Duct Efficiency under Full-Load or Modulating Conditions: Implications for Heat Pump Performance

Larry Palmiter and Erin Kruse, Ecotope Inc.
Paul Francisco, University of Illinois, Building Research Council

ABSTRACT

Ductwork in unconditioned spaces results in annual energy losses of 15 to 30% for space heating in the Pacific Northwest. This paper presents a further development of the distribution efficiency equations in showing that under full-load (or design) conditions the distribution efficiency is nearly independent of outdoor temperature. However, when there are internal gains, the full-load efficiency tends toward zero as the outdoor temperature approaches the heating balance-point. The full-load efficiency also applies in the case where there is a constant airflow rate through the equipment, but the equipment capacity is modulated to just meet the space-conditioning load.

The full-load distribution efficiency is lower than that which applies to equipment of larger capacity that is cycling off and on. Residential air-source heat pumps, when using both the compressor and auxiliary strip heat, modulate in such a way that the distribution efficiency approximates the full-load efficiency. This leads to lower duct efficiencies relative to gas or electric furnaces employed with the same duct system. The full-load efficiency also applies for the case of continuous air handler fan operation

The theory is illustrated with an example of detailed performance calculations for a heat pump in a Pacific Northwest city. The possibility of using the full-load distribution efficiency as a universal (i.e., valid in all locations) rating method for duct efficiency is also mentioned.

Introduction

This paper extends previously formulated equations for distribution efficiency to the situation where the useful heat delivered to the conditioned space is just equal to the load. This can be referred to as the full-load efficiency. It is the efficiency realized under design conditions provided there is no over-sizing or safety factor.

However, the basic assumptions are that the air handler fan runs continuously and the equipment capacity varies to meet the load. The equipment will be running at full capacity for the design condition case; however there are other situations that also meet the assumptions. One case of special importance is heat pumps operating below the compressor balance point. This is usually modeled by assuming the compressor and air handler fan run continuously while the auxiliary strip heat cycles off and on as needed to meet the house load.

The assumptions are also met for the case where the air handler fan is operated continuously (often recommended by manufacturers of electronic air cleaners, and also by some HVAC installers who believe continuous fan operation promotes comfort). It should be noted that in the case of continuous fan operation, the full-load efficiency applies only when the equipment capacity is greater than the fan heat (the fan heat is equal to the total power consumed by the fan). Below that point, the system capacity is constant and equal to the fan heat.

The full-load efficiency may therefore also be called the modulating efficiency. Note that the assumption here is that the capacity is modulated at a constant flow rate through the air handler. There are other ways of modulating equipment capacity, e.g. changing the flow rate and the capacity together so as to maintain a fixed temperature rise across the equipment.

This paper is organized as follows. First we outline the equations for distribution system efficiency. These are then simplified for the case where both supply and return ducts are located outdoors and the air leakages on the supply and return side are equal. The full-load efficiency is then derived for this simple case. Next we indicate how to modify the equations to account for the effect of internal gains and unbalanced duct air leakage. The last major section uses some of the output from a detailed bin model calculation for the efficiency of a heat pump in the Boise, Idaho climate to illustrate the modulating efficiency.

It should be noted that the full/modulating efficiency also applies to the more complex case of supply and return duct buffer zones with their own conductance to ambient and infiltration rates. We have derived an analytic solution for the complex case also, but the resulting equations are too long and complex to include in this paper. The major characteristics of the full-load efficiency in the complex cases are the same as for the simple case.

Distribution System Efficiency

The steady-state distribution efficiency model applies equally for both heating and sensible cooling. For these equations, the sign convention for the cooling mode is that the loads, the equipment capacity, and the heat delivered are negative. For simplicity, the discussion in this paper will use the terminology appropriate to the heating mode.

The distribution efficiency is the ratio of the total useful heat delivery rate to the house to the equipment capacity. The useful heat is delivered via three mechanisms: heat delivered through the supply registers, heat delivered due to regain, and the heat loss or gain due to imbalances in the natural infiltration rate of the heated space. The distribution efficiency can be expressed as

$$\eta_{distrib} = \eta_{del} + \eta_{regain} + \eta_{inf} \tag{1}$$

where $\eta_{distrib}$ = overall distribution efficiency
η_{del} = delivery efficiency (heat delivered through registers)
η_{regain} = change in efficiency due to regain from buffer zones
η_{inf} = change in efficiency due to change in house infiltration rate

The steady-state delivery efficiency for a residential distribution system is defined as the rate of heat delivery from the supply registers to the room air divided by the capacity of the heating system. The delivery efficiency can be less than zero. This occurs when lack of capacity or excessive duct losses result in the delivered supply air being below room temperature. The delivery efficiency can also be greater than one. This occurs when some external factor maintains one or both buffer zone temperatures above the room temperature.

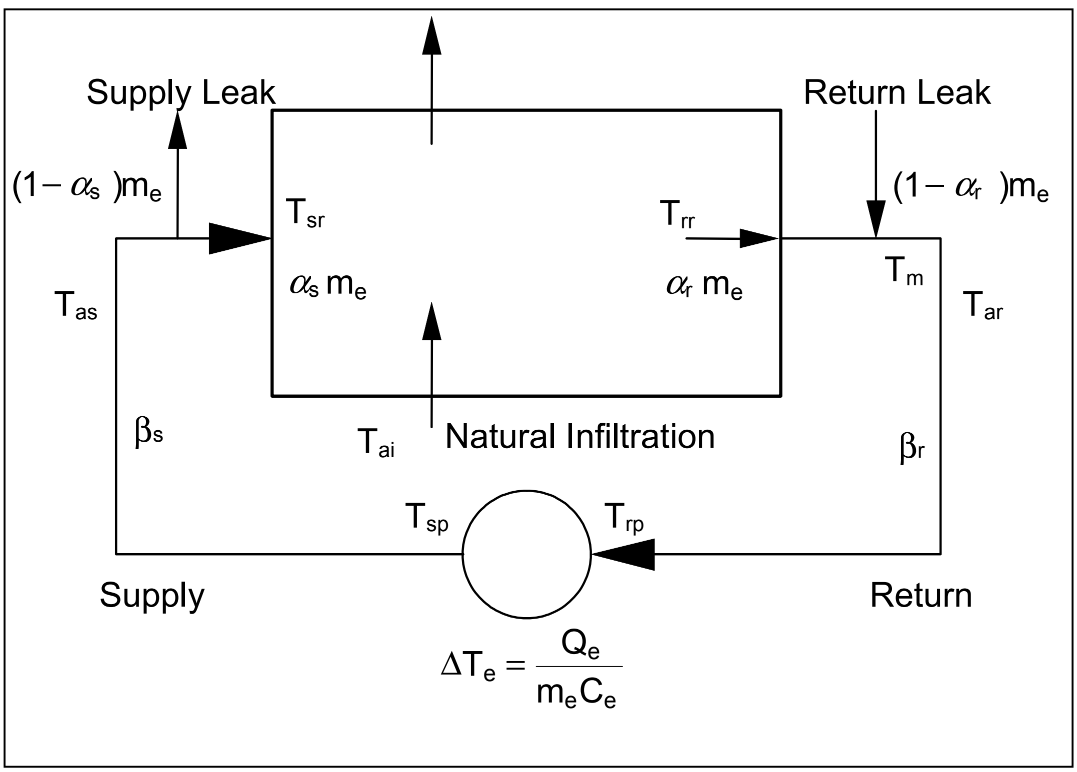
The regain term accounts for the fact that duct air leakage and conduction losses change the temperatures in the unconditioned spaces (buffer zones) in which the ducts are located, which in turn changes the rate of heat loss from the conditioned space. The regain term can be either positive or negative. It is zero when the ducts are located outdoors.

The infiltration interaction term occurs when the air leakage on the supply side differs from that on the return side. When there is more supply leakage than return leakage, there is more flow leaving the conditioned space through the return registers than enters through the supply registers. This unbalanced flow from the home depressurizes the home relative to ambient, thus causing an increase in the infiltration rate of outside air. When the return leakage is larger than the supply leakage, the home is pressurized relative to ambient and the infiltration rate of ambient air is reduced. Thus the infiltration interaction term can either increase or decrease the apparent heating load and distribution system efficiency.

Delivery Efficiency

The basic delivery efficiency model was developed by one of the authors (LP) in 1993 and first published in Palmiter and Francisco (1997). It was subsequently corrected and extended in Francisco and Palmiter (1999). The latter reference provides the most complete set of equations and derivations for residential distribution efficiency. The delivery efficiency equations were incorporated into Standard 152 (ASHRAE 2004).

Figure 1. Schematic Diagram of Duct Leakage Parameters



The basic terms involved in the definition of the delivery efficiency equation are illustrated schematically in Fig. 1. It is assumed that the air leakage from or to the ducts occurs at the register end of the duct system. There is a simple modification of the equations that allows the leakage to occur at the plenums or any intermediate point; see the references cited in the last paragraph. In addition, it is assumed that there is no infiltration airflow between the house and the buffer zones (this assumption is not very realistic but significantly simplifies the analysis). It is also assumed that the air entering the return registers is at room temperature. At the bottom of Fig. 1 is the heat balance for the equipment.

With reference to Fig. 1, the derivation of the delivery efficiency model can be outlined as follows. Room air is mixed with a fraction (1-α_r) of air from the return buffer zone. The mixed air flows through the return ducts and loses heat by conduction to the return buffer zone. The equipment adds heat to the air stream providing a temperature difference ΔT_e between the return plenum and the supply plenum. Heat is lost by conduction from the supply ducts to the supply buffer zone. A fraction (1-α_s) of the supply air is lost to the supply buffer zone just before the registers. Finally the remaining supply air is delivered through the registers. Figure 1 also shows the house infiltration flows. The equations presented below are valid in any consistent set of units.

Both the model developed by Palmiter and Francisco (1997) (with the modifications found in Francisco and Palmiter (1999)) and the one in Standard 152 use the same equation for the steady-state delivery efficiency under heating conditions

$$\eta_{del} = \alpha_s\beta_s - \alpha_s\beta_s(1-\alpha_r\beta_r)\frac{\Delta T_r}{\Delta T_e} - \alpha_s(1-\beta_s)\frac{\Delta T_s}{\Delta T_e} \tag{2}$$

where α_s = leakage efficiency for supply ducts
α_r = leakage efficiency for return ducts
β_s = conduction efficiency for supply
β_r = conduction efficiency for return
ΔT_s = temperature difference between indoors and the ambient for the supply
ΔT_r = temperature difference between indoors and the ambient for the return
ΔT_e = temperature rise across heat exchanger

Leakage efficiency can be defined as the fraction of the air handler flow that is delivered to or removed from the conditioned space by the ducts. It is equal to one minus the air leakage fraction.

From standard heat exchanger theory, the conduction efficiency can be expressed as

$$\beta_x = \exp\left(\frac{-A_x}{m_e c_p R_x}\right) \tag{3}$$

where A = surface area of ducts
m_e = mass flow of air through air handler fan at operating conditions
c_p = specific heat of air
R = duct unit thermal resistance
x = s for supply ducts or r for return ducts

The conduction efficiency is the fraction of the temperature difference between the inlet of the temperature of the duct (mixed temperature for the return ducts and supply plenum temperature for the supply ducts) and the supply/return buffer zone temperature that remains at the outlet end of the ducts. It equals one minus the heat exchanger effectiveness of the duct.

The heat delivery rate through the registers can then be written as

$$Q_{del} = \eta_{del} Q_e = (\alpha_s \beta_s \Delta T_e - \alpha_s \beta_s (1-\alpha_r \beta_r)\Delta T_r - \alpha_s (1-\beta_s)\Delta T_s) m_e c_p \tag{4}$$

where $Q_e = m_e c_p \Delta Te =$ equipment capacity

Delivery Efficiency for Balanced Leakage with Ducts Outdoors

The equations in this paper are restricted to the case where the ducts are outside, so the regain term in the equations is zero. For the same reason, both the supply and return buffer zone temperatures in Eq. (4) are set equal to the ambient temperature. If we also assume the supply and return air leakages are equal, the infiltration interaction term is also zero. With these assumptions the distribution efficiency equals the delivery efficiency and the efficiency equation and the heat delivered equation then simplify to

$$\eta_{del} = \alpha_s \beta_s - \alpha_s (1-\alpha_r \beta_r \beta_s)\frac{\Delta T_o}{\Delta T_e} \tag{5}$$

$$Q_{del} = \eta_{del} Q_e = \alpha_s \beta_s Q_e - \alpha_s (1-\alpha_r \beta_r \beta_s)\Delta T_o m_e c_p \tag{6}$$

where $\Delta T_o =$ inside temperature minus ambient temperature

The equation for Q_{del} has a simple interpretation. The first term on the right side is the fraction of the heat injected into the air stream by the equipment that is delivered to the house. If we run the system with the heat off, neglecting the fan heat, the first term is zero.

The second term on the right represents the heat exchanger loss from the duct system. This is essentially another conductance between the house and outside in which the ducts act as a heat exchanger when the fan is running. This term is negative when ΔT_o is greater than zero. In fan-only mode, this means the supply air delivered to the home is colder than the return air leaving the home. The delivery efficiency in this case will also be negative.

Full-load Efficiency for Balanced Leakage

With the above equation for steady-state heat delivery, one can calculate the heat balance for the condition where the house load equals the useful heat delivery:

$$Q_{load} = UA\Delta T_o = \alpha_s \beta_s Q_e - \alpha_s (1-\alpha_r \beta_r \beta_s)\Delta T_o m_e c_p \tag{7}$$

where $UA =$ the steady-state heat loss coefficient for the home

The delivery efficiency in Eq. (5) can be rewritten as,

$$\eta_{del} = \frac{\alpha_s \beta_s Q_e - \alpha_s (1 - \alpha_r \beta_r \beta_s) \Delta T_o m_e c_p}{Q_e} \tag{8}$$

Solving Eq. (7) for ($\Delta T_o / Q_e$) and substituting into Eq. (8) leads to the equation for steady-state delivery efficiency under full-load conditions:

$$\eta_{full} = \frac{\alpha_s \beta_s}{1 + \frac{m_e c_p}{UA} \alpha_s (1 - \alpha_r \beta_r \beta_s)} \tag{9}$$

The expression for full-load efficiency shows that it is not a function of either the indoor or outdoor temperatures. It depends only on the ratio of capacity flow rate through the equipment to the heat loss coefficient of the home and the duct parameters. The second term in the denominator is equal to the heat exchanger conductance of the duct system divided by the conductance of the house to ambient. When all of the duct efficiencies are equal to one (i.e., perfect ducts), the full-load efficiency is also equal to one. Note that the β's are functions of the ratio of the supply or return duct heat loss coefficients to the heat capacity flow rate of the equipment. For a fixed flow rate, fixed parameters, and fixed house heat loss coefficient, the full-load efficiency is independent of climate.

Effect of Gains

When there are solar and internal gains to the conditioned space, the full-load efficiency must be modified. The heat balance on the conditioned space becomes

$$Q_{load} = UA \Delta T_o - Q_{gains} = \alpha_s \beta_s Q_e - \alpha_s (1 - \alpha_r \beta_r \beta_s) \Delta T_o m_e c_p \tag{10}$$

where Q_{gains} = $UA \Delta T_{gains}$ = the rate of combined solar and internal gains
ΔT_{gains} = the temperature increase of the house due to gains

This leads to a new equation for the full-load efficiency

$$\eta_{full} = \frac{\alpha_s \beta_s}{1 + \frac{m_e c_p}{UA} \alpha_s (1 - \alpha_r \beta_r \beta_s) F_{gains}} \tag{11}$$

where $F_{gains} = \dfrac{1}{1 - \dfrac{\Delta T_{gains}}{\Delta T_o}}$ = gains factor

In contrast to the situation without gains, the full-load efficiency now depends on the temperature difference between indoors and outdoors and also on the temperature rise in the homes due to gains.

As the outdoor temperature approaches the house balance point, ΔT_o approaches ΔT_{gains} and the gains factor approaches infinity causing the full-load efficiency to approach zero.

Effect of Interaction of Unbalanced Leakage with Infiltration

The change in load due to the interaction of unbalanced duct leakage with the house natural infiltration can be written as

$$Q_{inf} = c_p (m_{in} - m_{nat})(T_{out} - T_{in}) \tag{12}$$

where m_{in} = mass infiltration rate of the house when equipment is on
m_{nat} = natural mass infiltration rate of the house, i.e. when equipment is off

Using the fan model developed by Palmiter and Bond (see Francisco and Palmiter 1999)

$$m_{in} = \begin{cases} m_{nat} + \frac{1}{2}(\alpha_r - \alpha_s) m_e & \text{if} \quad \frac{1}{2}|\alpha_r - \alpha_s| m_e < m_{nat} \\ 0 & \text{if} \quad \frac{1}{2}|\alpha_r - \alpha_s| m_e > m_{nat} \quad \text{and} \quad \alpha_s > \alpha_r \\ (\alpha_r - \alpha_s) m_e & \text{if} \quad \frac{1}{2}|\alpha_r - \alpha_s| m_e > m_{nat} \quad \text{and} \quad \alpha_r > \alpha_s \end{cases} \tag{13}$$

The first equation above corresponds to the most commonly encountered case in which the unbalanced duct leakage is less than twice the natural infiltration rate. In the second case, the home is fully pressurized (outflow through all cracks) and the infiltration rate is zero. The third case occurs when the unbalanced duct leakage is greater than twice the natural infiltration rate of the home. The first two cases are easily modeled because it is not necessary to know the natural infiltration rate.

The Q_{inf} term needs to be added to Q_{del} in the house heat balance equation in order to produce the full-load efficiency for the case with unbalanced leakage.

Detailed Bin Model Calculation

The effect of duct efficiency on heat pump performance is illustrated in Fig. 2 by the results of a detailed prediction using the modified bin method. The temperature bin data used was based on 30 years of data for Boise, Idaho. Unlike the simplified equations discussed above, the bin model accounts for all of the effects involved with ducts located in the attic and crawl space. These include the regain (if any), the impact of duct leakage on the ventilation rates of the attic and crawl space, internal and solar gains, a ground node temperature connection in the crawl space, and the conductance of the attic and crawl space to both the house and ambient.

The full set of duct equations presented in (Francisco and Palmiter 1999) were solved analytically for the buffer zone temperatures which were then used to calculate the useful heat delivered and other quantities of interest.

The heat pump was modeled using detailed data from the manufacturer's product literature. The model calculates both heating and sensible cooling. The restriction to sensible cooling is appropriate for Boise because of the dry climate. The model also accounts for the part load factor (PLF) using a coefficient of degradation of 0.06. This value was estimated from the cooling performance specifications; we assumed the same value would apply in heating mode.

Figure 2. Heat Pump Bin Model: Boise, ID

Average Annual Temperature, Boise (F)	51.1	House UA (Btu/h-F)	377
Floor Area (ft^2)	1350	Internal and Solar Gains (Btu/h)	5127
Boise Heating Degree Days, Base 65 (F-day)	5861	Boise Cooling Degree Days, Base 65 (F-day)	754
Heating Set Point (F)	65	Cooling Set Point (F)	75
Heating Balance Point (F)	51.8	Cooling Balance Point (F)	62.0
High Efficiency Heat Pump Size (tons)	3.5	Calculated Heat Pump C_d	0.06
Heat Pump HSPF	9.0	Heat Pump SEER	14.5
Supply Leakage, Percent Air Handler Flow	10%	Return Leakage, Percent Air Handler Flow	10%
Supply Duct Insulation Level	R-4	Return Duct Insulation Level	R-4
Supply Duct Location	Crawl	Return Duct Location	Attic

To (F)	Hrs	DDbal (F-day)	Cap (Btuh)	KWin (KW)	COP	Qbase (Btuh)	Qdis (Btuh)	Fon	PLF	ΔTe (F)	Qneed (Btuh)	Qhpin (Btuh)	Qauxin (Btuh)	Duct Eff	Sys Eff
107	1.0	-2	-35723	-3.83	2.74	-16951	-26121	.67	.98	-26.4	-23274	-8691	0	.73	1.95
102	9.1	-15	-36505	-3.63	2.95	-15066	-27577	.56	.97	-26.8	-20024	-6982	0	.75	2.16
97	55.9	-81	-37263	-3.44	3.17	-13181	-29014	.47	.97	-27.2	-16992	-5537	0	.78	2.38
92	148.6	-186	-38022	-3.25	3.42	-11296	-30451	.39	.96	-27.7	-14149	-4296	0	.80	2.63
87	235.4	-245	-38780	-3.09	3.68	-9411	-31887	.31	.96	-28.1	-11472	-3259	0	.82	2.89
82	308.9	-257	-39433	-2.93	3.95	-7526	-33233	.24	.95	-28.4	-8942	-2377	0	.84	3.17
77	382.5	-238	-40017	-2.77	4.24	-5641	-34518	.17	.95	-28.7	-6541	-1624	0	.86	3.47
72	487.6	-202	-40600	-2.60	4.57	-3756	-35803	.11	.95	-29.0	-4255	-983	0	.88	3.82
67	559.8	-116	-41183	-2.44	4.95	-1871	-37088	.05	.94	-29.4	-2073	-444	0	.90	4.21
62	653.0	0	51567	3.56	4.25	0	43201	.00	.94	36.6	0	0	0	.84	3.34
57	713.9	0	47950	3.48	4.04	0	40070	.00	.94	34.1	0	0	0	.83	3.17
52	790.8	0	44975	3.40	3.88	0	37495	.00	.94	32.0	0	0	0	.83	3.03
47	934.0	186	42000	3.31	3.71	1799	33438	.06	.94	29.9	2271	648	0	.79	2.77
42	873.3	356	37158	3.24	3.36	3684	28467	.14	.95	26.6	4843	1522	0	.76	2.42
37	837.3	515	32317	3.16	2.99	5569	23496	.25	.95	23.3	7730	2705	0	.72	2.06
32	768.6	633	29633	3.10	2.80	7454	20393	.38	.96	21.6	10942	4065	0	.68	1.83
27	441.5	456	26950	3.06	2.58	9339	17290	.56	.97	19.8	14701	5850	0	.64	1.60
22	262.3	325	25083	3.01	2.44	11224	14894	.77	.99	18.7	19030	7909	0	.59	1.42
17	139.1	202	23217	2.96	2.30	13109	12498	1.00	1.00	18.1	23922	10114	706	.55	1.21
12	69.3	115	21292	2.89	2.16	14994	10052	1.00	1.00	20.4	27001	9875	5709	.56	.96
7	40.0	75	19367	2.82	2.01	16879	7605	1.00	1.00	22.7	30080	9636	10713	.56	.83
2	28.2	58	17617	2.77	1.87	18764	5310	1.00	1.00	25.1	33159	9437	15542	.57	.75
-3	13.9	32	15867	2.71	1.72	20649	3015	1.00	1.00	27.4	36237	9238	20371	.57	.70
-8	6.7	17	14117	2.65	1.56	22534	720	1.00	1.00	29.7	39316	9039	25199	.57	.66
-13	3.3	9	12367	2.59	1.40	24419	-1575	1.00	1.00	32.0	42395	8840	30028	.58	.63
-18	1.8	5	10617	2.53	1.23	26304	-3870	1.00	1.00	34.4	45474	8641	34857	.58	.60
-23	.2	1	8867	2.47	1.05	28189	-6165	1.00	1.00	36.7	48552	8441	39686	.58	.59

Annual Summary

	DDbal	KWHbase	KWHneed	KWHhpi	KWHauxi	KWHtot	DuctEff	SysEff
Cooling	1342	3559	4236	1134	0	1134	.84	3.14
Heating	2983	7908	12027	4382	580	4962	.66	1.59
Total	4325	11467	16263	5517	580	6097	.71	1.88

Although the calculations for this example are more detailed and are stated in terms of the overall distribution efficiency, the qualitative behavior observed when the compressor runs continuously and the strip heat auxiliary cycles just enough to meet the load is the same as that previously discussed for the case with internal gains.

The first two rows of output in Fig. 2 contain column headings and corresponding units for the rows of bin results below. The first two columns show the number of hours, *Hrs*, at each outside temperature bin, T_o, for this location. Column three gives the annual heating or cooling degree days in at each bin temperature, DD_{bal}, using the heating or cooling balance point as the base. The values for cooling degree days, loads, and capacities are given as negative numbers. Columns four and five are the heat pump capacity, *Cap*, and input energy, KW_{in}, taken from manufacturer's data, these include all the effects of defrost and fan energy. The heat pump input energy also includes the indoor and outdoor fan energy. Column six, the coefficient of performance (COP) is calculated as,

$$COP = Cap / 3413 KW_{in} \tag{14}$$

The house base load, Q_{base}, in column seven, is for the house with perfect ducts. Column eight is the total heat delivered, Q_{dis}, to the conditioned space by the compressor, including the effects of regain and the infiltration interactions to the house and both buffer zones. This does not include the auxiliary heat, it is for the compressor only. Note that Q_{dis} becomes negative in the heating season when running the compressor alone would cool the house. The fractional on-time, F_{on}, in column nine, is the fraction of the hour that the heat pump needs to run given the COP, and the duct losses. The heat pump part load factor *PLF*, found in column ten, is calculated as,

$$PLF = 1 - C_d (1 - F_{on}) \tag{15}$$

where C_d = the coefficient of degradation

Column eleven is the temperature rise across the equipment, ΔT_e. The house load including duct losses, Q_{need}, is found in column twelve, while the next two columns are the energy input for the compressor, Q_{hp}, and the auxiliary heat, Q_{aux}, required to meet this load.

The final two columns of Fig. 2 represent the duct efficiency, *DuctEff*, and the system efficiency, *SysEff*, respectively. The duct efficiency is the ratio of the house load with perfect ducts to that including duct losses, or

$$DuctEff = Q_{base} / Q_{need} \tag{16}$$

The system efficiency is the efficiency of the heating and cooling devices including the effect of the ducts. This is calculated as the ratio of the energy required to condition the house with perfect ducts to the energy required with duct losses, or

$$SysEff = Q_{base} / (Q_{hp} + Q_{aux}) \tag{17}$$

Note that in the bin model output in Fig. 2 the heating and cooling numbers are distinguished by the negative sign on the cooling numbers.

Following the bin data are seasonal and annual summaries of a subset of variables. The first of these is the degree days, which are simply summed over the cooling season, heating season, and the year. In the next column, the annual base load for the house with perfect ducts, KWH_{base}, is calculated as

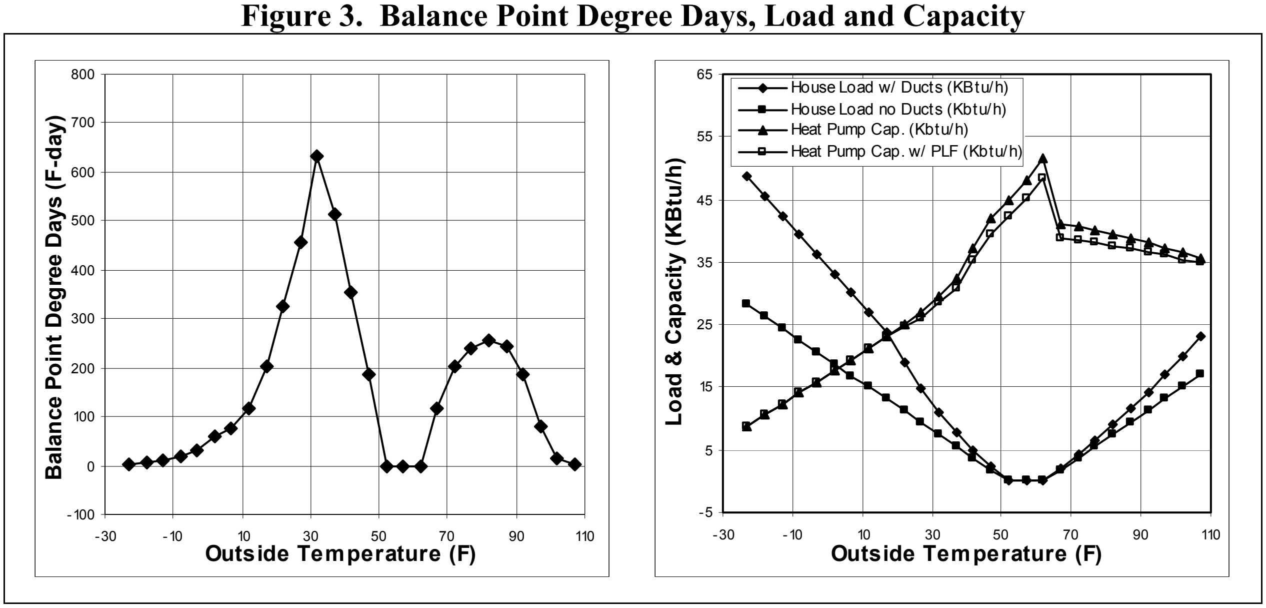
$$KWH_{base} = (\sum_{bins} Q_{base} Hrs) / 3413 \qquad (18)$$

where the summation is performed across all bins for the total annual summary, or across the heating or cooling bins for the seasonal summaries. A similar methodology is used to summarize the load of the house including duct losses, KWH_{need}, the input energy required by the compressor with duct losses, KWH_{hp}, and the auxiliary energy required with duct losses, KWH_{aux}, in the following columns. These last two columns are summed across each row to produce the total seasonal and annual energy input required, KWH_{tot}, for the house with duct losses. The summaries of duct and system efficiencies are found in the last two columns. These are also calculated using previous summaries:

$$DuctEff = KWH_{base} / KWH_{need} \qquad (19)$$

$$SysEff = KWH_{base} / KWH_{total} \qquad (20)$$

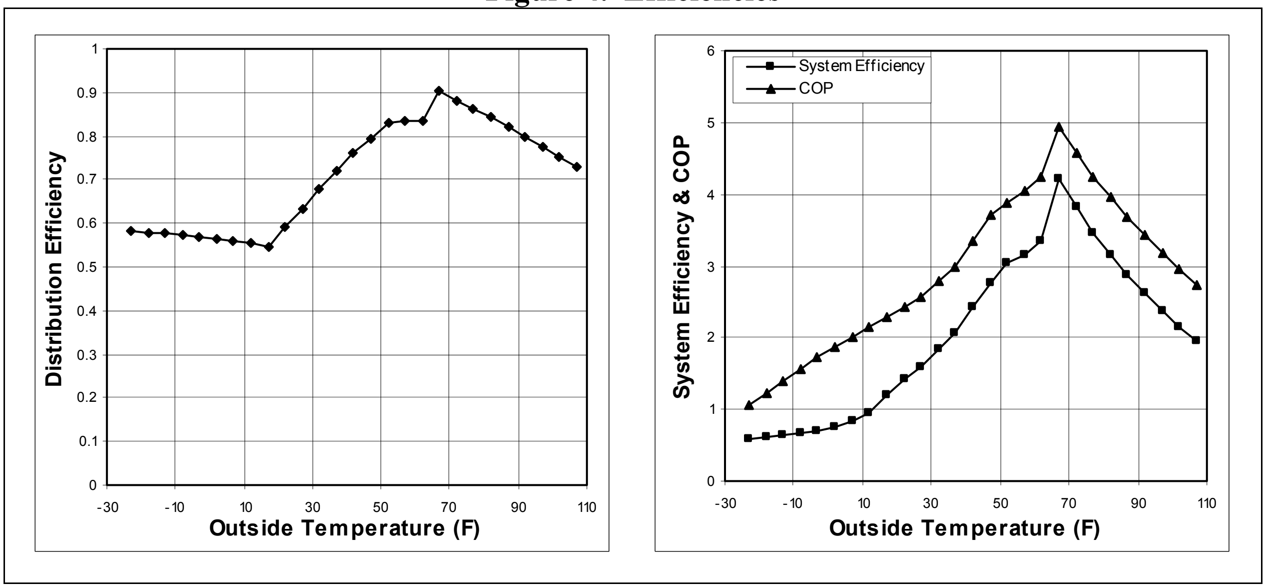
Figure 3. Balance Point Degree Days, Load and Capacity



The left panel of Fig. 3 shows the annual heating and cooling degree days associated with each outdoor bin temperature. Most of the heating load occurs between 17 F and 47 F with a peak at 32 F.

The right panel of Fig. 3 shows the capacity of the compressor with and without the part load factor and the heating load with and without duct losses. The balance point temperature of the heat pump is the outdoor temperature at which the compressor is just able to meet the heating load (including duct losses and defrost penalty). This occurs just above 17 F for the case with duct losses. Without duct losses the balance point would have been at about 5 F. This illustrates how duct losses change the heat pump balance point. Below the heat pump balance point the vertical distance between the compressor capacity and the load is the amount of strip heat required to meet the load. The graph illustrates the large increase in strip heat due to the duct losses.

Figure 4. Efficiencies



The distribution efficiencies are shown in the left panel of Fig. 4 for both heating and cooling. At outdoor temperatures above 17 F, the compressor is able to meet the load without auxiliary strip heat. The efficiency in heating mode increases from 0.55 at 17 F to 0.79 for the 47 F bin. For the bins at 17 F and below, the efficiency shows the characteristic pattern of modulating efficiency with gains. It slowly increases as the outdoor temperature decreases. Without gains the efficiency would remain constant over the range of outdoor temperatures where the strip heat modulates to meet the load.

The distribution efficiency in cooling mode is much higher than in the heating mode. On average over the year the cooling duct efficiency is 0.84 versus 0.66 for the heating mode. Note the efficiency is especially high in the bins centered at 67 F and 72 F. This is due to the fact that the solar and internal gains generate a need for cooling when the outdoor temperature is lower than that of the indoor air at 75 F. Even though the buffer zone temperatures are somewhat warmer than the outdoor temperature, they are still cool enough that the duct "losses" are actually a benefit. Under these circumstances fan-only operation would still provide some useful cooling.

The right panel of Fig. 4 shows the heat pump COP and the overall system efficiency. The system efficiency includes the effects of the equipment COP, the part load factor and the duct efficiency. The difference between the two curves is primarily due to duct losses.

Conclusions

The full-load/modulating efficiency of the distribution system has several interesting properties. These properties remain valid whether using a simplified version of the model restricted to the ducts located outdoors (as in most of this paper) or a more complex model with the conductance of the buffer spaces to ambient and the buffer space infiltration rates taken into account.

Firstly, in the absence of internal and solar gains, the full-load/modulating efficiency is independent of the indoor and outdoor temperatures for a fixed flow rate through the air handler, fixed duct properties, and fixed infiltration rates.

Secondly, when gains are present the efficiency at large ΔT_o asymptotically approaches the efficiency in the no-gains case and as one approaches the heating balance point of the home, the efficiency approaches zero.

The first property suggests that there may be some merit in the idea of using the full-load efficiency as an index of duct system performance for energy rating systems. It is also useful for sizing equipment to meet the combined load of ducts and house at design conditions.

References

ASHRAE. 2004. *Standard 152, Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems*. American Society of Heating, Refrigeration, and Air-Conditioning Engineers.

Francisco, P. and L. Palmiter. 1999. *Improvements to ASHRAE Standard 152P*. Ecotope, Inc., Seattle, WA.

Palmiter, Larry and P. Francisco. 1997. *Development of a Practical Method for Estimating the Thermal Efficiency of Residential Forced-Air Distribution Systems.* EPRI Report Number TR-107744. Electric Power Research Institute, Palo Alto, CA.