HVAC and Lighting Interaction, a Waste Heat Factor by Any Other Name

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ABSTRACT

The interaction of efficient lighting retrofits with HVAC systems has been of great interest to utilities and evaluators both as a source of additional savings and as a potential reduction in savings, particularly for gas and electric utilities in northern climates. The basics are well known—reducing the internal heat gain of a building through lower wattage lighting will decrease cooling load but increase heating load. There is less agreement in the mechanism, magnitude, and timing in the evaluation literature, but these are addressed in technical journals and handbooks. For example, convective and radiant heat transfer from lighting is discussed in ASHRAE handbooks (ASHRAE 2001) and lighting heat transfer is calculated in the Trane® Trace model. Issues that impact HVAC and lighting interaction include night setup (cooling mode) or night setback (heating mode), thermal storage effects, and the use of economizers for cooling.

This paper surveys existing WHF used in a variety of Technical Resource Manuals (TRMs) and evaluations to calculate savings and reviews the primary material used for these calculations. The paper then discusses laboratory research on heat transferred from recessed lighting to occupied spaces.

Introduction

Utilities seek to reduce electrical consumption and demand by encouraging the installation of efficient lighting systems that require less electricity to produce a desired light level. Lighting systems convert only a fraction of their electrical input into useful light output. Much of the rest is released directly as heat into the space around the fixtures, and the light energy is converted to heat over time. Therefore, any upgrade of the lighting system that reduces input wattage reduces the amount of heat that must be removed by the air conditioning system. Conversely, in the winter the reduced heat output must be made up by a building's heating system. This interaction between lighting and HVAC systems is termed a waste heat factor (WHF).

Utilities and utility program evaluators seek to understand the WHF in order to more accurately calculate actual savings in utility program implementation and evaluations. An explanation of the WHF and techniques for calculations is presented in the next sections.

What Is Included in a WHF?

A WHF answers two basic questions: "How much less of the lighting system's heat must be removed by the cooling system after installation of efficient lighting?" and "How much heat must be added during the winter to make up for the lower heat production of efficient lights?"

A cooling energy WHF of 1.1 means that annually for every kWh saved by an efficient lighting system an additional 0.1 kWh is saved in the cooling system. Similarly, a demand WHF of 1.15 means that for a particular peak period each reduction of 1 kW in lighting saves an additional 150 Watts in peak cooling energy. A heating energy WHF of 0.9 means that annually for every 1 kWh saved though efficient lighting, 100 Watts of additional heat must be supplied to the building. Equation 1 shows a simplified method for calculating a WHF for the cooling season.

Equation 1. Cooling WHF

$$WHFc = \frac{Fraction\ of\ year\ with\ mechancial\ cooling\ x\ interaction\ with\ HVAC}{System\ C.O.P}$$

where:

Fraction of year with mechanical cooling:	Expressed as a percentage and will vary widely with climate and with residential or commercial buildings. For homes, it is 0.5 or less for most moderate climates. The reason an Equivalent Full Load Hour Factor (EFLHF) is not used is that it includes air conditioner cycling.
Interaction with HVAC:	This will vary with the type of light fixture, its placement, and the type of HVAC system. It is not particularly well known and is estimated even in most simulation models.
System COP:	The coefficient of performance (COP) is the heat removed divided by the energy used. Dividing the SEER by 3.412 gives a seasonal COP.

An example calculation:

$$WHFc = \frac{\frac{123}{365} days\ x\ 0.9}{2.93} = 0.1035\ or\ 10.35\%$$

where:

Fraction of year with mechanical cooling:	123 days might be a typical cooling season in a moderate or northern climate.
Interaction with HVAC:	For fixtures that are outdoors and in unconditioned spaces the factor is 0, for floor and table lamps it is 1, and for recessed cans it is between 0 and 1 (as shown in this paper).
System COP:	For an average effective field seasonal energy efficiency ratio (SEER) of 10, the COP is 2.93.

In general, the WHF for heating is calculated in a similar manner—the length of heating season and the efficiency of the heating system are inserted into Equation 1. Because heating is often provided by other fuel sources, such as natural gas, the WHF will be calculated in different ways. Where a utility and its stakeholders care only about electric demand, fuel aspects are ignored and only the portion of electric heat in a territory is considered. Where all fuel types are considered appropriate, energy conversion and efficiency factors are used.[1]

For residential heating and cooling, some homeowners let their house temperature float during shoulder seasons and days where heating or cooling needs are light. Thermal mass of the home will also reduce heating and cooling needs during the shoulder season. A homeowner might not turn on the air conditioner if the peak daily temperature reaches, say, 78 degrees Fahrenheit (°F) because the thermal mass of the house is slow to warm and because they anticipate a cooler evening when the windows may be opened. This is one of the reasons why published full load cooling hours (FLCH) overestimate actual air conditioner use for many climates (Cadmus 2009-2013).

Adjusting WHF for the Population of Buildings

The WHFs calculated for cooling and heating are applied to a population of bulbs by calculating varying WHF for bulb placements (e.g., 0 for bulbs outside) and for heating and cooling systems (air conditioners, heat pumps, furnaces, boilers). For example, Figure 1 shows in a residential field survey of 2,447 sockets in the northeastern United States, we found that 8% of sockets are located in exterior spaces (Cadmus 2012). Therefore, assuming that garages are not conditioned, any WHF would be multiplied by 92% because the other 8% would not interact with HVAC systems.

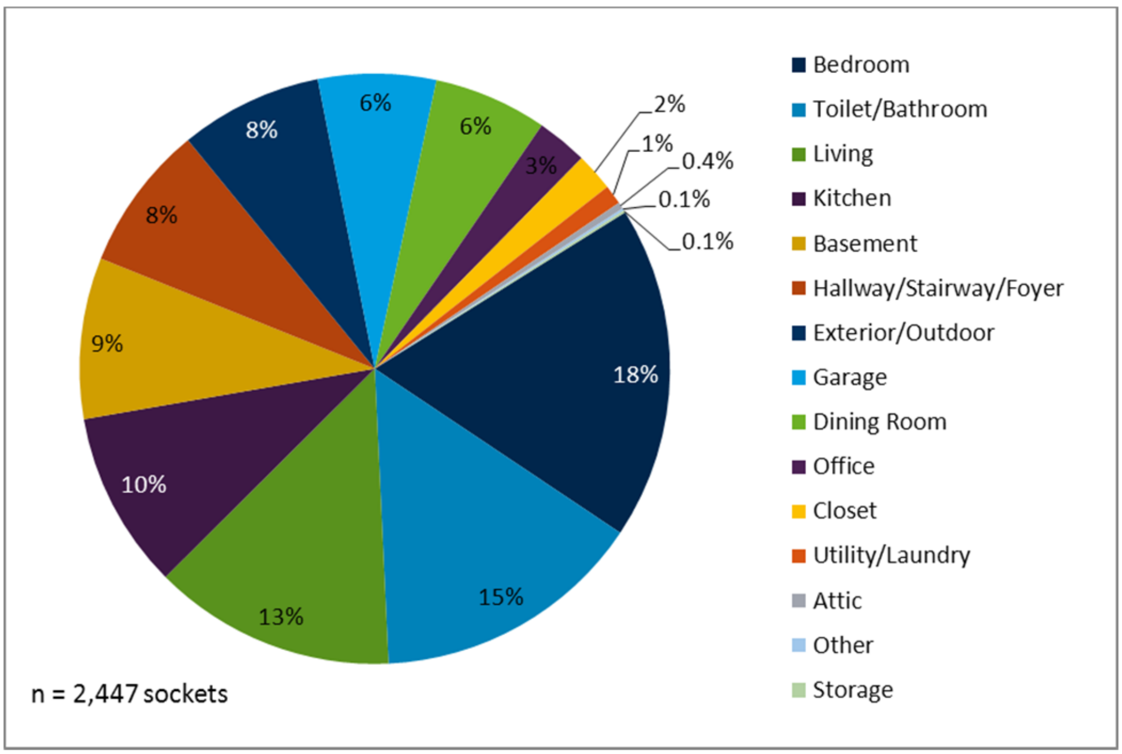

Figure 1. Example of Socket Locations for a Residential Lighting Evaluation

One policy issue with a combined WHF is how to treat different fuel types. For a population of houses with electric heating and cooling systems, the annual consumption savings are simply a matter of accounting for system type, and the heating and cooling factors can be combined. For summer peaking utilities, the demand savings are entirely based on cooling and on summer demand for electricity. For

[1] Fuel oil contains roughly 140,000BTU, a kWh contains 3,412 BTU, and a therm of gas contains 100,000 BTU.

buildings with fossil fuel heat and electric cooling fuel, switching policy issues must be considered in accounting for interaction. An efficient lighting system will reduce summer electricity consumption and demand but will increase gas consumption in the winter.

WHF in TRMs and Evaluations

Waste heat factors vary among states as evidenced by different approaches in deemed savings values captured in TRMs. As shown in Table 1, the residential cooling WHF varies from state to state and averages about 7%. The variation are in part due to the length of the cooling season but also to differing assumptions regarding HVAC interaction, user behavior, and average system COP. The Mid-Atlantic TRM (VEIC 2011) uses a low COP of 2.5, which is equivalent to an EER of 8.5, and a relatively high interaction factor of 0.45 that essentially indicates that nearly all bulbs are fully interacting with the HVAC system for nearly half (180 days) of the year.

Table 1. WHF in Several Residential TRMs

Sources of WHF	Heating Adjustment	Cooling Adjustment	Adjusted WHF	Includes Saturation of Cooling Equipment
Regional Technical Forum (RTF)	-22%	7%	86%	Y
New York TRM (TecMarket Works 2010)	-12%	3%	91%	Y
Vermont TRM (VEIC 2012)	N/A	6%	106%	
Ohio TRM (DPS 2010)	N/A	7%	107%	
Mid-Atlantic TRM 2012 (VEIC 2011)	N/A	14%	114%	Y
State of Illinois Energy Efficiency TRM 2012 (Illinois 2012)	N/A	6%	106%	
Multifamily section of Illinois TRM (Illinois 2012)	N/A	4%	104%	
Mean		**6.64%**		

Table 2 shows non-residential WHFs for heating and cooling for various TRMs. The factors are larger and vary much more widely than the residential factors in Table 1. For example, the heating penalty averages 39% versus a range of 12% to 22% for residential TRMs. The cooling factors average 14% versus about 7% for residential. This is not surprising given the wide variety in commercial building design. A small medical office will have many of the attributes of a house while a large commercial building can be core-dominated with cooling necessary 12 months of the year.

Table 2. Comparison of Heating and Cooling WHF for Commercial Buildings

Source	Fraction of Annual Lighting Energy Removed by Heating System	Heating Efficiency	Heating Penalty (kBtu/kWh)	Heating Penalty (kWh/kWh)	Fraction of Annual Lighting Energy Removed by Cooling System	Cooling COP	Cooling Benefit (kWh/kWh)	Notes
Rundquist 1993	0.31				0.48			Average of Boston and Springfield, Massachusetts, values.
ASHRAE w/efficiency of 75%, COP of 3.0	0.31	0.75	1.41	0.41	0.48	3.00	0.159	
Optimal Energy 2008	0.27	0.75	1.08	0.32	-	-	-	Uses ASHRAE fractions.
Connecticut (CLP, 2007)			0.79	0.23	0.50	2.40	0.208	
MA Utilities 2004	0.44		1.49	0.44	0.26	3.06	0.086	heating: 0.7x7.5/12 cooling: 0.7x4.5/12
New Hampshire 2004	0.44		1.49	0.44	0.26	3.06	0.086	Identical to Massachusetts Utilities report (2004).
LBNL Modeling Study, Ozman							0.190	National number; no COP given but is likely lower than 3.0.
Ohio C&I Calculator							0.120	
VT TRM (VEIC, 2012)	0.27	0.75	1.24	0.36	0.29	2.50	0.116	Uses ASHRAE fractions, heating multiplied by 0.7.
Colorado TRM, Appendix E (Xcel)			0.89	0.26			0.110	
DPS 2010			2.40	0.70			0.115	Values shown for small retail in Poughkeepsie, New York.
Average	***0.32***	***0.75***	***1.33***	***0.39***	***0.40***	***2.74***	***0.14***	
Maximum	***0.44***	***0.75***	***2.40***	***0.70***	***0.50***	***3.06***	***0.21***	
Minimum	***0.27***	***0.75***	***0.79***	***0.23***	***0.26***	***2.40***	***0.09***	

Understanding Heat Flow in Lighting Fixtures

It is important to understand the heat flow in lighting fixtures because heat can fully enter a space and interact with the building's HVAC system or leave the space and not contribute to the WHF. In a residential environment, a ceiling-mounted fixture such as a recessed can will allow some heat to rise into an attic space and be lost from conditioned space. In a commercial building, heat can rise above a dropped ceiling and be lost from the space or rise above the ceiling and enter a return air plenum where all of the heat enters the HVAC system. However, data are limited regarding how much heat enters living spaces served by a fixture and how much rises into unoccupied space.

This paper describes an experimental study of the heat transfer characteristics of recessed lighting in which we measured and documented the amount of heat released by several types of recessed fixtures through a drywall ceiling cross-section. This measurement is important, because residential calculations and commercial modeling approaches all require an estimate of the amount of heating leaving a space.

Experimental Setup

Researchers built a calorimeter using closed cell foam residential insulation. The calorimeter consisted of a box with known heat transfer parameters (Figure 2). Three different setups were constructed with varying insulation values: a box with 0.5-inch walls and an R-value of 3.3; a box with 2-inch walls and an R-value of 10; and a box with double-thick 2-inch walls and a total R-value of 20. A standard 45-Watt incandescent indoor floodlight was mounted into two different fixture setups: insulation contact rated (ICR); non-insulation contact rated; and both mounted into three-quarter-inch drywall. The bottom of the drywall-mounted fixture was left open as it would be in an actual installation. Four Onset TMCx-HD ambient temperature sensors inside the box measured the air temperature as the light heated up the interior of the box and recorded the steady state temperature once a heat balance had been achieved.[2] A temperature sensor on the outside of the box measured the ambient temperature of the testing room.

Figure 3 shows the temperature sensors entering the top of the box, and Figure 4 shows the bottom of the calorimeter where the recessed fixture is open, which allows the heat to descend and be dispersed into the testing area, similar to how heat is distributed in a home—descending heat enters the living space.

Experimental Procedure

The fixture's power draw was metered using a Watt's Up? PRO plug-through power meter to account for any variation from the bulb's rating.[3] Heat loss was calculated using the dimensions and R-values of the calorimeter's walls. The test was first run to determine how much of a bulb's power could be accounted for in the calorimeter. Because all of the power drawn by the bulb is directly converted either to heat or to light, which is in turn converted to heat, the power drawn by the bulb should match the heat loss of the calorimeter in Watts. The lower half of the test setup was covered to completely enclose the bulb and four additional temperature sensors were mounted inside the lower box to gather the temperature data. The ICR fixture was tested two ways: bare and with R-19 batt insulation covering the fixture.

[2] The TMCx-HD ambient temperature sensor is a product of Onset Computer Corporation. http://www.onsetcomp.com/
[3] The PRO plug-through power meter is a product of Watt's Up?. https://www.wattsupmeters.com/secure/index.php

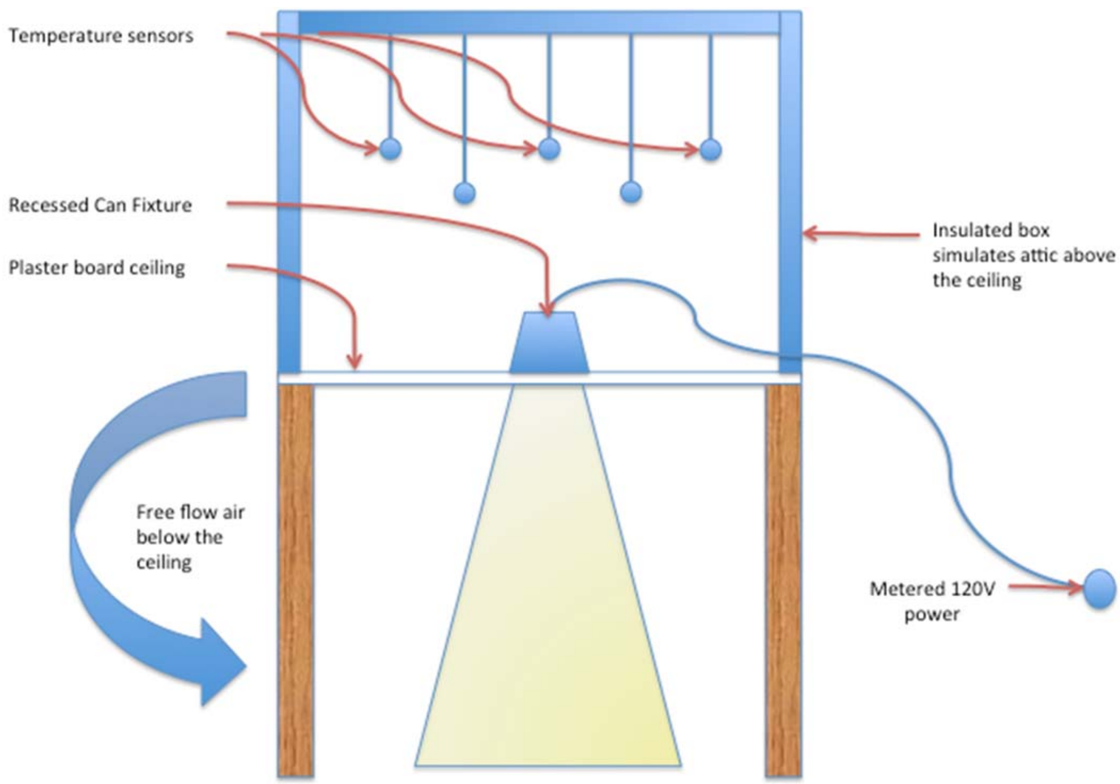

Figure 2. Calorimeter Test Setup

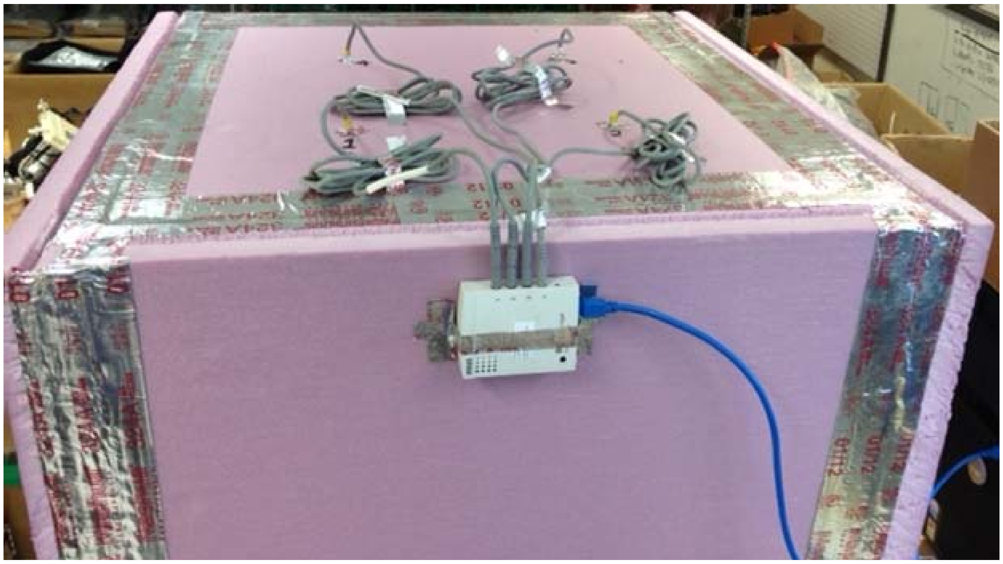

Figure 3. View of Top of Calorimeter Showing Temperature Sensors

Figure 4. View From Below Showing Recessed Fixture

Each test was run until steady state was achieved. Temperature readings in the upper and lower chambers were averaged to calculate the heat lost from the calorimeter for comparison with the energy drawn by the bulb. Because at steady state the mass of the air inside the box was insignificant, the heat released by the bulb should match the calculated amount of heat escaping through the insulation walls. At steady state the bottom box had an inside temperature of 112.3°F and the outside temperature was 69.62°F. The following equation was used to calculate the heat lost from the bottom box:

Equation 2. Steady State Raw Energy for Calibration Test

$$\frac{\left(\frac{(112.3°\text{F}-69.62°\text{F})\times\left(\frac{24.75\,ft^2}{10.85\,{}^{hour\times°\text{F}\times ft^2}/_{BTU}}\right)}{3.412\,{}^{BTU}/_{Watthour}}\right)}{240.0\,{}^{intervals}/_{hour}} = 0.1188\,Watthours$$

Averaging the heat lost during the calibration test yielded 27.68 Watts. Repeating the process using the top box and the edge of the drywall housing the fixture yielded a total heat loss of 46.96 Watts. The average power used by the incandescent light bulb during the test was 48 Watts. Consequently, the calibration test accounted for 97.84% of the power drawn by the bulb. This factor showed that the method worked well. The 97.84% was used as a calibration factor for the remaining tests to better match heat flow in the experimental setup.

Heat Distribution Testing

Each of three fixture arrangements was tested with three different levels of insulation, for a total of nine tests. Tests continued until the box reached a steady state temperature. The varying thicknesses of insulation resulted in different steady state temperatures, which allowed us to see if different temperatures above the bulb impacted the heat transfer from the bulb. In an actual home, the temperature of the space above the bulb will vary not only by fixture type but also by insulation, shape of the space above, and the season.

Figure 5 shows each of the nine tests to illustrate the steady state temperature achieved above the fixture,[4] which is analogous to the temperature in an attic or ceiling cavity above the light and the percentage of heat rising from the bulb. The remainder (or 1% rising value) is the amount of heat that enters living space and potentially interacts with the HVAC system. For each fixture type, the three data points represent the three insulation thicknesses with the rightmost data point showing the highest steady state temperature caused by the highest insulation level on the calorimeter.

As expected, the non-ICR fixture released the most heat because these fixtures are constructed of thin single-wall metal and cannot touch insulation because their metal casings heat to excessive temperature. The steady state temperature affected these fixtures most; varying temperatures caused the portion of the rising heat to vary from 17% to 33% or released the least amount of heat to the living space (67% to 83%).

The ICR fixture without insulation released the next-highest amount of heat upwards (14% to 22%) and slightly more to the living space below (78% to 86%).

The ICR fixture with insulation released the least heat upwards (13% to 16%), and the results were least sensitive to temperature above.

[4] See full table in Appendix 1. Final Recessed Lighting Data.

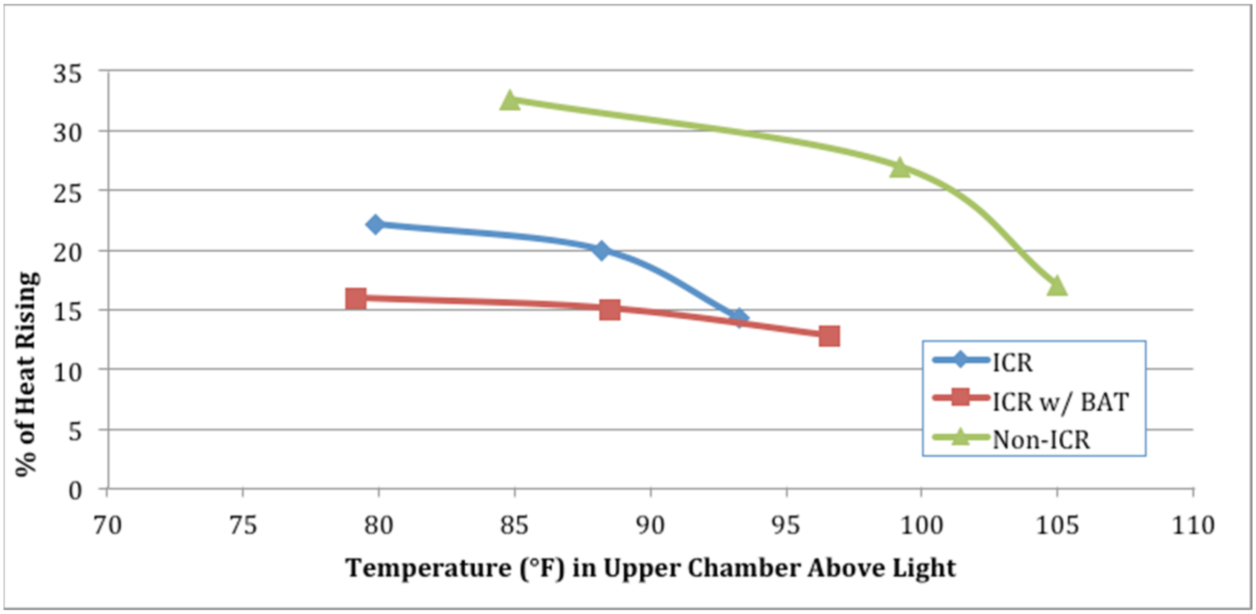

Figure 5. Heat Released Upwards by Recessed Fixtures

A few interesting trends emerge from Figure 5. At cooler attic temperatures that would occur during non-peak cooling days, the heat lost from the non-ICR can approach 35%. At cooler temperatures during shoulder and heating seasons, it is likely that much more of the heat is lost upwards and that the heating penalty for bulbs in older style cans is low.

As the temperature in the space above the fixture rises, the amount of heat moving upwards decreases to less than 20% for all three fixture types. This makes sense because conductive and convective heat transfer is driven by a temperature gradient. This means that the recessed cans begin acting like other fixtures in the living space at high attic temperatures, releasing most of their heat to the space below. On a peak cooling day where attic temperatures could reach 120°F, it appears that heat rising into an attic is a minor consideration.

With better sealed insulated fixtures, the amount of heat rising decreases (i.e., insulated ICR versus non-ICR). This means that increased insulation of the cans could increase cooling load from the lights, no matter their efficiency. Future testing will use different kinds of light bulbs to determine whether the trend would continue across different bulbs, especially new, high-efficiency compact fluorescent lights (CFL) and light-emitting diode (LED) bulbs. Future testing will also include linear fluorescents mounted into dropped ceilings to simulate lighting systems used in larger commercial buildings.

Another aspect to model is air movement. The experiment design eliminates air movement, but residential attics are vented and buildings are often leaky. If there is air movement, even if the air is warm, the convective flow component of heat transfer may be significant.

Release of Heat in Commercial Lighting Fixtures

There are many configurations of linear fluorescent fixtures in commercial spaces. Figure 6 shows examples from routines in Trane Trace™ building simulation software showing some of the models' assumptions.[5] While these values appear plausible, we recommend additional testing be conducted to check these values. We may conduct laboratory tests on some of these fixture types later this year.

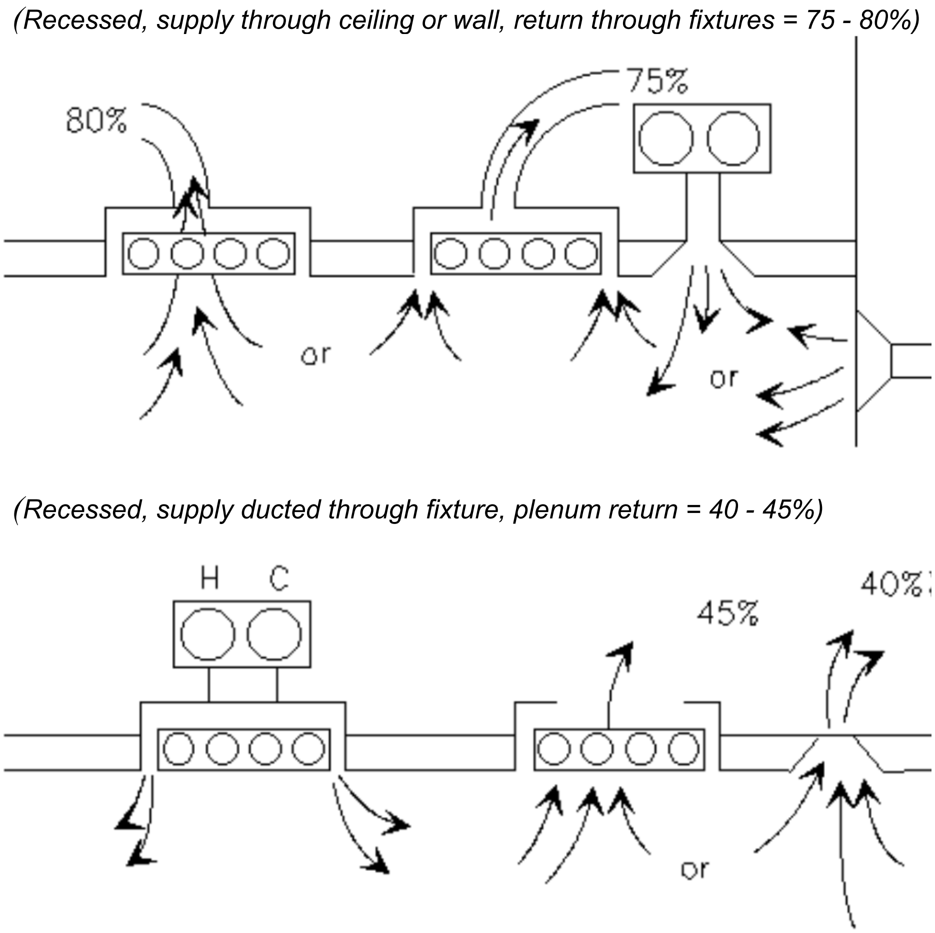

Figure 6. Heat Flow Assumed in Trane Trace Simulation Model

Conclusions

Interaction between lighting and HVAC, although simple in concept, varies greatly by fixture, HVAC type, climate, and building type. Calculations vary too, as methods for calculating WHF are not yet standardized. Some evaluations use basic estimates of the length of the heating and cooling season in weeks; others use secondary materials and simulation modeling to more precisely calculate the actual interaction of lighting and HVAC systems. This has caused the WHF claimed in various utility territories and states to vary widely. We recommended that additional future efforts concentrate on reducing variability in calculating WHF.

In this paper we explored several of the components of WHF that need to be part of even a basic calculation of WHF (including system COP and efficiency), the portion of bulbs used in conditioned space, and the length of the heating and cooling seasons. We examined one aspect in detail—the release of heat from recessed cans—in part to provide new information and in part to show that even this small

[5] Trace™ software is a product of Trane®. http://www.trane.com/

aspect of WHF varies greatly with fixture type and insulation and with the temperature in the space above the can. Even ICR cans with insulation will still lose 15% of their heat upward, and non-ICR cans will lose nearly 30% of the heat generated. During the heating season, this upward heat loss may approach 50% for non-ICR cans; this means that a heating penalty for these fixtures may be exaggerated. We recommend that more testing be conducted, including laboratory testing and *in situ* testing where temperatures are collected in spaces above recessed cans.

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Regional Technical Forum

Appendix 1. Final Recessed Lighting Data

Date	Time (hours)	Fixture Type	Insulation	Box Surface Area (sq. ft.)	Box R-value	Adjustment Factor	Steady State Temp. (F)	Energy Used (watt-hr)	Power (adjusted) (W)	Percentage of Bulb Power Going Up
10/9-10/10	22	ICR	Single	17.417	10.85	0.97836	89.19	205.977	9.570	19.94%
10/26	2.14167	ICR	Double	20	20.85	0.97836	93.24	14.377	6.862	14.30%
10/18-10/19	19	ICR	Small	19.339	5.885	1	79.84	201.626	10.612	22.11%
10/17-10/18	22	ICR w/ BAT	Single	13.597	10.85	0.97836	88.46	76.836	7.236	15.08%
10/16-10/17	16	ICR w/ BAT	Double	16.28	20.85	0.97836	96.15	96.54	6.167	12.85%
10/19-10/21	44	ICR w/ BAT	Small	15.259	5.885	1	79.12	337.18	7.663	15.96%
10/11	4.833	Non-ICR	Single	17.417	10.85	0.97836	99.15	61.232	12.950	26.98%
10/12	4.417	Non-ICR	Double	20	20.85	0.97836	104.95	35.444	8.202	17.09%
10/23-10/25	43	Non-ICR	Small	19.339	5.885	1	84.79	672.881	15.648	32.60%

ICR = insulation contact rated