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Pills for Sick Duct Systems

Duct systems often fail when designers overlook the fundamentals

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Melvin Ramsey wrote a series of articles in the 1950s and 1960s which were published in "Heating and Ventilating" and "Air Conditioning, Heating and Ventilating" magazines, copyrighted by the Industrial Press. Subsequently all these articles were consolidated and published in a book "Tested Solutions to : Design Problems in Air Conditioning and Refrigeration". The Industrial Press has since closed down.

The Simple Duct System in **Figure 1** has one outlet near the fan, and an identical one about 50 ft. from the fan. Since it is desired that they deliver the same cfm to the same size room, outlet velocities of each must be the same. How does the pressure drop between the fan and outlet A compare to that between the fan and outlet B?

Figure 1

Engineers, *who ought to know better*, have told me that the pressure loss between the fan and the outlet B is greater! Since these are parallel circuits between a common fan and rooms in which the pressures are equal, how can the pressure drop to A and B be different?

If ductwork is designed on the mistaken assumption that, with equal flows, the loss between the fan and B will be greater, when the system is placed in operation, air flow will so proportion itself that the pressure losses through the two circuits are equal. Then A will deliver more air than desired; B, less. To rectify the error, some restriction, such as a damper, will be put in the circuit to A to reduce flow.

Some time ago, I had the problem of correcting the distribution in a system which had been designed on the assumption that the pressure drop to some outlets near the fan would be less than that to the most remote outlet. To get the desired flow from each outlet, it was necessary to impose as much as 0.25 inch of water resistance between the main duct and an outlet near the fan.

Schemes that won't work

The main duct was in a furred ceiling, making any access to it impractical. Duct branches to the ducts were not over a foot long, as in **Figure 2**. The space was inadequate for a single damper because (1) the damper would have to be nearly closed to lose 0.25 inch and (2) the two streams of air from each side of the damper would have let through only a small part of the grille area instead of over the whole face of the grille, resulting in poor distribution of air to the room. **Figure 3** illustrates what will happen. Note the air intake in the center of the grille.

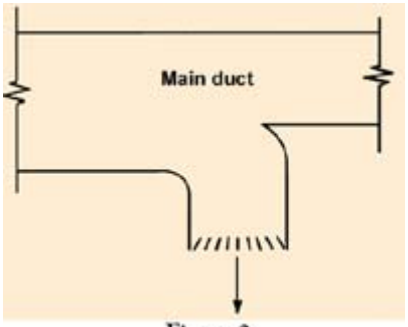

Figure 2

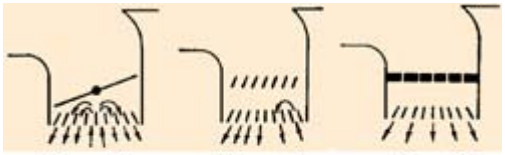

Figure 3 Figure 4 Figure 5

Poor air distribution results when single-blade (Figure 3) and multi-blade (Figure 4) dampers are used. Solution (Figure 5) is piece of Celotex with half-inch holes

A multi-blade damper, with all blades turning in the same direction, would have sent the air to one side of the grille and have given poor distribution, as shown in **Figure 4**. Note the air intake at one side. A multi-blade damper, with alternate blades turning in opposite directions, would have been much better but, unless a considerable number of blades were used, the space would still be too small for really even distribution over the face of the grille.

Perforated board is excellent damper

The problem was solved simply and effectively by inserting a piece of ½ -inch thick Celotex, into which a number of holes had been drilled, as shown in **Figure 5**. The air through each hole reaches a maximum velocity at a vena-contracta which has an area about 0.6 times the cross section of the hole. The pressure loss is somewhat less than the head corresponding to the velocity at the vena-contracta. The value of that head is given by

$V = 4005\sqrt{h}$, where V is the velocity in fpm and h is the head, in inches of water. (The constant 4005 applies for standard air, 0.075 pcf.)

To lose 0.25 inch, V must be about 2000 fpm. With 2000 fpm through the vena-contracta, the average velocity through the holes must be about 1200 fpm. Total area of the holes is the cfm for the grille divided by 1200 fpm. The perforated insulating board has the holes close enough to result in even distribution. It imposes the approximate desired resistance and results in extremely little noise.

Air goes the wrong way

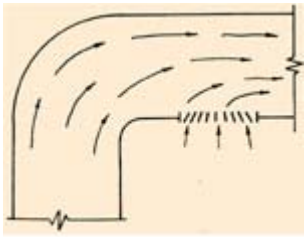

Figure 6 : Outlet too close to elbow

Another case in point involves the location of outlets too close to a change in direction, as shown in **Figure 6**. The designer intended that air leave the duct at the outlet shown. However, the air attains a higher than average duct velocity as it approaches the outside radius of the elbow. This can result in air entering part or all of the grille instead of leaving it. In a cotton weaving plant, a case like this was clearly indicated by the lint. The outlets delivering air were fairly free of lint. One, in a position similar to that in **Figure 6**, was well coated with lint over the whole outside face, showing that air was entering the duct through the grille.

Plenums often waste power

Plenums often cause needless pressure loss. **Figure 7** illustrates a case I recently encountered, with a plenum between a two-fan air conditioner and three ducts, each with a heating coil. (For convenience of discussion, assume the three branches are of equal size and air flow.)

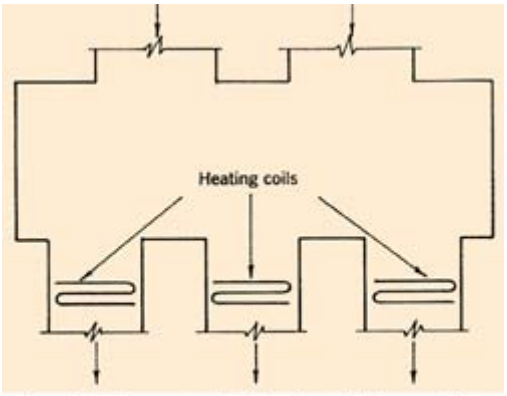

Figure 7 : Plenum connecting two-fan conditioner and three ducts, causes excessive pressure loss

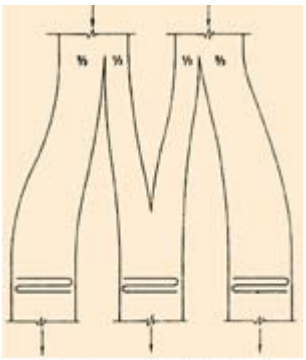

Figure 8 : This is the way the problem in Figure 7 should have been solved

Air velocity leaving the fans is about 1,700 fpm, which corresponds to 0.13 inches SP. The plenum effectively loses this velocity energy and, because of the great turbulence in a plenum arranged in this way, the actual loss will be even greater than the 0.13 inch of velocity pressure.

Transformation from fan outlet to the duct should be made with a gradual change in cross section. In this case, the change should have been a gradual increase in area from the fan outlets to the larger duct cross section, as shown in **Figure 8**.

The plenum looks simple and requires less work on the part of the designer, but it is probably not less costly because a plenum, as shown in **Figure 7**, requires heavier gage metal and some angles for bracing. Plenums are not too bad in residential systems using velocities of say 800 fpm (corresponding to 0.04 inch SP) because loss of such a velocity head may not be of much consequence. However, with some higher velocity industrial systems, losses caused by plenums have been costly power losses. The important thing is to realize that a plenum results in a loss of energy and should be used only if its simplicity compensates for the loss.

Scoops permit smaller ducts

The design of ducts can often result in lower cost as well as better air flow if full advantage is taken of the possibility of changing velocity pressure to static pressure and vice versa. For example, take the case of an air handling unit that delivers 26,000 cfm to a duct (**Figure 9**) running 200 ft in a cold storage room. The unit has two fans (each 13,000 cfm) and stands on the floor, discharging vertically. Fan outlet velocity is 1,800 fpm. Two ducts, one from each fan, rise with gradually decreasing cross section as they join into one duct which has reduced to 4 ft x 2 ft by the time it reaches the 27 ftceiling. The duct continues

down the ceiling for 100 ft with 4 ft x 2 ft size and then reduces to 3 ft x 2 ft for the last 100 ft.

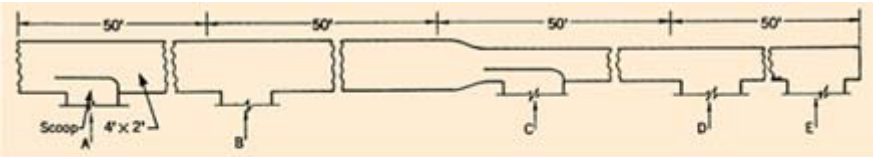

Figure 9 : Two-hundred foot duct delivering air to cold storage room. Scoop is needed at A to create sufficient static pressure. At B, static pressure is sufficient. At *C*, after reducer, a scoop is again needed to convert velocity pressure to static pressure

The velocity in the 4ft x 2 ft duct at the start is about 3,200 fpm, corresponding to 0.64 inch pressure, about 0.44 inch more than the 0.2 inch which corresponds to the 1,800 fpm fan outlet velocity. This results in a static pressure at the fan outlets of about 0.5 inch greater than in the duct as it starts across the room. However, the pressure at the beginning of the horizontal run of the 4 ft x 2 ft duct is about the same as the room (which we will call zero) so that the static pressure at the fan outlets is about 0.5 inch which is not excessive for a 200 ft distribution system.

The 4 ft x 2 ft duct runs about 100 ft, then changes to 3 ft x 2 ft for the last 100 ft, as shown in **Figure 9**. Actually there are outlets on both sides of the duct but, for convenience the illustration shows only those on one side.

With no static pressure at outlet A, a scoop is required to convert velocity pressure to static pressure. The cross section the scoop offers to the air stream is the same proportion of the whole-duct cross section, as the air flow from outlet A is to the total air flow at that point. As the air moves along the duct, the cfm decreases and since the duct size remains the same, the velocity decreases. As the velocity drops, a part (approximately 60% to 75% in cases of this type) of the energy becomes static pressure and this static pressure increase exceeds the friction loss. This results in a higher static pressure in the duct at points farther along in the direction of air flow.

At a point about 50 ft from the inlet, as indicated in **Figure 9**, the static pressure is enough above the room pressure to cause the air to leave the outlet at the proper velocity and no more scoops are necessary. After 50 ft more, the size is decreased, increasing velocity and decreasing static pressure. Therefore, scoops are used back of outlets for about the next 50 ft to the point where the static pressure has again increased to a value such that they are unnecessary.

The use of this design results in smaller duct at the beginning and less size changes. In many cases, if the duct is not too long, there may be no size changes at all. In some cases the static pressure changes very little for a good part or all of the duct length, as it does

between about 50 ft and 100 ft and the length from 150 ft to 200 ft of the duct shown in **Figure 9**.

In a certain textile plant, the distribution of 228,000 cfm to a room about 250 ft x 160 ft is taken care of by a 632 ft x 10 ft duct 250 ft long, running down the center of the 160 ft width. There are outlets on each side. The pressure in the duct at the beginning is less than the room pressure, but the reducing velocity results in a static pressure above the room for the last $^1/_2$ to $^2/_3$ of the length. Scoops are, of course, used until the static pressure reaches about $^1/_8$ inch, which is necessary for discharge from the grilles at the required velocity.

In the cases mentioned, the air was all delivered to a single room so that a variation of 5 or 10% above or below that desired from any outlet, would not appreciably affect the temperature or humidity in any part of the room. Since care was taken in the design, no dampers were installed in the duct in either case. Experience showed that dampers were not required. Had the outlets discharged to rooms where more precise distribution might be necessary or where it might not be possible to predict in advance the exact relative air rates, dampers probably would have been desirable for initial adjustment or later changes in adjustment to compensate for changed conditions.

When designing ductwork (or anything else) careful consideration of the laws of nature which apply and less tendency to merely follow what has become common practice, can often result in better operation and sometimes less costly construction. A little more time spent with the design will usually eliminate alterations after installation. It’s so much easier to change lines on paper than to alter ductwork or piping on the job.