A Practical Method for Estimating the Thermal Efficiency of Residential Forced-Air Distribution Systems

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Forced-air distribution systems with significant portions of ductwork running through unconditioned spaces have the potential for large heat losses. The magnitude of these losses depends on complex interactions of conduction and air leakage and between the supply and return portions of the distribution system. The interaction with natural infiltration is an additional complicating factor. To date there have been no practical models available for estimating the magnitude of these losses.

This paper presents a general-purpose, practical method for estimating the overall thermal efficiency of residential forced-air distribution systems. This method provides quick estimates of the impact of various loss components (e.g., conduction vs. air leakage and supply vs. return). The method accounts for the interaction of supply-side and return-side losses, as well as differential pressurization effects on the home due to duct air leaks.

In the basic form presented here, thermal regain is not included and duct leakage is assumed to be at the registers. A simple modification to the model for leakage in other locations is presented. Sensitivity of the model to different components is shown, and applications to real situations are discussed.

INTRODUCTION AND BACKGROUND

Recent studies have documented the importance of duct thermal losses. For instance, Olson et al. (1993) studied 22 all-electric homes in the Pacific Northwest. These homes were selected to have at least half of the ductwork in unconditioned spaces. The average duct heat delivery efficiency of these homes was 56% while the average duct system efficiency (which includes regain from duct losses to the conditioned space) was 71%, compared to 98% duct system efficiency in two homes with all interior ducts. In a subsequent study six of these homes, selected specifically for large air leakage, underwent aggressive duct air sealing which reduced the air leakage from the ductwork to outside by an average of 70%, improved both the delivery and system efficiency by an average of 16%, and reduced the system efficiency losses by an average of 44% (Palmiter, Olson & Francisco 1995).

There is growing interest in the efficiency of duct systems on the part of utilities, weatherization programs, building code regulators and others. However, the relative magnitudes of supply losses versus return losses and conduction losses versus air leakage losses has been unclear, leading to uncertainty concerning which losses should receive the most attention with regard to both repair efforts and future construction design. The situation is further complicated by the fact that the return losses affect the supply losses, and air leakage affects both the conduction losses and the infiltration rate of the home.

This paper presents a general-purpose, practical method for estimating the overall steady-state thermal efficiency of residential forced-air distribution systems. This method differentiates between losses on the supply side and the return side of the distribution system, and provides quick estimates of the impact of various losses (e.g., the relative importance of a 10% return air leak vs. a 10% supply air leak, or a 10% supply conduction loss vs. a 10% supply air leak). The method also accounts for the interaction of supply-side and return-side losses, as well as differential pressurization effects on the home due to duct air leaks. This paper summarizes the model developed in Palmiter & Francisco (1996), which provides a more detailed derivation and applications.

The model, as presented here, is restricted to estimating the efficiency under steady-state operation of the heating equipment, and to situations in which the temperatures of the buffer zones (e.g., attics, crawl spaces, etc.) are known. The effects of duct losses on the buffer zone temperatures and the concomitant reduction of heat losses from the conditioned space to the buffer zone (regain effect) are not included. This is an important issue which requires further study. Palmiter and Francisco (1996) extends the model to allow estimation of crawl space temperatures, and prediction of duct efficiency and space heat use over the heating season for both furnaces and air-source heat pumps.

OVERVIEW OF DUCT LOSSES

Heat loss occurs through two mechanisms, conduction loss through the duct walls to the buffer zones and air leakage out of the supply ducts or into the return ducts. The conduction losses have been recognized for a long time and ductwork in newer homes is almost always insulated to reduce this loss. The importance of air leakage has only been recognized recently.

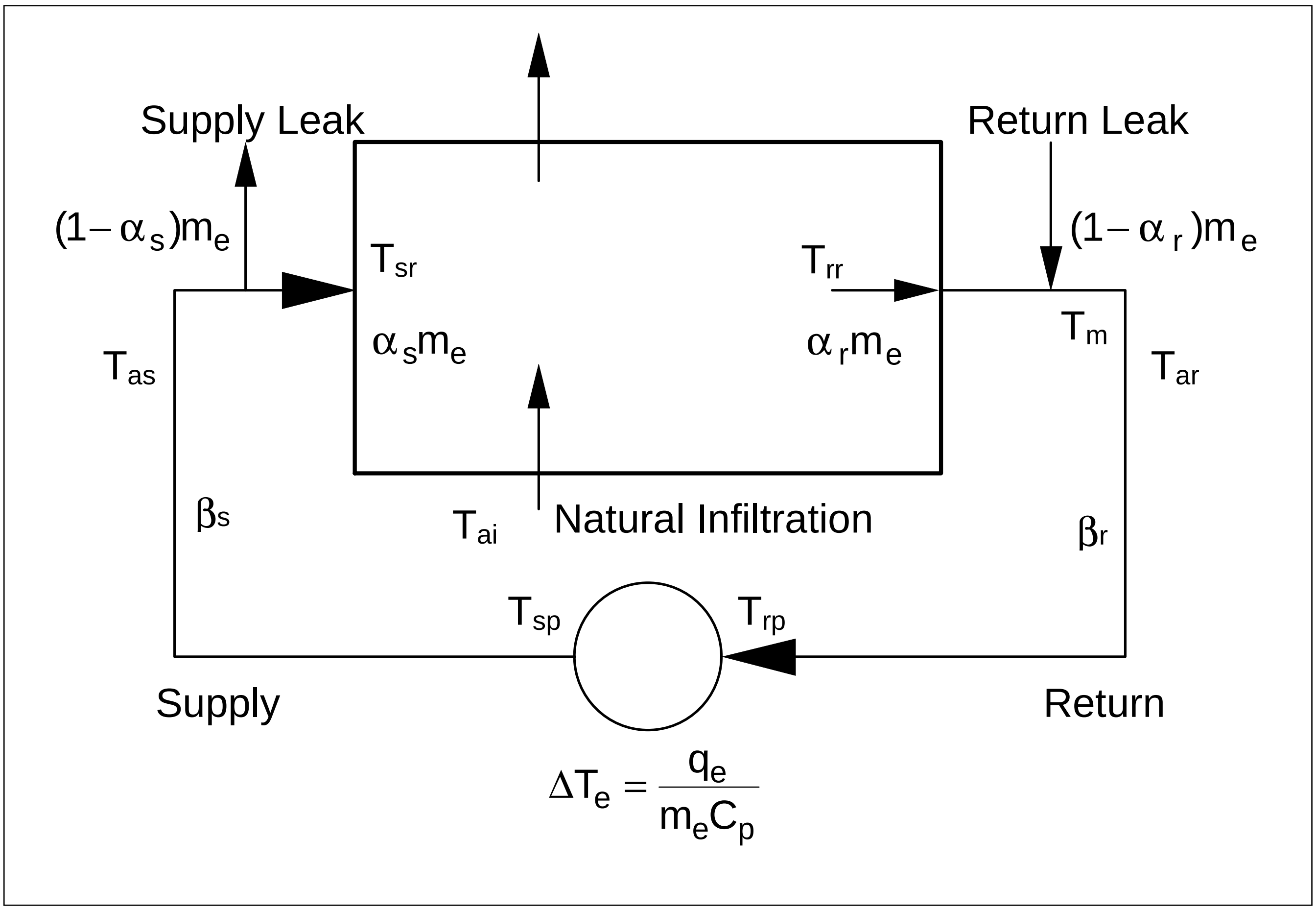
A simplified schematic of a one-pipe duct system is given in Fig. 1. In this figure, the ductwork is all located in two buffer zones, one on the return side at temperature T_{ar} and one on the supply side at temperature T_{as}. The temperature labeled T_{rr} is the temperature of the air entering the return register and the temperature labeled T_{sr} is the temperature of the air leaving the supply register. The heating equipment, represented by the large circle below the heated space, is characterized by the mass flow rate m_e and by the output heating capacity q_e which determine the temperature rise from the return plenum to the supply plenum, ΔT_e.

The derivation of the model is simplified if we assume that air leakage occurs only at the register end of the pipes; a fraction $(1 - \alpha_s)$ of the mass flow rate at temperature T_{sr} leaks from the supply pipe just before the air enters the house and a fraction $(1 - \alpha_r)$ of the mass flow rate at temperature T_{ar} leaks into the return just after the air leaves the home. Leakage at this position will result in smaller thermal losses than leakage at any other position, while leakage at the plenum end of the ducts will result in greater thermal losses than at any other position. For moderate leakage rates the impact of position is small. We show later how to modify the model to remove this restriction.

As noted above, return losses affect supply losses. When T_{ar} is less than T_{rr}, conduction or air leakage on the return side will reduce the return plenum temperature which will in turn reduce the supply plenum temperature, so that return losses will reduce supply losses.

The schematic also shows the natural infiltration and exfiltration of air with respect to the conditioned space, assuming the indoor air is warmer and the infiltration is primarily

Figure 1. *Simplified Schematic for Duct Losses*



buoyancy driven (stack effect) with air entering the space near the bottom and exiting near the top. If the return air leakage is not equal to the supply air leakage, the leakage is said to be unbalanced. If the supply leakage is larger than the return leakage, the air handler removes more air from the space than it delivers, which will reduce the internal pressure of the space resulting in an increase in infiltration and a decrease in exfiltration. If the return leakage exceeds the supply leakage, there will be an increase in exfiltration and a decrease in infiltration. If the supply and return leakages are balanced, the infiltration and exfiltration rates across the envelope are unaffected by the air handler. This interaction of duct leaks with natural infiltration has a significant impact on overall thermal losses.

In general, the thermal losses from the duct system will increase the temperatures of the buffer zones unless the zones correspond to outdoor ambient. If the home is thermally connected to the buffer zones, there will also be a reduction in heat loss from the conditioned space to the buffer zones. This is an important obstacle to the calculation of the overall system efficiency, since it requires accurate thermal simulation of the buffer zones, as well as the distribution system. In the crawl space case, the large thermal storage effect of the ground must be considered. In the attic case, solar radiation and infra-red sky radiation are important. In both cases, the air flow rates to and from ambient, as well as to and from the conditioned space are needed and difficult to calculate. This issue requires further study, see Palmiter and Francisco (1996).

Although the discussion of the model is presented in terms of a duct system which is located entirely in the buffer zones, in most homes there will be a significant portion of the ductwork in interior walls and floors. In this case, all estimates of air leakage must be based on the leakage rate to outside and the conduction losses must be based only on that portion of the ductwork which is located in the buffer zones.

CONDUCTION EFFICIENCY

Assume that the flow through a pipe is at steady-state, and that the ambient air temperature is constant. Then standard heat-exchanger theory for the case where one temperature is constant leads to the following definition of the conduction efficiency.

$$\beta = \frac{T_{out} - T_a}{T_{in} - T_a} = \exp\left(-\frac{UPL}{mC_p}\right) \qquad (1)$$

so that

$$T_{out} - T_a = \beta(T_{in} - T_a) \qquad (2)$$

where

- β = the conduction efficiency of the pipe
- T_a = the ambient temperature (F)
- T_{out} = the outlet temperature (F)
- T_{in} = the inlet temperature (F)
- U = the heat transfer coefficient (Btu/(h·F ft^2))
- P = the inside perimeter of the pipe (ft)
- L = the length of the pipe (ft)
- m = the mass flow rate of the air in the pipe (lbm/h)
- C_p = the specific heat (Btu/(lbm·F))

This also agrees with the equation for duct heat loss given in ASHVE (1951, p. 685). The conduction efficiency provides a simple means of calculating the outlet temperature, given the other parameters. It should be remembered that β is a function of the mass flow rate in the pipe and that it gives the fraction of the temperature difference between the inlet and ambient temperatures which remains at the outlet.

When the exponent in Eq. (1) is small (e.g. with a thermally short pipe) this can be approximated as

$$\beta \approx 1 - \frac{UPL}{mC_p} \qquad (3)$$

For thermally short pipes, which includes most insulated ducts, this shows that the fractional losses $(1 - \beta)$ are proportional to U, P and L and inversely proportional to the mass flow rate.

The exponent in Eq. (1) can also be rewritten as

$$\frac{UPL}{mC_p} = \frac{4L}{\rho V C_p d R} \qquad (4)$$

where

- ρ = the density of the air in the pipe (lbm/ft^3)
- V = the velocity of the air in the pipe (ft/h)
- d = the diameter of the pipe (ft)
- R = the thermal resistance of the pipe ((ft^2·F·h)/Btu)

This expression shows that for fixed velocity, length, fluid, R-value, inlet temperature, and ambient temperature, the fractional loss is approximately inversely proportional to the diameter. This is pertinent to residential duct systems because the velocity in the larger pipes is typically comparable to or somewhat greater than that in smaller pipes. For instance, under these conditions, the fractional loss in a 6 inch supply pipe will be 2.67 times larger than in a 16 inch return pipe of the same length.

This solution for conduction efficiency is easily extended to more complex branching duct systems with pipes of varying size, insulation and flow rates. For the typical branching tree structure of a residential supply system, one can proceed as follows. Starting at the furnace calculate β for the section of trunk to the first take-off using the design values for length diameter, insulation, and flow rate. In the same way, calculate a β for each section of trunk between take-offs, and for each final branch run to a register. The overall β for a particular register is simply the product of all the βs for the trunk segments up to that register and the branch to the register. Each register has its own β and the overall β for the entire supply system is the mass-flow-rate weighted average of the register βs. The same procedure can be used for multiple returns. A by-product of the calculation is the temperature at each take-off and register, providing a quick check for cold delivery temperatures. Such a procedure could be easily incorporated into a computerized duct design program like the one distributed for Manual-D by the Air Conditioning Contractors of America.

The procedure just described assumes the entire supply or return subsystem is exposed to the same ambient temperature. However, it is not difficult to further extend it to allow for different segments to be exposed to different ambient temperatures. It is important to remember that for the purpose of determining the thermal losses to the buffer zone, the conduction losses are only calculated for the portion of the duct system which lies in the buffer zone.

A simple method of estimating an overall β for the supply or return subsystem, which will be adequate in most cases, is to calculate the total *UPL* product of the system and use this along with m_e in equation (1) for β.

AIR LEAKAGE EFFICIENCY

Assume that the leakage is concentrated at one place along the pipe, and that the percentage loss is a constant. The air leakage efficiency of the pipe is defined as

$$\alpha = \frac{m_{out}}{m_{in}} = 1 - \text{Leak Fraction} \tag{5}$$

where

- α = the air leakage efficiency of the pipe
- m_{out} = the outlet mass flow rate from the pipe (lbm/h)
- m_{in} = the inlet mass flow rate into the pipe (lbm/h)

Getting a value for the actual air leakage efficiency of the pipe requires a measurement of the air leakage to outside and knowledge of the mass flow rate delivered by the equipment.

OVERALL DUCT EFFICIENCY WITH COMBINED CONDUCTION AND AIR LEAKAGE

Consider a house modeled as in Fig. 1, with the air leaks at the registers and the ducts in unconditioned spaces, and where

- ΔT_e = the temperature rise across the equipment (F)
- q_e = the capacity of the equipment (Btu/h)
- m_e = the mass flow rate through the equipment (lbm/h)
- T_{sp} = the temperature of the air in the pipe at the supply plenum (F)
- T_{rp} = the temperature of the air in the pipe at the return plenum (F)
- T_{as} = the temperature of the air around the supply duct (F)
- T_{ar} = the temperature of the air around the return duct (F)
- T_{sr} = the air delivery temperature from the supply register to the house (F)
- T_{rr} = the temperature of the air in the pipe at the return register (F)
- T_m = the temperature of the air in the return pipe just past the leak (F)
- T_{ai} = the temperature of infiltration air (F)

and the subscripts *s* and *r* refer to the supply and return ducts, respectively.

From Eq. (1) the conduction efficiency for the supply and return sides are, respectively

$$\beta_s = \frac{T_{sr} - T_{as}}{T_{sp} - T_{as}} \quad , \quad \beta_r = \frac{T_{rp} - T_{ar}}{T_m - T_{ar}} \tag{6a,b}$$

If there is no interaction with natural infiltration inside the house, the overall duct system efficiency η can be defined as the ratio of the heat delivery rate to the house q_{sr} to the output capacity of the equipment q_e of the equipment,

$$\eta = \frac{q_{sr}}{q_e} = \frac{\alpha_s m_e C_p (T_{sr} - T_{rr})}{m_e C_p \Delta T_e} \tag{7}$$
$$= \frac{\alpha_s (T_{sr} - T_{rr})}{\Delta T_e}$$

The difference between the temperature of the air in the return plenum and the air around the return duct can be expressed in terms of the return-side air leakage efficiency, the return-side conduction efficiency, and the difference between the temperature of the house and the temperature of the air around the return duct by using Eq. (6b) and by setting up an energy balance before and after the return-side leak. This gives

$$T_{rp} - T_{ar} = \frac{T_{rp} - T_{ar}}{T_m - T_{ar}} \frac{T_m - T_{ar}}{T_{rr} - T_{ar}} (T_{rr} - T_{ar}) \tag{8}$$
$$= \beta_r \alpha_r (T_{rr} - T_{ar})$$

By multiplying Eq. (7) by ΔT_e and by using Eq. (6a), Eq. (8), and the relationship

$$\Delta T_e = T_{sp} - T_{rp} \tag{9}$$

the definition for the overall duct efficiency can be written as

$$\begin{aligned} \eta \Delta T_e = \alpha_s[\beta_s\{\Delta T_e + [T_{ar} - T_{as} \\ + (\beta_r \alpha_r (T_{rr} - T_{ar}))]\} \\ - T_{rr} + T_{as}] \end{aligned} \tag{10}$$

In this form, which can be regarded as a one line derivation of the duct model, all terms have dimensions of temperature, and by starting from the innermost term and expanding, each successive operation results in a physically meaningful temperature difference. For example, as shown above, the innermost term is equal to the difference between the return plenum temperature and the return zone ambient temperature. At the next level we get the difference between the return plenum and the supply ambient. Adding ΔT_e gives the temperature difference between the supply plenum and the supply ambient. Multiplying by the supply conduction efficiency gives the temperature difference between the supply register and the supply ambient, and, finally, subtracting the return register temperature and adding the supply ambient temperature gives the temperature difference between the supply register and the return register, which agrees with Eq. (7).

Eq. (10) can be rewritten in the form

$$\begin{aligned} \eta = \alpha_s \beta_s - \alpha_s \beta_s (1 - \alpha_r \beta_r) \frac{\Delta T_r}{\Delta T_e} \\ - \alpha_s (1 - \beta_s) \frac{\Delta T_s}{\Delta T_e} \end{aligned} \tag{11}$$

where

$$\Delta T_r = T_{rr} - T_{ar}\,, \Delta T_s = T_{rr} - T_{as} \tag{12}$$

This form has the property that each term is dimensionless, and is attractive in that the supply and return temperature difference terms are separated and linear. It also has the feature that the only required measured temperatures are those of the supply and return zones and the return register.

The model is easily extended to allow for a fraction f of the air leak to occur at the register end of the duct and the remainder at the plenum end. The equations are the same as given above, but the value of β is modified as follows:

$$\beta = \exp\left(- \frac{UPL}{(f + (1 - f)\alpha) m C_p} \right) \tag{13}$$

There are several important implications illuminated by Eq. (11). One is that, regardless of the temperature differences the overall efficiency can be no better than the product of the supply conduction efficiency and air leakage efficiency. On the other hand, the return-side does not have any impact if the house temperature and the temperature of the air around the return duct are the same. Another implication of Eq. (11) is that as the temperature difference across the equipment decreases the overall efficiency will also decrease. This raises special concern for heat pumps, where the temperature change is usually much lower than for other types of air handlers, such as a furnace. Also, this suggests that as the capacity of the equipment decreases or the flow rate increases the overall efficiency will be reduced.

HEAT LOSS TO BUFFER ZONES

Estimation of the impact of duct losses on the buffer zone temperatures (e.g., when the duct model is incorporated into a simulation model) requires equations for the heat transfer from the duct system to the buffer zones. The heat loss q_{LS} from the duct system to the supply zone can be written as

$$\begin{aligned} q_{LS} = [(1 - \alpha_s \beta_s)(\Delta T_e) \\ - (1 - \alpha_r \beta_r) \Delta T_r) \\ + \alpha_s (1 - \beta_s) \Delta T_s] m_e C_p \end{aligned} \tag{14}$$

and the heat loss q_{LR} from the duct system to the return zone can be written as

$$q_{LR} = (1 - \beta_r) \Delta T_r m_e C_p \tag{15}$$

EXTENSION OF DUCT MODEL TO INCLUDE INTERACTION WITH NATURAL INFILTRATION

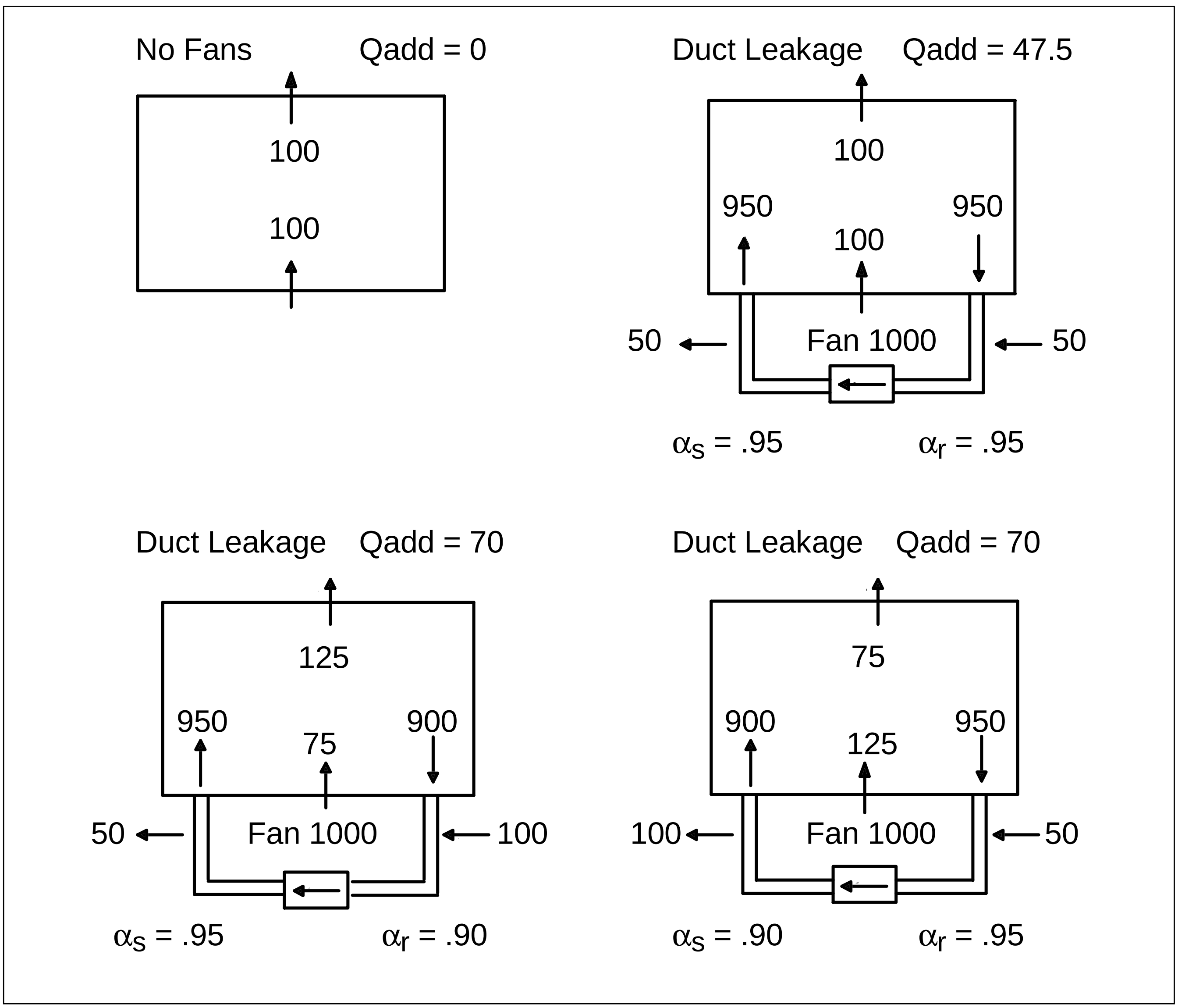
If the supply and return air leakage efficiencies are equal, such that the mass flow through the leaks is the same, then the pressure within the building is unchanged by operating the system and there is no interaction between the distribution system and natural infiltration. However, when the leakage in the supply and return ducts is unbalanced the internal pressure does change, altering the infiltration and exfiltration through the envelope. This interaction between the ducts and the natural infiltration of the house affects the overall duct system efficiency. Designating the loss due to interaction with natural infiltration as η_{in} and subtracting this term from Eq. (11) gives

$$\eta = \alpha_s\beta_s - \alpha_s\beta_s(1 - \alpha_r\beta_r)\frac{\Delta T_r}{\Delta T_e} - \alpha_s(1 - \beta_s)\frac{\Delta T_s}{\Delta T_e} - \eta_{in} \tag{16}$$

The interaction term can be estimated using the fan model developed in (Palmiter and Bond 1991a; Palmiter & Bond 1991b; Palmiter & Bond 1992) which has also been incorporated in ASHRAE (1993). It is important to note that the natural infiltration rate to be used in this context includes only infiltration through the thermal envelope; it should not include infiltration through leaks in the ducts. The natural infiltration rate can be determined by a tracer decay test with the supply and return registers sealed or estimated from pressure test data and an infiltration model. Also, note that we have identified T_{rr} with the temperature of exfiltrating house air.

The effects on air flow rate though the home are illustrated in Fig. 2. There are two distinct cases: small unbalanced leakage where the unbalanced duct leakage is less than twice the natural infiltration rate, and large unbalanced leakage. The small unbalanced leakage case is common, and leads to an easier calculation, as it is not necessary to estimate the natural infiltration rate. The equations for η_{in} as well as the added flow through the home are given below. We also

Figure 2. *Effect of duct leakage on natural infiltration.*



show the overall efficiency equation for the simple case where all of the temperature differences are equal.

Case I: Small unbalanced leakage

$$\frac{1}{2}\left|\alpha_r - \alpha_s\right| \le \frac{Q_{nat}}{Q_e} \tag{17}$$

$$\eta_{in} = \frac{1}{2}(\alpha_r - \alpha_s)\frac{\Delta T}{\Delta T_e} \tag{18}$$

$$Q_{add} = \left[\frac{1}{2}(\alpha_r + \alpha_s) - \alpha_r\alpha_s\right]Q_e \tag{19}$$

where Q_{nat} is the natural infiltration flow rate, Q_{add} is the additional air flow through the home due to operation of the air handler, Q_e is the flow through the air handler, and

$$\Delta T = T_{rr} - T_{ai} \tag{20}$$

When $\Delta T_r = \Delta T_s = \Delta T$ Eq. (16) reduces to

$$\eta = \alpha_s\beta_s - \left[\frac{1}{2}(\alpha_r + \alpha_s) - \alpha_s\beta_s\alpha_r\beta_r\right]\frac{\Delta T}{\Delta T_e} \tag{21}$$

Case II: Large unbalanced leakage

$$\begin{aligned} &\frac{1}{2}(\alpha_{max} - \alpha_{min}) \ge \frac{Q_{nat}}{Q_e}\ , \\ &\alpha_{max} = \max(\alpha_s,\alpha_r)\ , \\ &\alpha_{min} = \min(\alpha_s,\alpha_r) \end{aligned} \tag{22}$$

$$\eta_{in} = \left[\alpha_{max} - \alpha_s - \frac{Q_{nat}}{Q_e}\right]\frac{\Delta T}{\Delta T_e} \tag{23}$$

$$Q_{add} = [\max(\alpha_r,\alpha_s) - \alpha_r\alpha_s]Q_e - Q_{nat} \tag{24}$$

When $\Delta T_r = \Delta T_s = \Delta T$ Eq. (16) reduces to

$$\eta = \alpha_s\beta_s - \left[\alpha_{max} - \alpha_s\beta_s\alpha_r\beta_r - \frac{Q_{nat}}{Q_e}\right]\frac{\Delta T}{\Delta T_e} \tag{25}$$

The heat loss due to infiltration interaction, q_{LH}, is

$$q_{LH} = \eta_{in} m_e C_p \Delta T_e \tag{26}$$

AN EXAMPLE

Consider a house with air leaks as shown in Fig. 1, with ten 12 ft. long, 6 in. diameter supply ducts in a radial arrangement, one 15 ft. long, 15 in. diameter return duct, and an equipment volumetric flow rate of 1000 cfm. Assume that the ducts are insulated to R-4. For calculating the supply-side conduction efficiency, either multiply the length or divide the flow rate by ten. Then, using Eq. (1),

$$\beta_s = 0.957 \quad , \quad \beta_r = 0.986$$

If the equipment output is 51195 BTU/h (15 kW) then the temperature rise across the equipment is 47.4 F. Consider a house heated to 65 F with an outdoor temperature of 41.3 F, for a temperature difference of 23.7 F. Under these conditions, if the temperatures around the supply and return ducts are the same as outdoor temperature,

$$\frac{\Delta T_s}{\Delta T_e} = \frac{\Delta T_r}{\Delta T_e} = \frac{\Delta T}{\Delta T_e} = 0.5$$

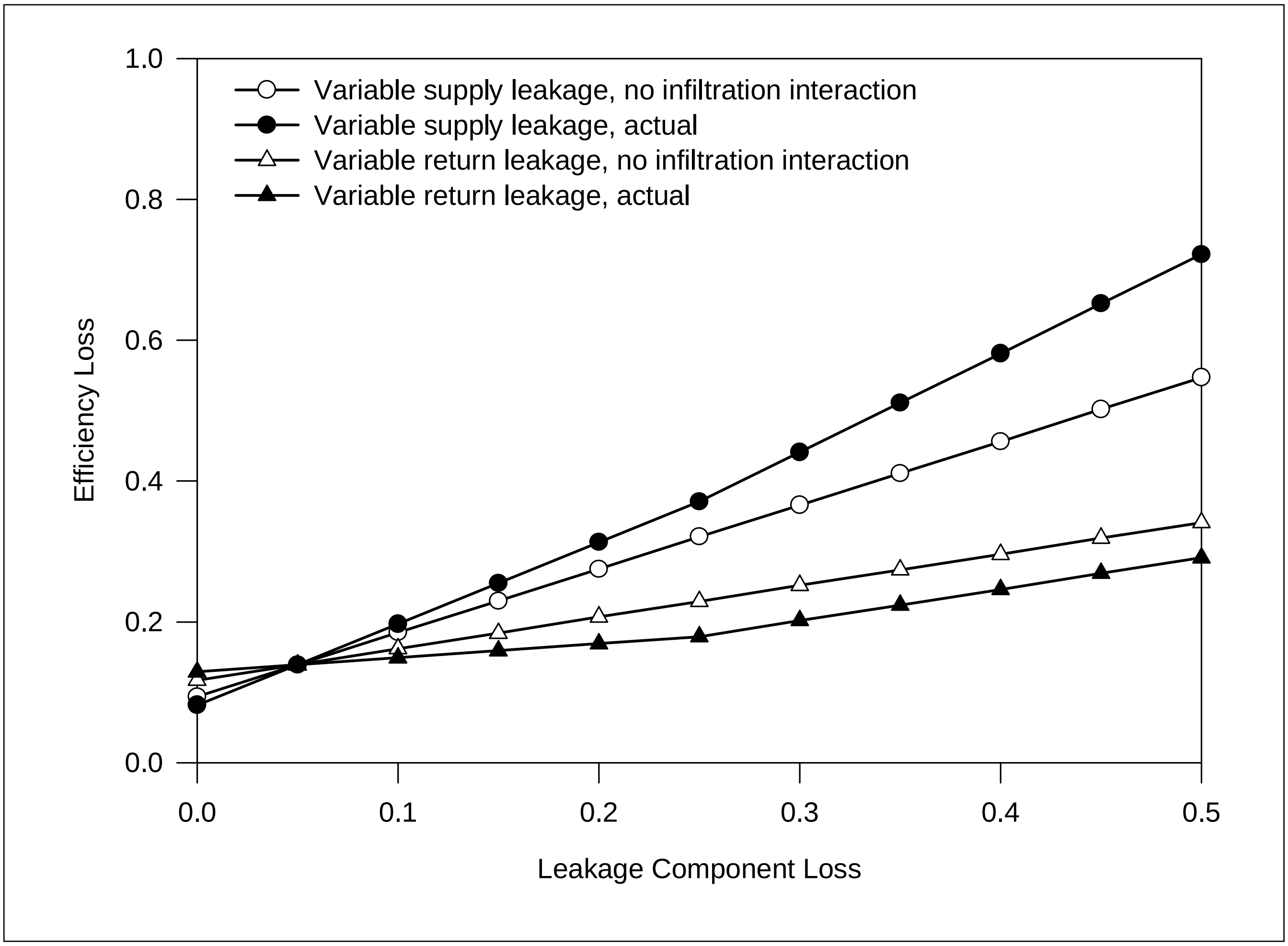
Further, assume that natural infiltration is 100 cfm. Then, using Eq. (21) and Eq. (25), and alternately holding the supply and return air leakage efficiencies constant at 0.95 while letting the other vary from 0.5 to 1, provides the efficiency loss results shown in Table 1. For comparison results are given both for leaks at the register end of the ducts and for leaks at the plenum end of the ducts. It is evident that the location of the air leakage has only a small effect on the efficiency loss.

These results are presented in graphical form in Fig. 3, which for comparison also includes the efficiency losses for these

Table 1. *Effects on Overall Efficiency of Varying Air Leakage Efficiencies*

AIR LEAK EFFICIENCY		OVERALL EFFICIENCY LOSS (%)	
Supply	Return	Leak at House	Leak at Plenum
1.00	1.00	7.1	7.1
1.00	0.95	8.2	8.2
0.90	0.95	19.7	20.4
0.80	0.95	31.3	32.5
0.70	0.95	44.1	45.9
0.60	0.95	58.1	60.6
0.50	0.95	72.2	75.2
0.95	1.00	12.9	13.3
0.95	0.90	14.9	15.3
0.95	0.80	16.9	17.3
0.95	0.70	20.2	20.6
0.95	0.60	24.6	25.2
0.95	0.50	29.1	29.7

Figure 3. *Effect of infiltration interaction on overall efficiency for example house. Supply and return leakages varied independently with the other held at 5%.*



two cases when the infiltration interaction term is not considered (i.e. use Eq. (11)). Note that when the return leak is greater than the supply leak, the infiltration interaction actually improves the overall efficiency. This is because the larger return leak pressurizes the home, which reduces the infiltration load.

EFFECT OF CHANGE IN MASS FLOW RATE

As noted previously, β and ΔT_e are functions of the mass flow rate through the air handler. The equations in the form given are not valid for situations where the flow rate is being varied. In these cases, the βs and ΔT_e must be replaced by their definitions. For the example home, the effect of varying the flow rate with constant output capacity is shown in Fig. 4. With no air leakage, the efficiency increases with flow rate, approaching an asymptotic limit of 0.978. With air leakage there is an optimum flow rate in the range from 800 to 1200 cfm. The optimum flow decreases with increasing air leakage. Flow rates below about 500 cfm result in very poor performance.

CONCLUSIONS

A simple engineering model of duct efficiency has been developed which accounts for supply and return interactions and the interaction of duct air leakage with natural infiltration. Although incomplete with regard to buffer zone temperatures, the model equations can be embedded in a multizone building simulation model. The combined model can then be used for simultaneous estimation of all pertinent quantities as they vary throughout the year.

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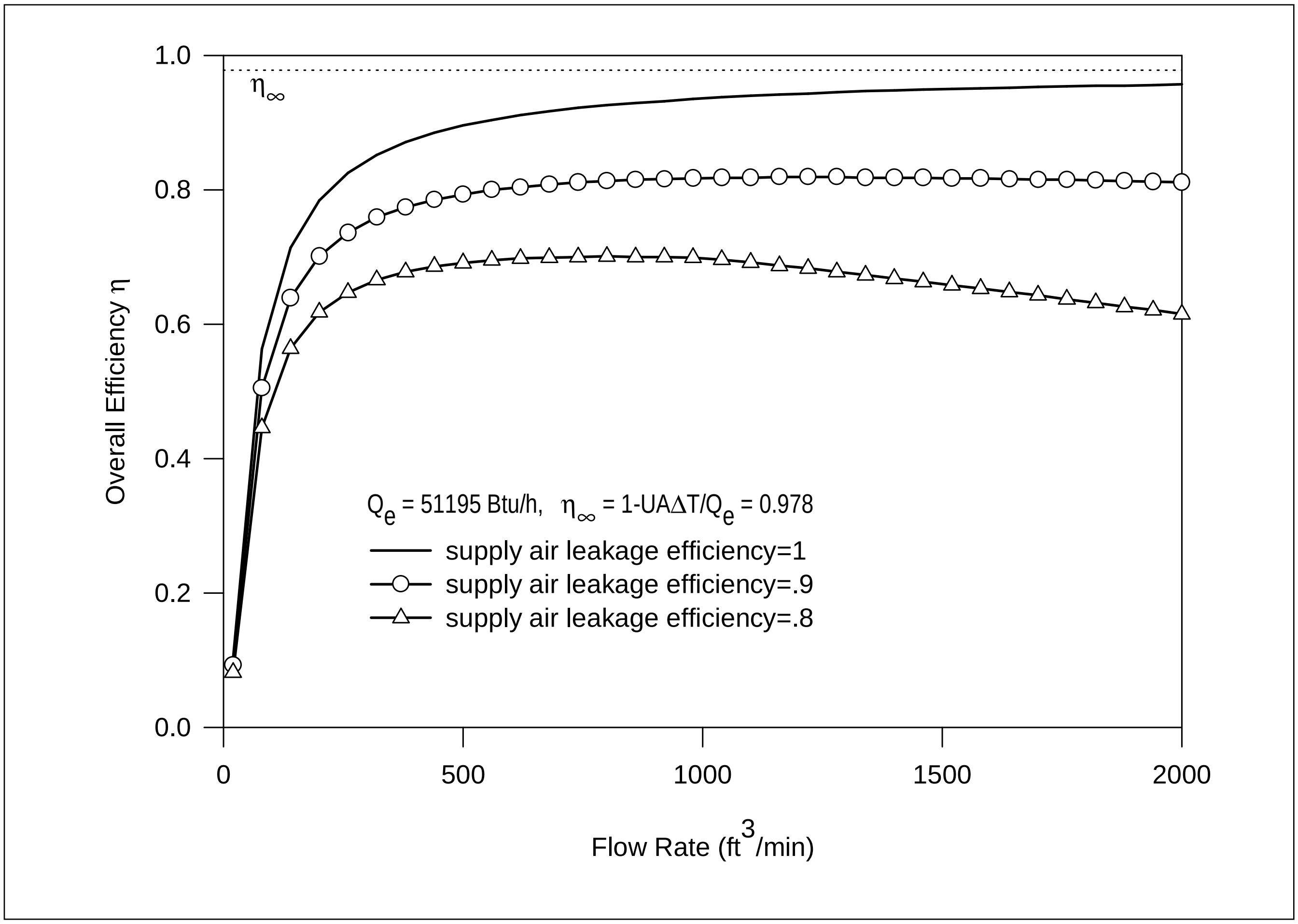
Figure 4. *Effect of varying equipment flow rates on overall efficiency at different supply air leakage efficiencies, for example house.*



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