of the mixing box both with filters and without. The maximum effectiveness he measured was 0.65 and the minimum was 0.12. The worst mixing occurred when the outdoor damper was open 15 degrees and the return damper open 75 degrees. At 45 degrees open on both sets of dampers, the mixing effectiveness was 0.22. The distribution of temperatures across the outlet plane is shown below in Fig. 1.

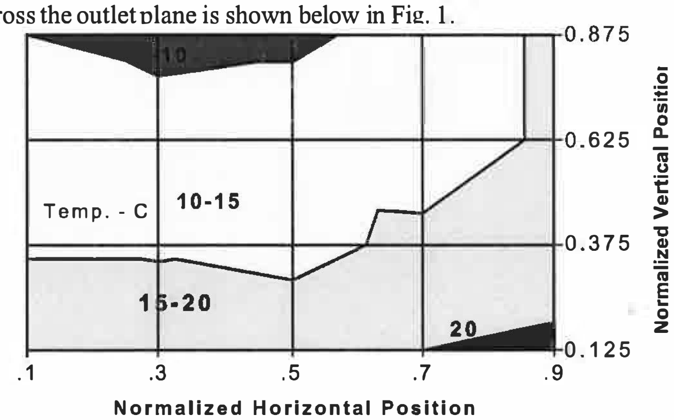

Figure 1 Temperature distribution at outlet plane of mixing box. Outside air enters from top, return air from back. Adapted from Robinson, 1998

The outside air duct approached the mixing box horizontally from the right at a 90 degree angle to the flow through the air-handling unit, then turned down to the inlet. This condition created a strong velocity component that created an asymmetric distribution horizontally across the mixing box as can be seen in the low temperatures in the upper left corner and the high temperatures at the lower right corner. In the test depicted in Fig, 1 the outside temperature was 8.4 C(47.1 F) and the return 23.3 C(73.9 F). Robinson's papers substantiate the suspicion that mixing is incomplete in typical air-handling unit mixing boxes.

Modeling a mixing box requires that the rate of airflow into both inlets be known at several damper positions. The inlet airflow rates will vary with the total airflow through the system, it is speculated, because of the differential between pressure drop in the return duct and that in most outside air ducts. The usual configuration is a short outside air duct and a long return air duct in which the pressure drop will vary in proportion to the square of the flow rate. Thus a single setting of damper positions will produce neither a constant actual nor a constant relative flow rate into the two inlets of the mixing box if the total airflow rate is changed. The importance of a known total airflow rate is even greater if a return fan is involved because of the coupling of two fans in series. Because of this one-way dependence, this investigation will include energy as well as momentum and mass transport.

Johnson Controls, in their Damper Manual (Johnson Controls, 1966), present the characteristics of parallel blade mixing dampers as shown in Figure 2. The installed characteristic is greatly influenced by the authority of the damper, which in the case of typical air handling unit mixing boxes, is usually quite low because the dampers are the size of the duct connections. Adding the individual damper flows can approximate total flow through a pair of mixing dampers. Figure 3 shows the result when the two dampers are of equal authority with pressure drops of 10% of the system total.

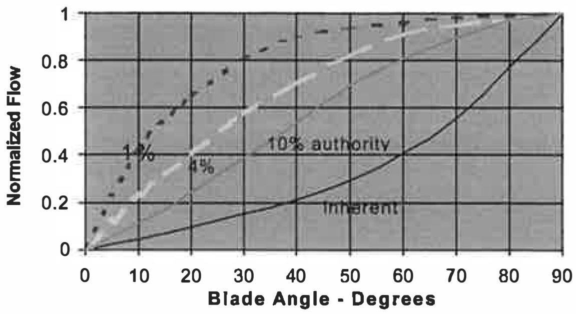

Figure 2 Normalized airflow characteristic of parallel blade dampers.

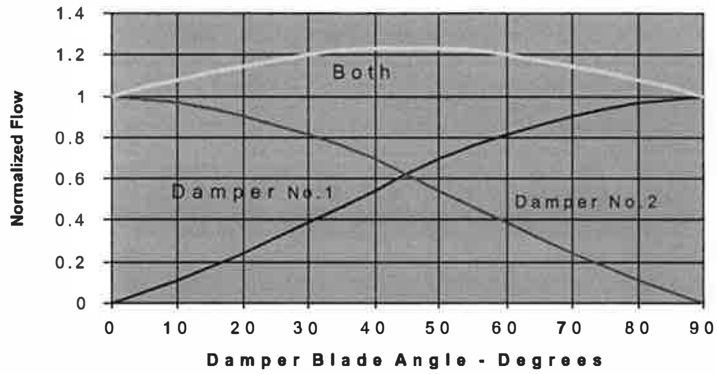

Figure 3 Normalized airflow when open damper pressure drop equals 10% of system pressure drop. Two dampers.

Alley (1988) investigated the effect of varying damper positions on pressure drop by installing two sets of dampers in an air duct. Each set of dampers occupied half of the duct area that was divided horizontally. The damper controls were arranged so one set closed as the other opened. Only total pressure drop across the two sets of dampers and, apparently, only total flow was measured. He found that the pressure drop across the dampers varied significantly with position and, for parallel blade dampers, was lowest at the 45^0 open angle.

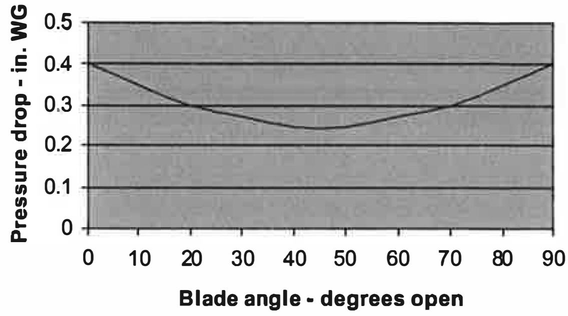

Figure 4 Parallel blade damper pressure characteristics from Alley, 1988

Airflow varies inversely with pressure squared and consequently airflow would increase by 25-30% when the dampers were opened to 45^0 from the initial condition of fully closed to one stream and fully open to the other. This relationship is shown in Figure 5. This curve correlates reasonably well with the Johnson Controls curve for damper pressure drop equal to 10% of system pressure drop, making it reasonable to assume Alley's dampers had this performance.

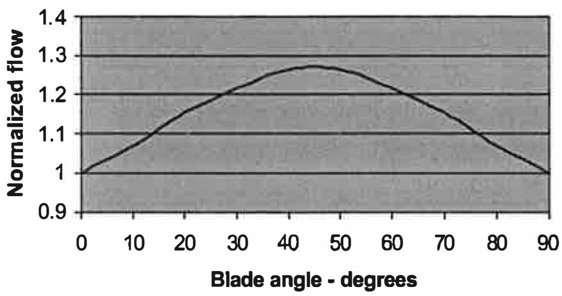

Figure 5 Normalized flow characteristic of a pair of parallel blade dampers

EXPERIMENTAL PROCEDURE

The mathematical modeling of fluid flows is emerging as a valid technique for studying air motion in HVAC systems (Baker, et al, 1994). This technique is computational fluid dynamics (CFD) and is the method adopted for this investigation. It requires the solution of the non-linear partial differential conservation equations for mass, momentum and energy throughout the computation domain. In this study a commercial finite volume code, CFX4 (CFX International) was used. It includes pre- and post-processors and a solver.

The AHU selected for modeling is that from the experimental units at the Energy Resource Station of the Iowa Energy Center, located in Ankeny, Iowa. This unit was selected because it was used in ASHRAE RP-1020 and because a large amount of data is available about the system's performance. It is a standard modular commercial variable air volume unit with a coil area of 0.56 square meters (6sf) and a nominal design airflow capacity of 1416 l/s (3000 cfm). The mixing box has its return air connection on the top and its outside air connection on the back. Downstream from the mixing box are filters, a heating coil, cooling coil and the fan. The duct connections are approximately 0.36 m by 1.17 m(14.2 in by 46.1 in). Neither duct approach is straight, but vaned elbows are located a short distance upstream.

The model was constructed using the preprocessor with inlet ducts extended three hydraulic diameters upstream so as to achieve fully developed flow at the mixing box. Uniform velocity profiles were imposed at the duct entrance. Similarly, to force confined flow at the mixing box outlet, a duct was extended three hydraulic diameters downstream. The mixing box dampers were placed at the end of the inlet ducts approximately 2cm upstream from the mixing box walls. The mixing box has two dampers, acting in parallel, and inclined toward each other, extending parallel to the long dimension of each duct connection. The damper profile modeled is a thin flat plate. The actual damper profile is a streamlined section and this profile will be used in future studies.

Test conditions have the dampers in both ducts open at 45 degrees with 50% of the flow from each duct. This gives an average velocity (modeled as a uniform velocity profile) of 3m/s (590 fpm) at the duct entrance. The Reynolds Number is in the range of 170,000, so the flow is fully turbulent. The temperature of the "outside" air stream is 273K (32 F) and the "return" air stream is 294K (69.8F). The ducts and mixing box have negligible heat transfer compared to the other heat transfers in the system, so they are modeled as adiabatic. CFD modeling parameters are steady incompressible turbulent flow at standard density and viscosity. The turbulence model is the standard k-epsilon model. Boundary conditions at the walls are law-of-the-wall for k and epsilon.

RESULTS

A vector plot along the Z plane is shown in Figure 6. The flow in the inlet ducts is fully developed just upstream of the dampers. Most of the flow goes between the two damper blades and eddies form downstream of the blades and at the corners of the mixing box where separation occurs. The two streams come together in the center of the mixing box but mixing proceeds slowly as the flow continues downstream. A large eddy forms at the top of the mixing box while the main flow is in the lower half of the box area.

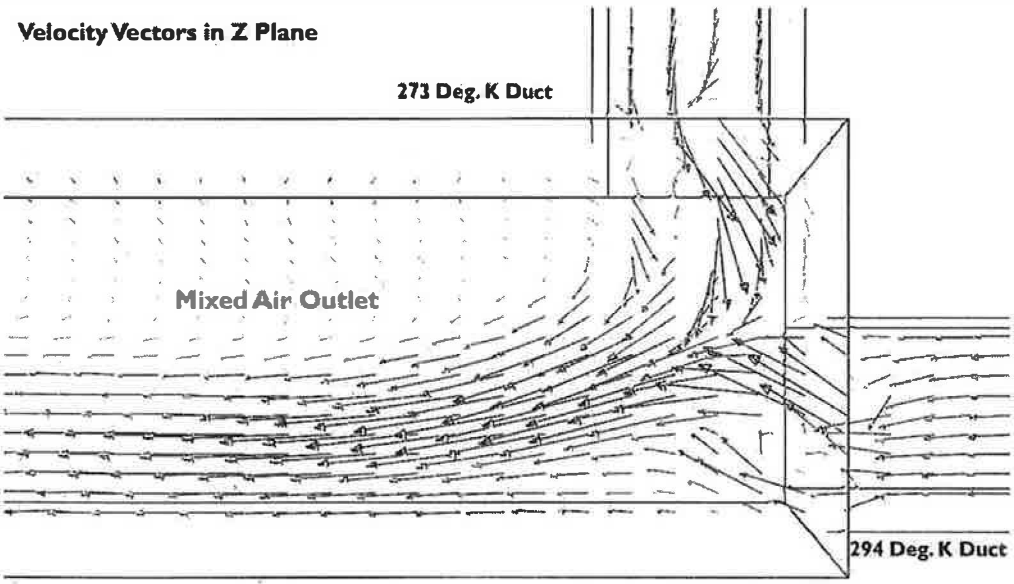

Figure 6. Velocity vectors on mixing box centerline in Y plane.

Figure 7 is a velocity vector plot along the X plane just inside the mixing box. Because the inlets are offset from the center in this particular box design, there is a large area of recirculation in the corner of the box.

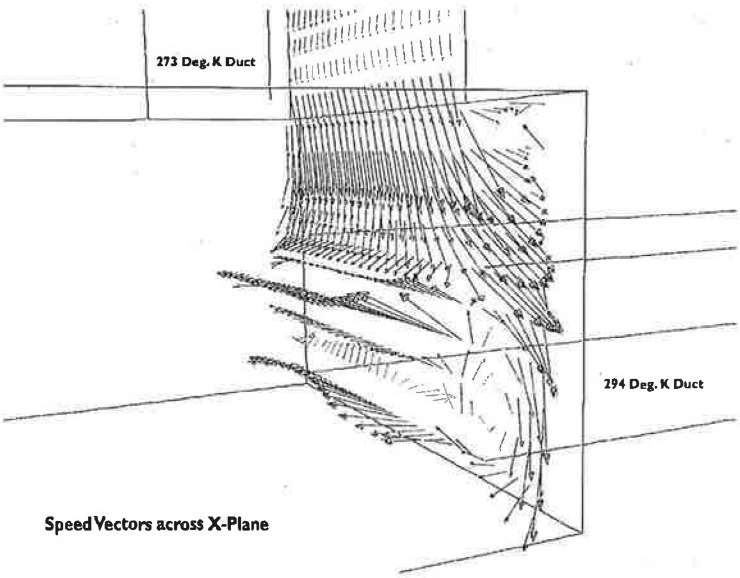

Figure 7. Velocity vectors in X plane just inside mixing box.

Temperature distribution in the Y plane is plotted in Figure 8. The warmer air from the right inlet duct clearly is forced to the bottom of the mixing box and its core maintains its temperature to the outlet of the mixing box. The cooler air from the top inlet is recirculated strongly in the eddy at the top of the box. The mixing layer between the incoming streams is very thin until the downstream end of the eddy is reached. If the right inlet duct were carrying air at below-freezing temperatures, a water coil immediately downstream of the mixing box would clearly be exposed to potential freezing hazard where the cold jet would persist unmixed.

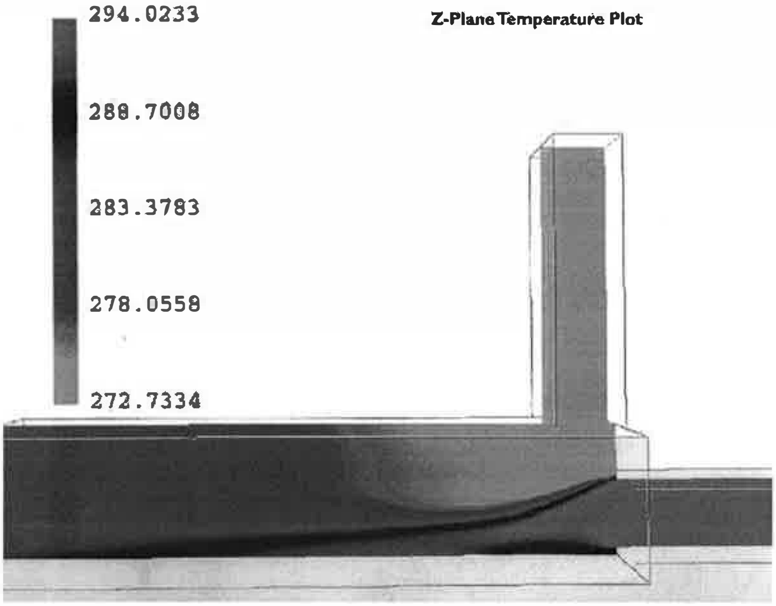

Figure 8. Temperature plot on mixing box centerline on Y plane

Figure 9 is a temperature plot along the X plane at the mixing box outlet looking upstream. This plot corresponds roughly to Robinson's measured distribution shown in Figure 1. The unmixed air in the eddy and that in the jet flowing along the lower part of the mixing box are visible. Again, if below-freezing air was in either inlet duct, a water coil would be vulnerable. Similarly, if the horizontal duct carried outside air, and further mixing did not take place at downstream filters, coils and fan, duct branches along the top of the supply duct possibly would not receive adequate fractions of ventilation air. Modified range mixing effectiveness in this instance is about 30% at the mixing box outlet and about 75% three meters (10 ft.) downstream.

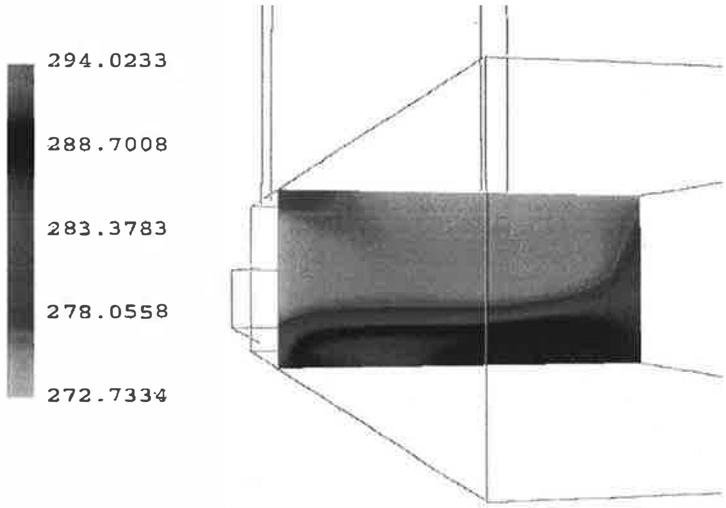

Figure 9. Temperature plot in X plane at mixing box outlet, looking upstream

CONCLUSIONS

This CFD study reveals details of flow in the mixing box that agree with expectations and with Robinson's field data. Mixing performance is poor and coil freezeup potential is clear. Ventilation air distribution is suspect. Mixed air temperature sensors and single-point freezestats located at most points in the mixing box outlet plane would not sense average or typical temperatures. The best location, based on this single condition, would be about 30% of the distance up from the bottom of the box in the mixing layer.

FUTURE STUDIES

This study is the first step in a planned series that will investigate air distribution at several combinations of damper positions and air temperatures. A study of pressure variations and airflow rates at various damper angles is also planned. Field tests for benchmarking would be desirable but are not likely without additional external support.

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