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Designing Return Duct and Exhaust Systems

A case history illustrates a common problem in the design and construction of exhaust systems

How many times have you designed an exhaust system and had the air balancer come in and say, "I can't get the design flow. We need to speed up the fan." The fan is cranked up and the situation improves, but the air flow still doesn't quite meet design at every grille. You know that you overdesigned the fan by 20 percent, and the duct traverse at the fan shows that the fan is indeed doing exactly what it should be doing. But you add up the readings taken at the grilles and the total is still 10 percent less than design. You have this uneasy feeling that things are not adding up, but you are aware of the inherent inaccuracies in measuring air flow and the "questionable virtue" of air balancing technicians. You also know that duct resistance and stack effect take their toll somehow, so you shrug it off and move on to the next project.

What is going on here? Are all air balancers liars? Do air measuring instruments somehow have all their inherent error biased to the low side? Have the laws of physics been amended when they are applied to exhaust systems? Let's take a look at a case history and see if we can make some sense out of this.

The example involves the construction required to finish shelled-in space in a new building that was part of a major hospital complex. The area being finished was to house the diagnostic radiology function, so the ductwork systems were large and quite complex. The exhaust system served two floors and was not symmetrical. The exhaust duct was wide and shallow and hung against the ceilings to allow the supply duct and other utilities to be installed below it in the corridor ceiling space.

The system design called for a 12,000 cfm exhaust fan operating at 2 in. WG static pressure, and the sum of the air flows required at the grilles was 8310 cfm. At the completion of balancing, the fan was exhausting 13,066 cfm, and the sum of the air flows at the grilles was 7931 cfm—379 short of design. That doesn't seem too bad if you only compare the measured air flow at the grilles to the design. However, if you look at what is happening at the fan, the situation is a lot less tidy. The fan is operating at 5135 cfm above the air flow measured at the diffusers. This has some significant financial implications in terms of fan drive energy and air infiltration. For example, this 5135 difference must be made up by infiltration. This air must be heated in the winter and cooled in the summer, the estimated annual cost of which is $8200. Also, the fan is operating well above the design cfm. The annual cost of added fan energy is $26,000.

To gain a better understanding of what was happening, the air balancing technician was directed to traverse the main branches of the exhaust duct on one of the floors. As shown in Fig. 1, there are three locations at which the duct was traversed. At each location, the air flow of the traverse was compared to the sum of the measurements at the upstream grilles, and the difference of the two measurements was calculated (Table 1).

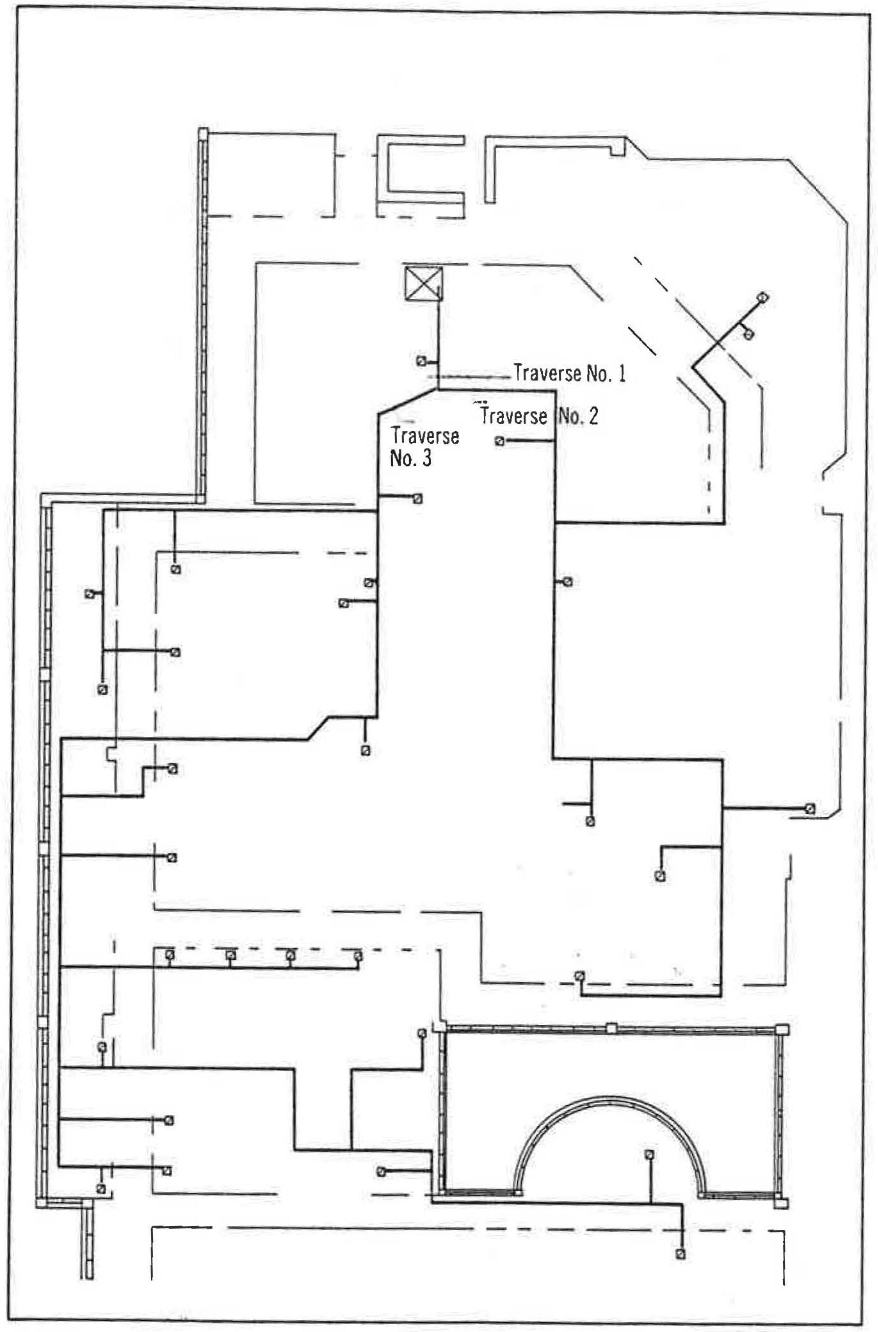
For the purposes of this discussion, I would like to ignore the inaccuracies due to measurement and look at what is happening in the system based on the measurements taken. The following observations can be drawn from the information in Table 1:

- With the exception of one branch, the calculated leakage rates are very high.
- The branch with the low leakage rate is almost certainly operating at a lower static pressure differential to atmosphere than the other branches because it is a shorter, less complex branch carrying a relatively small portion of the system cfm. It is reasonable to assume that the leakage in the system is proportional to the static pressure differential.
- The leakage is distributed throughout the duct system and is somewhat worse in the larger branches of asymmetric systems.

The situation described here is not an isolated incident. We recently engaged an air balancer to take similar measurements of two exhaust systems and one return air system. Each of the systems was approximately 25,000 cfm. The findings were very similar to those described for the radiology system in the example. The total system leakage rates ranged from 25.6 to 54.7 percent.

Let's return for a moment to the assumption that the measurements were accurate. It is reasonable to expect measurements within an accuracy of ±10 percent. Assuming the worst case, this will still give a total system leakage of 29 percent and branch leakage rates from 0 to 24 percent. There is no doubt that the problem is significant.

This is not being presented as a definitive statement of duct leakage because it is based on field measurements and a rather cursory analysis. Still, the observations and measurements are very consistent, and I believe that it indicates that the HVAC industry must undertake a major reevaluation of the design and construction of exhaust and return duct systems. Ω



1 Duct system layout.

Table 1—Air flow measurements taken at various locations.

Location	cfm (A)	cfm at upstream exhaust grilles (B)	cfm leakage (A - B)	Percent leakage
Fan	13,066	7931	5135	39.00
Traverse No. 1	4,190	3066 (red, blue + duct)	1124	26.80
Traverse No. 2	742	648 (blue duct)	94	6.48
Traverse No. 3	3,448	2275 (red duct)	1173	34.00

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