JM Johns Manville

81-1-ISD   Economic Considerations of Heat Transfer on Sheet Metal Duct

Other bulletins have demonstrated the need to add insulation to cooling/heating ducts in order to achieve proper air distribution in the conditioned space. This bulletin deals with the heat gain/loss by the supply air and how it effects the overall performance and cost of the HVAC system.

To achieve the proper degree of comfort in a controlled environment, the design engineer begins by calculating the heating/cooling load of the space to be conditioned. Once the space load is calculated, the engineer, in the case of air-to-air HVAC systems, will determine the volume and conditions of the supply air to be delivered to that space. The summation of space loads with the proper load diversification will account for the total building load.

<u>Cooling Effect of the Supply Air</u>

The cooling effect or sensible cooling of the room or space by the supply air is determined by the following equation:

$$Q_S = M \times 1.1 \times (T_r - T_1)$$

Where:

Q_S = Sensible cooling/heating (Btu/hr)

M = Volume of supply air flow (cfm)

1.1 = Unit conversion constant combining the air density, heat capacity of the supply air, and minutes to hours (Btu/hr/cfm/°F)

T_r = Temperature of the room (°F)

T_1 = Temperature of the supply air leaving the duct (°F)

The temperature of the supply air is lowered by removing heat from the air in a heat exchanger or cooling coil.
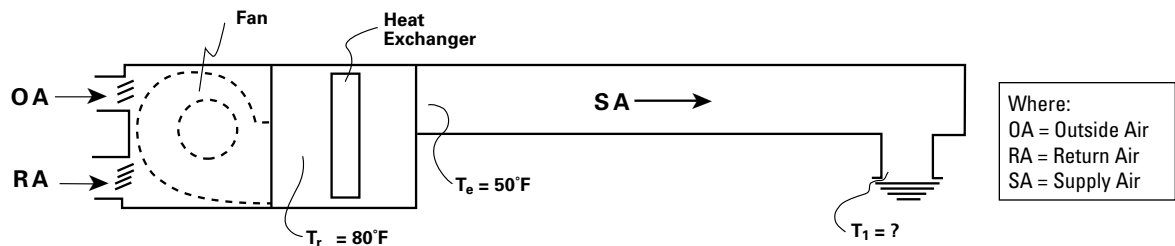
The size of the cooling coil is determined by the volume and condition of the supply air required by the space cooling load. For many reasons, one of which was discussed in the Engineering & Technical Bulletin dealing with condensation control on air conditioning ducts (80-1-ISD), the drop in the air temperature across the cooling coil is limited to 30°F ±5°F for most HVAC systems. The coil sensible cooling capacity is determined by the following equation:

$$Q_{sc} = M \times 1.1 \times (T_{ra} - T_e)$$

Since the temperature drop of the air across the cooling coil is limited, cooling loads can be thought of in terms of volume of air required. Since heat gained by the supply air in the air distribution duct will increase its temperature, the volume of air delivered to the conditioned space at the temperature leaving the duct still needs to be determined.

Heat Transfer Theory

In order to determine the correct air volume to the conditioned space, you must first find the temperature of the air leaving the duct. Heat gain/loss by a duct can be expressed by the following equation:

$$Q_d = U \times A \times \left[T_o - \left(\frac{T_e + T_1}{2} \right) \right]$$

Where:

U	=	Overall heat transfer coefficient [Btu/(hr•ft^2•°F)]
A	=	Area of the duct (ft^2)
T_o	=	Ambient temperature (°F)
T_e	=	Entering air temperature (°F)
T_1	=	Leaving air temperature (°F)

Heat gain/loss by the air in the duct can be expressed by the following equation:

$$Q_a = M \times (1.1) \times (T_1 - T_e)$$

Where:

M	=	Volume of supply air flow (cfm)
1.1	=	Unit conversion constant combining the air density, heat capacity of the supply air, and minutes to hours (Btu/hr/cfm/°F)

Under steady state conditions, the heat gain/loss by the duct is equal to the heat gain/loss by the air inside the duct. Therefore:

Duct heat (gain/loss) = Air heat (gain/loss)

Substituting into the above relation the equations for heat gain/loss…

$$Q_d = Q_a$$
$$U \times A \times \Delta T = \text{cfm} \times 1.1 \times \Delta T$$
$$U \times A \times \left[T_o - \left(\frac{T_e + T_1}{2} \right) \right] = \text{cfm} \times 1.1 \times (T_1 - T_e)$$

From the heating/cooling load requirements of the space to be conditioned and the heating/cooling coil specification, the values for the variables in the relation can be obtained. Only the temperature of the air leaving the duct (T_1) is an unknown value.

<u>Sample Problem</u>

As a sample problem, let's take a supply air duct with the following conditions:
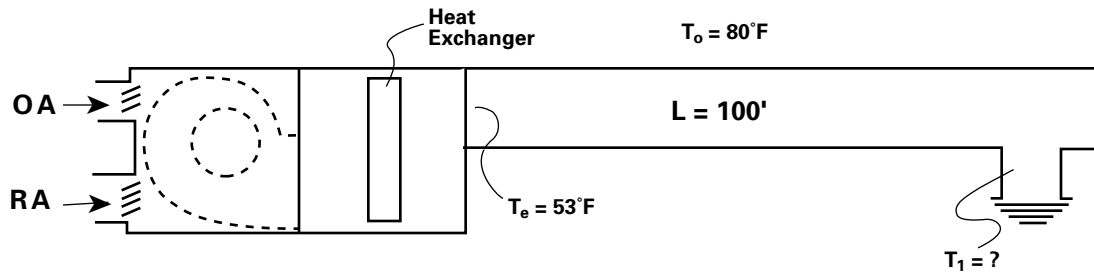
<u>Sheet Metal Duct</u>

Assumptions
Desired Room Temperature = 78°F
Room Cooling Load = 24,750 Btu/hr
Volume of Supply Air Flow (M) = 900 cfm (idealized with no heat gain in duct)

$$Q_d \quad = \quad U \times A \times \left[T_o - \left(\frac{T_e + T_1}{2} \right) \right]$$

Where:

Q_d	=	Heat gain by duct (Btu/hr)
U	=	0.64 for bare sheet metal according to NAIMA dynamic thermal test [Btu/(hr•ft²•°F)]
A	=	400 ft² (for a 12" x 12" x 100' long duct)
T_e	=	53°F
T_o	=	80°F
T_1	=	? °F

Sheet Metal Duct (cont'd)

$$Q_a = M \times 1.1 \times (T_1 - T_e)$$

Where:

$$Q_a = \text{heat gain by air}$$

$$M = 900 \text{ cfm}$$

$$T_e = 53°F$$

From previously established relation,

$$U \times A \times \left[T_o - \left(\frac{T_e + T_1}{2} \right) = M \times 1.1 \times (T_1 - T_e) \right]$$

and substituting the known variables,

$$0.64 \times 400 \times \left[80 - \left(\frac{53 + T_1}{2} \right) \right] = 900 \times 1.1 \times (T_1 - 53)$$

Solving for T_1:

$$T_1 = 59.2°F$$

or an increase in the supply air temperature of 6.2°F.

1" Thick Fiber Glass Duct

Following the same procedure for a duct under the same conditions but fabricated from 1" thick Mat-Faced Micro-Aire® Duct Board with a U of 0.19 from the NAIMA Dynamic Thermal Test:

$$U \times A \times \left[T_o - \left(\frac{T_e + T_1}{2} \right) \right] = M \times 1.1 \times (T_1 - T_e)$$

For 1" thick fiber glass duct with:
ID = 12" x 12"
OD = 14" x 14"
A = 466.6 ft^2

substituting for the known variables,

$$0.19 \times 466.6 \times \left[80 - \left(\frac{53 + T_1}{2} \right) \right] = 900 \times 1.1 \times (T_1 - 53)$$

and solving for T_1:

$$T_1 = 55.3°F$$

or an increase in the supply air temperature of 2.3°F.

<u>Sensible Cooling Effect of the Air Supply</u>

The cooling effect of the air delivered to the conditioned space by the bare sheet metal duct is calculated as follows:

$$Q_s = M \times 1.1 \times (T_r - T_1)$$

Where:

$$T_r = \text{Temperature of the room (°F)}$$

$$T_1 = \text{Temperature of the supply air leaving the duct (°F)}$$

Therefore

$$Q_s = 900 \times 1.1 \ (78 - 59.2)$$

$$Q_s = 18,612 \ \text{Btu/hr}$$

Since it was determined that the cooling load of the conditioned space was 24,750 Btu/hr, the air volume required to satisfy the load at the air leaving temperature is calculated as follows:

<u>For Sheet Metal Duct</u>:

$$M = \frac{Q_s}{1.1 \ (78 - 59.2)}$$

$$= \frac{24,750}{1.1 \ (78 - 59.2)}$$

$$= 1197 \ \text{cfm}$$

<u>For Fiber Glass Duct</u>:

$$M = \frac{24,750}{1.1 \ (78 - 55.3)}$$

$$= 991 \ \text{cfm}$$

This is a difference in the air volume of sheet metal versus fiber glass of 206 cfm or 20.8%.

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Conclusion

As shown by the sample calculation, in order to meet the required room load, the HVAC system with a bare sheet metal duct required an additional air volume of 33%. This compared to an additional air flow of 10% for a fiber glass duct system.

By using these percentages for a building with a cooling load of 200 tons or 24,000,000 Btu/hr, the total air volume for an HVAC system with bare sheet metal ducts would be 118,195 cfm. For an HVAC system with fiber glass ducts, the required air volume would be 97,893 cfm, a difference of 20,302 cfm.

In actual practice, HVAC engineers increased the building load by 15% in order to compensate for the duct heat gain and leakage.

Because fiber glass duct will increase overall efficiency of the HVAC system, the engineer can reduce the capacity of the HVAC equipment and distribution system. This will allow the engineer to offer the owner the following benefits:

1. Lower initial capital outlay for the purchase of HVAC equipment.

2. Savings in the cost of the air distribution system.

3. Lower heating and cooling costs.

4. A quieter HVAC system.

5. No condensation problems.

6. A better space temperature control.

This is a partial list and additional benefits can be added. Another benefit is to have a company like Johns Manville behind the specified materials and components.

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