Leakage in Large-Building Duct Systems: Modelling the Savings for Various Applications

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ABSTRACT

Decisions about whether it is worthwhile to seal duct leakage in large buildings are based upon different needs in different applications, ranging from the need to meet diffuser/exhaust-grille flow requirements for ventilation regulations, to meeting fire-safety specifications, to maintaining zone pressurization/depressurization requirements in hospitals. However, many decisions about whether to seal duct leaks are based upon the energy and peak-electricity-demand implications of sealing that leakage. This paper discusses the varying energy and peak-demand savings mechanisms for different types of duct systems, starting with simple exhaust ventilation systems, and including Constant Air Volume systems as well as Variable Air Volume systems with different means for controlling Outdoor Air. The magnitudes of different savings mechanisms (fan power, Outdoor Air conditioning, terminal reheat, peak power reduction) will be compared for different system types, and the functional dependence on leakage level will be presented for each energy implication for each type of HVAC system.

KEYWORDS

Ducts, Leakage, Commercial Buildings, Energy Modelling, Energy Saving

1 INTRODUCTION

Duct leakage has often been an unseen culprit relative to the performance of air distribution systems in buildings. Unlike water leakage, which makes its presence undeniably obvious, duct leakage is often not obvious. Decisions about whether there is enough duct leakage to justify sealing that leakage in an existing large building depend upon understanding the magnitude of that leakage, as well as the implications of that leakage in different applications. These implications range from the need to meet diffuser/exhaust-grille flow requirements for ventilation regulations, to meeting fire-safety specifications, to maintaining zone pressurization/depressurization requirements in hospitals. However, many decisions about whether to seal duct leaks are based upon the energy and peak-electricity-demand implications of sealing that leakage.

The prevalence of duct leakage in non-residential buildings, as well as some of the energy savings mechanisms associated with reducing that duct leakage have been presented in various papers over the years. This includes work on European duct systems by Carrie et al. (Carrie, 2000), as well as work on US duct systems by Modera et al. (Modera, 2014). Other work includes modelling of energy savings from duct sealing for Variable Air Volume (VAV) systems by Franconi et al. (Franconi, 1998), as well as by Wray and Matson (Wray, 2003), and even an analysis of the specific impacts of duct sealing on Outdoor Air (OA) conditioning by Krishnamoorthy and Modera (Krishnamoorthy, 2016).

This paper builds upon earlier work, focusing on the implications of system type and building controls on the value of duct sealing, including the impacts on duct-sealing energy savings associated with different fractional outdoor-air and relief-air rates. The implications of sealing duct leakage are explored numerically using a hypothetical comparison of the impacts of sealing 20% leakage in three different types of exhaust systems, and two types of Constant Air Volume (CAV) supply systems. The implications of Outdoor Air (OA) and building pressure control for Variable Air Volume (VAV) systems are also explored.

2 MECHANISMS BEHIND ENERGY SAVINGS

There are several mechanisms by which duct leakage impacts energy use, however all these mechanisms result in changes in either fan power or thermal-conditioning energy use. The basic mechanism for duct leakage increasing fan power is that the fan must move more air to satisfy building needs when the ducts leak. This generally translates to either: a) providing specified flows at grilles or diffusers, or b) meeting the thermal conditioning needs of building zones, or both. The major input variables for determining the fan power impacts of duct sealing are the operating schedule of the fan system, and the relationship between flow and power. Concerning the latter, the fan power for exhaust systems typically scales with the flow cubed, whereas for supply systems it is generally assumed to scale with the flow raised to the power 2.4. The cube law stems from the fact that turbulent-flow duct pressure loss scales with the flow rate squared, however the 2.4 power is an empirical observation that likely stems from some pressure losses scaling with the flow raised to a power less than 2. Thus, the 2.4 power likely varies between buildings and systems, however that variation will not be addressed in this paper.

Thermal conditioning implications of duct leakage are far more complex, stemming from different mechanisms for different systems. For both fan power and thermal energy implications of leakage, the implicit assumptions behind the analyses in this paper are that the zone air flow and thermal conditioning needs of the building will be met to the same extent before and after sealing duct leakage. That said, other building needs might be met to differing extents before and after sealing. For example, depending on the control mechanisms for ventilation and building pressurization, those operating parameters may change after sealing. Specifically, if building pressurization and Outdoor Air flow rates are not actively controlled, both are likely to be reduced after sealing.

2.1 Thermal Conditioning Impacts

There are four categories of duct leakage impacts on thermal conditioning: 1) cooling required to remove the heat generated by excess fan power, 2) cooling or heating required to condition any changes in outdoor air flow associated with duct leakage, 3) changes in the amount of terminal reheat required due to duct leakage, and 4) changes in the enthalpy of air being exhausted from the building due to duct leakage. The first two categories have been analysed in several of the papers referenced above, however the latter two categories have received much less attention.

Concerning the cooling required to remove fan heat, the basic idea is that all fan power ultimately turns into heat, which then must be removed by the cooling system. However, for exhaust systems this is not the case, and for supply systems the analysis can be complicated by the changes in Sensible Heat Ratio seen by the cooling coil associated with fan heat (which is all sensible). In addition, as will be seen below, some of that fan heat is exhausted from the building in relief air flows, in particular fan heat associated with fans. All these complications for supply systems will not be addressed in this paper, instead we will use the following simple approximations: 1) all supply fan heat shows up as a load, 2) half of the fan power is in the return fan, and 3) the fractional relief air times the return fan heat represents the fan heat that does not show up as a load.

The second mechanism, increased outdoor air conditioning due to duct leakage, was discussed in detail by Krishnamoorthy and Modera (Krishnamoorthy, 2016), however that paper assumed that the Outdoor Air (OA) flow into the building was a fixed fraction of the supply air flow, which implies that the increased fan flow associated with duct leakage results in increased outdoor air flow. Another implicit assumption in that paper is that the OA flow is all exhausted through the building envelope at the zone-air enthalpy. Examining building operations more carefully, this latter assumption is not appropriate in many buildings, namely any buildings that have relief air flows. The first assumption is also not valid for buildings that control the absolute OA flow and/or the pressure differential across the building shell through some combination of damper controls and supply/return/relief fan controls. The problems with these assumptions are most obvious for hospitals, which generally operate at high OA fractions (50-100% of supply flow), and which therefore must have relief flows to avoid over-pressurization of the building envelope. Moreover, because the outdoor air flows are so large for hospitals, they are very likely to have active building pressurization control. The impact of assuming that all exhausted air is at zone-air conditions can be quite dramatic for hospitals, which exhaust a large fraction of the building air rather than recirculating that air. What this means is that a significant fraction of supply duct leakage into the return plenum (i.e. ceiling-plenum return) is exhausted from the building through the relief damper, thereby changing the enthalpy of the air leaving the building. This is the third category of thermal conditioning impacts due to duct leakage listed above. It should be noted that hospitals are also more likely to have ducted return systems, but that such installations are not analysed in this paper, as such an analysis depends upon the ratio of supply to return leakage levels.

Finally, the fourth category of thermal conditioning impact, changes in reheat due to duct leakage, has not seen much attention. Moreover, it is important to note that this impact only arises for leakage downstream of terminal boxes (i.e. downstream of the terminal reheat coils), and the impact has opposite signs for CAV and VAV systems. For this analysis, the implicit assumptions are that zone loads are the same with and without duct leakage, and therefore. assuming that the supply air temperature is unchanged by duct leakage, that the supply air flows into those zones are the same with and without duct leakage. This assumption could be questioned, as higher velocities in trunk sections when there is duct leakage could result in lower thermal losses to the return plenum.

In the case of CAV systems, the assumption of constant zone air flow means that the fan flow needs to be adjusted manually after sealing to maintain the same zone air flow. For VAV systems it is assumed that the zone thermostats automatically adjust the zone flows, and that the supply fan automatically adjusts its flow based upon the flow through the VAV boxes (i.e. it is controlled to maintain a specific pressure at some point in the supply trunk, which automatically responds to the openings of the VAV dampers, and to any leakage from the supply trunk).

Returning to the location of the leakage, it should be clear that leakage upstream of the terminal reheat coils will not impact the amount of reheat, based upon the assumption that the air temperature arriving at the terminal boxes is not impacted by duct leakage. On the other hand, leakage downstream of the reheat coil does impact reheat. In the case of CAV systems this impact can be quite dramatic, as the amount of reheat scales with the air flow passing through the terminal box, which is higher when there is duct leakage downstream of the terminal box. This stems from the assumption that the zone air flow remains unchanged after sealing, and that the thermostat is therefore producing the same temperature of that air after sealing. Thus, the extra reheated air is leaked into the return plenum, and the extra heat in that return air must be removed by the cooling coil to produce the desired supply air temperature, resulting in a two-fold impact. For a VAV system, when reheat is being used to actually heat zones (versus to avoid over-cooling zones), the effect of downstream duct leakage is the same as for CAV systems. On the other hand, when reheat is being used to increase the temperature of the minimum ventilation air flow to the zone to avoid over-cooling a zone, sealing duct leakage has the opposite effect on reheat. This is because leakage downstream of the VAV box reduces the amount of cold air being delivered to the zone, thereby reducing the over-cooling, and thereby reducing the need for reheat. Due to these complexities, and the complexity of coming up with a simple estimate of the amount reheat required in a given building in a given climate, the reheat impacts of duct leakage are only addressed in a cursory manner in the discussion section of this paper.

3 LEAKAGE IMPACT COMPARISON

To illustrate the variability in savings associated with sealing duct leakage in different applications, a comparison of the impacts of sealing the same amount of duct leakage in two exhaust-system applications and two CAV supply-system applications. The analyses are performed assuming that 100% of the leakage is sealed. Although that is not generally achievable, the math becomes much simpler, and the savings magnitudes will roughly scale when the more typical 80-90% sealing is achieved. The basic assumptions for systems being sealed are presented in Table 1, which shows that all applications are assumed to have the same zone flows, duct leakage, and total pressure differential seen by the fan. In addition, all applications are assumed to be in New York City for climate purposes, and the zone flow is assumed to be the same before and after sealing, implying that the fan speed is adjusted to make that the case. Also, for all cases initial duct leakage is assumed to be 20%, initial total fan pressure is assumed to be 500 Pa, fan efficiency is assumed to be constant at 50%, and fan motor efficiency is assumed to be constant at 80%. Finally, for all thermal conditioning calculations, it is assumed that the conditioning is done with a heat pump, with a constant COP of 3 for both heating and cooling.

Table 1: Assumptions for Comparison of Duct Sealing Impacts

Application	Zone Flow [l/s]	Initial OA Fraction	Initial Relief Air Fraction	Operating Hours
CAV Office Supply	23,600	20%	10%	Weekdays 07-18
CAV Hospital Supply	23,600	70%	60%	8760 h/year
Office Exhaust	23,600	N/A	N/A	Weekdays 07-18
Multifamily Exhaust	23,600	N/A	N/A	8760 h/year

3.1 Exhaust System Savings

Starting with the exhaust systems, the fan power savings is calculated as follows:

$$\text{Initial Fan Flow} = \text{Zone Flow} / (1\text{-}\%\text{Leakage}) = 23{,}600/(1\text{-}20\%) = 29{,}500\ [\text{l/s}] \quad (1)$$

$$\text{Initial Fan Power} = 29{,}500\ [\text{l/s}]/1000\ [\text{l/m}^3] \times 500\ [\text{Nt/m}^2]/50\%/80\% = 36.9\ \text{kW} \quad (2)$$

$$\text{Final Total Fan Pressure} = 500*(23{,}600/29{,}500)^2 = 320\ [\text{Pa}] \quad (3)$$

$$\text{Final Fan Power} = 23{,}600\ [\text{l/s}]/1000\ [\text{l/m}^3] \times 320\ [\text{Nt/m}^2]/50\%/80\% = 18.9\ \text{kW} \quad (4)$$

To determine annual savings, the difference between these two power levels needs to be integrated over all operating hours in the year.

For exhaust systems, the only thermal conditioning savings is associated with having to condition less outdoor air after sealing, however the outdoor air conditioning savings is application dependent. For an exhaust-ventilated apartment building, any excess exhaust air shows up as an increased infiltration load, assuming that the leakage is within the conditioned envelope of the building. To calculate the savings over an entire year, the integrated indoor to outdoor enthalpy differential needs to be calculated for the climate in which the system is located using appropriate conditions for the indoors during each season. For this we define:

$$\text{ECDH} = \sum (h_{OA} - h_{zone}) \quad if\ (h_{OA} - h_{zone}) > 0, \text{ else } 0 \quad (5a)$$

$$\text{EHDH} = \sum (h_{\text{zone}} - h_{\text{OA}}) \quad if\ (h_{\text{zone}} - h_{\text{OA}}) > 0, \text{ else } 0 \quad (5b)$$

$$Q_{cool} = 1.2[\text{kg/m}^3] \times \text{Flow}[\text{l/s}]/1000\ [\text{l/m}^3] \times \text{ECDH}\ [\text{kJ h/kg}] \quad (6a)$$

$$Q_{\text{heat}} = 1.2[\text{kg/m}^3] \times \text{Flow}[\text{l/s}]/1000\ [\text{l/m}^3] \times \text{EHDH}\ [\text{kJ h/kg}] \quad (6b)$$

where:

h_{OA} is the enthalpy of the incoming outdoor air [kJ/kg], and

h_{zone} is the enthalpy of the air leaving the zones [kJ/kg]

For an exhaust system in a commercial building, the impact is different, as excess exhaust air shows up as potential changes in flow through the OA intake of the supply air system. Assuming that the pressure across the envelope of the building is being controlled, the amount of outdoor air decreases after sealing, assuming further that there is always more air needed for pressurization than for ventilation. In this case, the cooling sum in Equations 5a and 6a remain the same, however the heating sum in Equations 5b and 6b changes, as the supply system is producing cold air, which means that the penalty for bringing in excess outdoor air is based upon the enthalpy difference between the mixed air (combination of outdoor air and zone air) and the supply air, i.e., substituting ($h_{\text{sup}} - h_{\text{mixed}}$) into Equation 5b. Note that h_{OA} comes into this calculation, as $h_{\text{mixed}} = h_{\text{zone}} \times (1\text{-}\ \%\text{OA}) + h_{\text{OA}} \times \%\text{OA}$.

The savings described above were turned into a breakdown of annual savings in Table 2, using 2200 hours of operation for the office application (200 days x 11 h/day), and ECDH and EHDH values for apartment building and office applications, calculated using TMY3 data for New York City.

Table 2: Duct Sealing Impacts for Sealing 20% Exhaust Leakage for 23,600 l/s, 500 Pa system in NYC

Application	Fan Power [kWh]	ECDH [kJ h/kg]	EHDH [kJ h/kg]	Cooling [kWh]	Heating [kWh]	Total Savings [kWh]
Office Exhaust	39,600	3606	99	8509	234	48,300
Multifamily Exhaust	157,600	6387	22,550	15,073	53,217	225,900

3.2 Supply System Savings

Turning to the CAV supply systems, the fan power savings is calculated using Equations 1-4, except that the Final Total Fan Pressure $=500*(23{,}600/29{,}500)^{1.4} = 366$ [Pa] is used in Equations 3 and 4, yielding a Final Fan Power of 21.6 kW.

For supply systems, the thermal conditioning impact of duct sealing includes fan power, which is calculated as follows:

$$\text{Fan Heat Cooling Electricity Savings} = (36{,}900-21{,}600)[\text{W}]/3[-]*(1-0.5*\%\text{relief}) \quad (7)$$

This translates to 5100W*(1-0.5*10%) = 4845W for the office building, and 5100W*(1-0.5*60%) = 3570W for the hospital, based on parameters in Table 1.

To better understand the analysis of outdoor air conditioning savings, the air flow pathways for the CAV supply system are illustrated in Figure 1. In this Figure the red arrows represent hot outdoor air being pulled into the building, the light green arrows represent air at zone conditions, and the blue arrows represent air at supply conditions. The red dotted line represents the envelope of the building. Note that some of the zone air leaves the building as “Pressurization Air”, some is returned to the supply fan, and some is exhausted as relief air. As for the supply air, some is leaked out into the ceiling-plenum return through supply duct leakage, some of which is returned to the supply fan, and some of which is exhausted as relief air. Another important distinction for the CAV system analysis is that they are assumed to be non-changeover, meaning that the air handler provides cold air all year round, rather than switching to warm air in the winter. This makes sense for any system that is serving both core and perimeter zones, as core zones need cooling year-round.

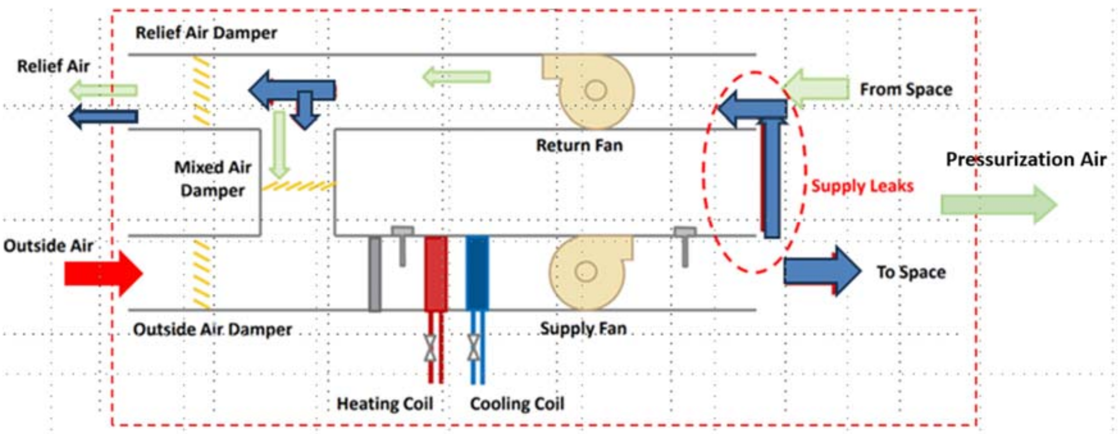

Figure 1: Schematic of Air Flow Pathways for CAV Supply System

Focusing on Figure 1, the savings associated with conditioning OA depends upon how the building is adjusted after duct sealing. It is assumed that the fan is slowed down after sealing to produce the air flow into the zones, but in addition to slowing down the fan, the dampers controlling the OA flow and the relief flow must be adjusted to keep the same ventilation air flow rate and building pressure as before sealing. Thus, savings analyses are performed assuming that this damper adjustment occurs, and assuming that it does not happen. Although this adjustment might or might not happen in an office building setting, it is assumed to always occur in a hospital setting. For any application where the damper adjustment occurs, the effect of exhausting supply leakage through the relief damper is captured by equations 8 to 10:

$$Q_{cool,0\%} = \rho \times \dot{V}_{OA} \times (\square_{OA} - h_{zone}) \tag{8}$$

$$Q_{cool,\mathrm{leaky}} = \rho \times \dot{V}_{OA} \times (\square_{OA} - h_{zone}) + \rho \times \dot{V}_{leak} \times \dot{V}_{relief} / (\dot{V}_{leak} + \dot{V}_{zone} - \dot{V}_{env}) \times (\square_{zone} - h_{sup}) \tag{9}$$

$$Q_{cool,\mathrm{savings}} = \rho \times \dot{V}_{leak} \times \dot{V}_{relief} / (\dot{V}_{leak} + \dot{V}_{zone} - \dot{V}_{env}) \times (\square_{zone} - h_{sup}) \tag{10}$$

where:

$\dot{V}_{OA}$ is the incoming outdoor air flowrate [l/s],

$\dot{V}_{leak}$ is the supply duct leakage flowrate [l/s],

$\dot{V}_{relief}$ is the outgoing relief air flowrate [l/s],

$\dot{V}_{zone}$ is the supply air flowrate to the conditioned zones [l/s],

$\dot{V}_{env}$ is the outgoing air flowrate through the building envelope [l/s], and

$\square_{sup}$ is the enthalpy of the supply air [kJ/kg].

It is worth noting that this savings is independent of outdoor air conditions, which is not the case if the dampers are not adjusted. In that case there is an additional savings associated with reduction in outdoor air flow when the fan is slowed down. The change in outdoor air flow is the product of the OA fraction, the fractional leakage, and the fan flow. The annual implications of that flow change can then be calculated using Equation 6a, while the annual implications of Equation 10 are calculated by simply multiplying Equation 10 by the number of operating hours in a year. For heating of outdoor air, the only savings occur when the dampers are not adjusted, in which case the savings are calculated by using the OA flow change in Equation 6b, along with EHDH calculated using Equation 11, which only indicates heating savings when the OA is so cold as to make the mixed air enthalpy lower than the supply air enthalpy, which is the same value used for office exhaust systems:

$$\mathrm{EHDH} = \sum \left(h_{sup} - h_{mixed}\right) \quad if \ \left(h_{sup} - h_{mixed}\right) > 0, \text{ else } 0 \qquad (11)$$

The annual implications of fan power reduction are calculated from the instantaneous 36.9 kW-21.6 kW, and the fan-power cooling savings in Equation 7. These values need to be multiplied by operating hours, which in the office-building example are 2200 h, and for the hospital example are 8760 hours.

The results of these calculations are summarized in Table 3.

Table 3: Duct Sealing Impacts for Sealing 20% Supply Leakage for 23,600 l/s, 500 Pa system in NYC

Application	**Fan Power [kWh]**	**Fan-Heat Cooling [kWh]**	**Extra OA Cooling [kWh]**	**Total OA Cooling [kWh]**	**OA Heating [kWh]**	**Total Savings [kWh]**
Office CAV Supply (fixed %OA)	33,638	10,659	1,702	3,938	47	49,984
Office CAV Supply (fixed OA, $\Delta P_{building}$)	33,638	10,659	0	2,236	0	46,533
Hospital CAV Supply (fixed OA, $\Delta P_{building}$)	133,942	31,273	0	53,426	0	218,641

The larger savings for hospitals in Table 2 is not surprising, considering that they operate roughly four times as many hours in a year, however some of the results in Table 2 merit some explanation. For example, the fan power savings for hospitals is 4 times higher than for offices, but the fan heat savings is only three times higher. This is because much of the excess fan heat is exhausted through the higher relief air flows in hospitals. The flip side of this is that the relief flows dramatically increase the outdoor air conditioning savings, by a factor of 24, due to a large fraction of supply duct leakage being simply exhausted to the outdoors. The results for all the different exhaust and CAV systems can be normalized by expressing the savings in annual kWh saved per cfm sealed, which are summarized in Figure 1.

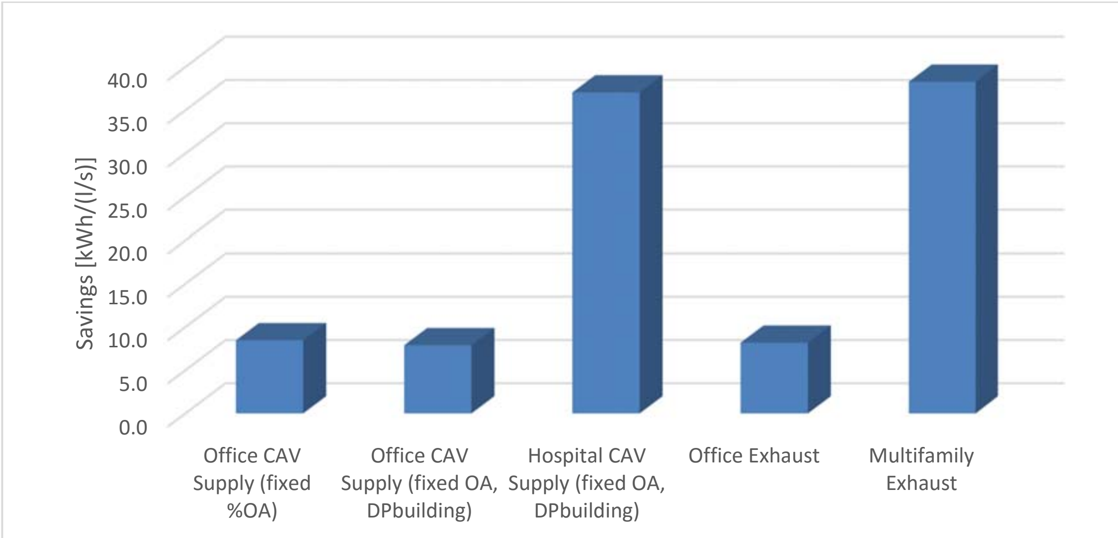

Figure 2: Normalized energy savings from duct sealing

4 DISCUSSION

Although this paper provides fairly comprehensive analyses of duct-sealing savings for CAV and exhaust systems, three key issues were not addressed, namely: 1) the impact of duct leakage on terminal reheat energy use, 2) the magnitude of duct leakage savings for VAV Systems, and 3) the impact of energy recovery on exhaust or relief air.

Starting with the issue of terminal reheat, although the mechanisms for reheat savings were discussed above, the magnitude of the savings was not calculated. This magnitude can however be roughly estimated by assuming that fan power is roughly 40% of HVAC energy use, that reheat is 20% of HVAC energy use for a CAV system, and that 50% of the system leakage is downstream of the terminal boxes. Based upon these assumptions, sealing the 10% leakage downstream of the terminal boxes (50% of 20% leakage) means that the reheat would be reduced by 10%, and that the cooling required to recondition the reheated air being recirculated would be roughly twice the reheat energy (cooling it down to zone-air enthalpy, then down to supply air enthalpy). Thus, for the CAV example above, the annual fan power for the office building is 36.9 kW x 2200 hours = 81,180 kWh, which means the total reheat energy use would be 40,590 kWh, 10% of which is 4,059 kWh, followed by 7306 kWh (2*4059*(1-10%relief)) of cooling, which represents a 23% increase in savings. The fractional increase in savings turns out to be lower for the hospital example, namely 13%, because of the larger relief fraction in that example.

Turning to VAV systems, the savings mechanisms are similar to those for CAV systems, however for the same design flow and pressure there should always be lower absolute savings for VAV systems, as they always have lower fan power due to the fan ramping down at part load, and lower absolute leakage downstream of VAV boxes due to reduced flows at low loads and constant fractional leakage downstream of VAV boxes. Also, VAV systems will not experience the reheat savings associated with sealing leakage downstream of terminal boxes in CAV systems. In fact, there should be a small absolute increase in terminal reheat due to sealing of downstream leakage in VAV systems, as described above.

As for the impact of energy recovery on exhaust or relief air, it should be clear that these systems will have lower savings with respect to outdoor air conditioning, as the overall magnitude of these energy flows will be considerably lower. That said, executing energy recovery could be fairly straightforward for relief air, but not so much for exhaust systems. The issue for exhaust systems is that an additional fan plus supply ductwork would be needed to execute that recovery, unless the exhaust is all located close to supply-air air handling unit.

Finally, it should be noted that for simplicity the pressure difference seen by the fan was assumed to be the same in all example applications, however this is generally not realistic. Supply systems typically have significantly larger fan pressure differentials due to filters, coils, and pressure drops in terminal boxes, all told approximately 2 to 3 times larger than exhaust systems for the same flows. This significantly increases the savings for supply systems relative to exhaust systems.

5 CONCLUSIONS

The key conclusion to be drawn from this paper is that the savings associated with sealing a given amount of duct leakage varies dramatically between applications, with the largest factors being the operating hours of the system, the relief air fraction, and the impact of sealing on OA conditioning needs. Specifically, the savings per l/s of leakage sealed ranges from 8 kWh per l/s to 38 kWh per l/s, a factor of 4.

Another important conclusion is that there can be significant thermal conditioning savings that are independent of climate, which are tied to the relief air fraction, making this an important savings mechanism in hospitals. Moreover, this savings occurs even if the outdoor air and relief air flows remain constant before and after sealing.

6 REFERENCES

Aydin, C., Ozerdem, B., (2006) Air leakage measurement and analysis in duct systems, *Energy and Buildings*, 38 (3) 207–213.

Carrie, F. R., Bossaer, A., Andersson, J.V., Wouters, P., Liddament, M.W., (2000) Duct leakage in European buildings: status and perspectives, *Energy and Buildings,* 32 (3) 235–243.

Franconi, E., Delp, W., Modera, M.P., (1998) Impact of Duct Air-Leakage on VAV System Energy Use. *Lawrence Berkeley National Laboratory Draft Report,* LBNL-42417.

Krishnamoorthy, S., Modera, M., (2016) Impacts of duct leakage on central outdoor-air conditioning for commercial-building VAV systems, *Energy and Buildings* 119, pages 340–351.

Modera, M., Wray, C., Dickerhoff, D., (2014) Low Pressure Air-Handling System Leakage in Large Commercial Buildings, *HVAC&R Journal* Volume 20, Issue 5, pages 559-569.

Wray, C., Matson, N., (2003) Duct leakage impacts on VAV System Performance in California Large Commercial Buildings. *Lawrence Berkeley National Laboratory Report,* LBNL-53605.