COMPUTERIZED BALANCING: GARBAGE IN, GARBAGE OUT

By ***LES MILEWSKI,***
Director of Engineering,
Metal Industries, Inc.,
Clearwater, Fla.

There's no doubt that new HVAC systems will be entirely DDC controlled in the near future (one to three years). This year at my company, for example, pneumatic controls, while still the biggest seller, dropped below 50 percent of all controls for the first time in history. We expect that the final result will even show DDC controls to top pneumatics in 1993 as the best seller, after all the dust has cleared. I also feel analog electronic controls will disappear even faster than pneumatics as DDC prices drop below analog levels on their way toward pneumatic prices. (This is for 100+ zone jobs; on small jobs, it's analog, then DDC, and then pneumatic in an increasing cost order.)

So what happens to the balancer when you have a job where all the adjustments—maximum and minimum flows, heating flows, day and night set points, heating steps, etc.—not only can be entirely set from a remote location (even another city) but often have no means to be adjusted at the zone without specialized gear? Very few basic changes occur. The biggest effect is that the balancer has to learn a new technology of controls and will have to invest in some specialized tools to work with the DDC controls. He is still the final word on how much air is coming out of each diffuser. Multiple diffusers on a single terminal are not even addressed by the DDC system, and their balance work is totally unaffected by the DDC invasion.

Most importantly, the controls people believe they know the cfms better, more quickly, and more accurately than the balancer. This is simply a false assumption. The DDC controls man is using the terminal manufacturer's data on terminal size and pickup signal versus cfm. With good transducers, he can read and resolve these readings to a fraction of a cfm if desired. Unfortunately, many of the velocity pressure transducers in use today come up a little short in the stability area, but they are gradually improving.

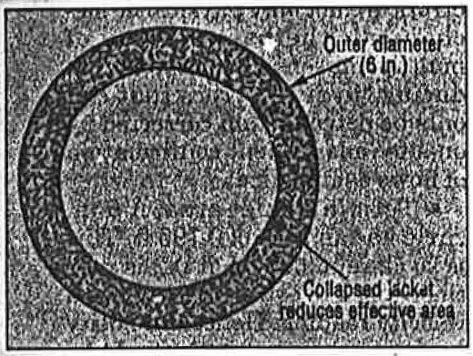
The weak link with all this is the terminal flow sensor's accuracy. The DDC controls may be able to read and resolve a reading like 1124.5 cfm, but the actual flow could easily be several hundred cfm higher or lower, depending on pickup accuracy, duct configuration, transducer accuracy, pickup tubing size and length, and actual effective area of the terminal inlet.

The bottom line is this: The only repeatable, calibrated, certifiable air flow information to the zone is that from the balancer's equipment. Even basic things like a forgotten diffuser connection or an open, unused outlet would only be caught by the balancer during his checkout. Getting back to the DDC measurement problem, the errors are not only physical or mechanical; the basic theory of multi-point sensing is flawed, producing "in situ" errors that can only be detected and corrected by on-site independent confirmation of flows as installed:

Flow and velocity formulas are combined and customized for each terminal in a DDC system

- Terminal inlet area. A 6 in. circle has an area of 0.196 ft. This is so close to 0.20 that 0.2 is routinely used to estimate cfm from a velocity reading. Hence, a velocity pressure signal of 0.25 would be read as 2000 fpm times 0.2 area for a 400 cfm flow. The fact is, however, that terminals are typically 1/8 in. or more undersized to accept slip on ductwork. The true area of a 5 7/8 in. diameter terminal is 0.188. This times the same 2000 fpm reading equals 376 cfm—already a 6 percent error.
- But it can get much worse. The controlling effective area for a flow pickup within a foot or less of the duct connection can be the flexible duct itself. Take a 1 or 2 ft piece of 6 in. flexible duct and squeeze it lengthwise like an accordion—similar to what would

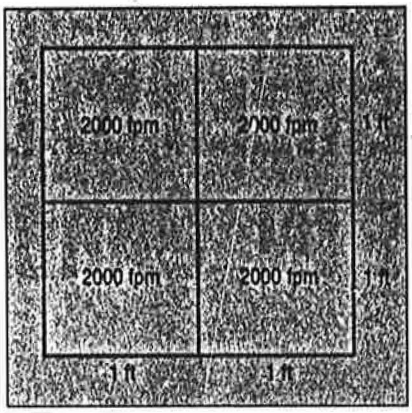
happen had you climbed a ladder with a 2 ft piece and found you only needed 14 or 15 in. Then, look at the inside (Fig. 1). This area reduction can be as much as 1/2 in. all around, producing a 5 in. diameter approach. This reduction concentrates the flow of air toward the center and away from the wall of the terminal inlet. The terminal pickup, even a multi-point averaging type, doesn't have any pickup points within 1/2 in. of the wall anyway, so it only reads the higher velocity, which indicates a higher-than-actual cfm. If we assume there is only a 1/4 in. perimeter effective reduction at the pickup (in addition to the 1/8 in. smaller diameter), we get an



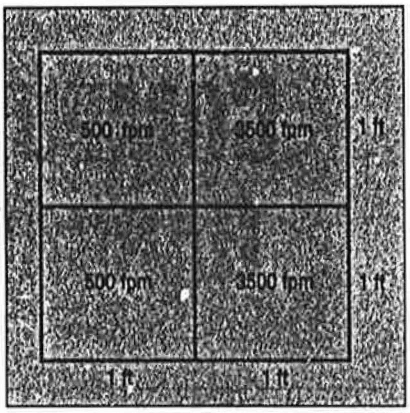
1 Effective area can be reduced as much as 1/2 in. all around, producing a 5 in. diameter approach.

effective area based on 5 3/8 in. or 0.158 ft. The same 0.25 in. pickup reading means only 315 cfm is flowing—not the computed 400 cfm.

- Pickup tube size and diameter. Many DDC transducers consume air. It is a small amount—typically 10 to 20 sccm, depending on the manufacturer and the signal strength. This is not a problem with factory-installed DDC systems, but some field installations have remote transducers for convenience. The air has to flow twice the distance from pickup to transducer (in the high side, down to the transducer, and back out the low side). If these tubes are too long or too small, the transducer sees a



2 Illustration of the example—2 by 2 ft square duct is divided into four imaginary 1 by 1 ft sections.



3 Illustration of upset flow due to an elbow or other disturbance upstream in the same sample duct.

smaller-than-actual signal and underestimates the air flow.

If the zone wanted 400 cfm, in the first case the DDC system would read 400 cfm but actually deliver 376 cfm. In the second case, again the actual is lower—possibly 315 cfm. The third case, however, sees a lower signal than actual, increases flow to the desired 0.25 signal, and delivers too much air.

The other effects of installation—elbows just in front of terminals and pickup designs—both result in flow reading errors that are rooted in the flawed concept of multi-point averaging sensing. To best explain this, let's use a simple 2 by 2 ft square duct divided into four imaginary 1 by 1 ft sections, as shown in Fig. 2. For simplicity of mental math, let's round off the standard formula:

$$P_v = \left(\frac{\text{velocity}}{4005}\right)^2$$

to

$$P_v = \left(\frac{\text{velocity}}{4000}\right)^2$$

Perfectionists who refuse to drop the 5 may assume slightly higher altitudes or warmer air to get to 4000. Now, let's start with the perfect cross-sectional flow situation of 2000 fpm in each 1 ft square. This times 4 ft equals 8000 cfm. A pitot tube sampling in the center of each square would read 0.25 in.

Now, let's upset the flow due to an elbow or other disturbance upstream, but for simple mental math again, let's assume the disturbance is even across each 1 ft section, producing the actual velocities in each section as shown in Fig. 3. Note that we still have 8000 cfm.

Now, let's look at the velocity pressure in each section:

$$P_v = \left(\frac{500}{4000}\right)^2 = 0.0156 \text{ in.}$$

and

$$P_v = \left(\frac{3500}{4000}\right)^2 = 0.766 \text{ in.}$$

The above readings would be obtained if we used a simple pitot tube and took separate readings in each section. Here the fallacy of the multi-point averaging flow pickup steps in. The signal one would get off a perfect four-point averaging pickup, with one point in each of the squares, is 0.0156 in. + 0.0156 in. + 0.7660 in. + 0.7660 in., or 1.5632 in. This divided by 4 equals 0.391 in.

Using this averaged signal to compute the velocity, we get:

$$0.391 \text{ in.} = \left(\frac{V}{4000}\right)^2$$

or

$$V = \sqrt{0.391} \times 4000$$

This "measured" multi-point average velocity of 2500 fpm × 4 ft, or 10,000 cfm, is indicated instead of the actual 8000 cfm flowing. So here's a 25 percent error in a "perfect" averaging pickup.

This example was a bit dramatic because of the big spread in velocities selected. Repeat the problem with 1500 and 2500 as differences, and you get 8250 cfm instead of the correct 8000 cfm. I'm not saying averaging pickups are no good; remember, a one-point pickup would have read 2000 cfm or 14,000 cfm, depending on where in the example it happened to be sitting. The point is that each of the above items contributes some error to the indicated cfm.

There is only one solution to the problem of obtaining accurate settings on air terminals, and it lies with the balancer. Fortunately, all of the above mechanical errors—squashed duct, elbows, even pickup tubes—along with the flawed averaging pickup become constant after installation. So the solution is simple: the balancer or DDC controls person drives the terminals to some temporary maximum. The balancer "hoods" the diffuser and establishes an accurate flow figure. The DDC man then compares that flow figure to his flow reading and establishes the correction factor needed, with each terminal having its own "in situ" factor.

Because the flow will still follow a square root formula (since compression is insignificant), a custom flow curve for each terminal can be produced. The original formulas:

$$\text{Flow (cfm)} = \text{velocity} \times \text{area}$$

and

$$\text{Velocity} = \sqrt{P_v} \times 4005$$

are combined and customized for each terminal by this approach. The 4005 is replaced by a custom constant and incorporates the area factor as well. The resulting formula is:

$$\text{Flow (cfm)} = T_c\sqrt{V_{sig}}$$

I prefer V_{sig} to P_v to avoid confusing the universal term for velocity pressure, P_v, with the signal generated by the manufacturer's pickup, which is not true P_v. Ω