Final Report

Energy Savings Analysis and Thermal Comfort Performance of XChanger

Submitted to:

XChanger Companies, Inc.

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Executive Summary

Traditional heating and cooling systems operate by supplying and returning air overhead with the goal of creating a thermally well-mixed space. Thermal stratification can develop due to the buoyancy of warm air, especially in spaces with tall ceilings. In cooling, the traditional system can lead to higher return air temperatures than necessary. In heating, the traditional system can lead to overheating the unoccupied upper portion of the space. The traditional system therefore will consume more energy to bring the space to set point than necessary.

The XChanger system retrofits the traditional, overhead system by altering where air is supplied and returned. In cooling, air is supplied low in the occupied zone and returned at a middle-level. The goal in cooling is to only condition the occupied space, letting the upper zone stratify. In heating, air is supplied near the ceiling and returned near the floor. The goal in heating is to create a well-mixed space by drawing down warm air from the ceiling to the occupied zone.

The University of Maryland Center for Environmental Engineering (CEEE) investigated the performance of the XChanger system as part of the Maryland Industrial Partnership (MIPS) program. CEEE is a leader in the research of energy conversion technology with broad experimental and software experience. The XChanger system was experimentally and numerically investigated in terms of its energy savings potential and thermal comfort performance. A summary and the results of this work are contained in this report.

A manufacturing facility serves as an experimental testbed of the XChanger system. Ten vertical temperature profiles (floor-to-ceiling) were measured over the course of 15 months, with the XChanger installation occurring in month six. An analysis of the site's utility bills is used to calculate the monthly energy savings of a series of four retrofits, one of which is the XChanger system.

Building energy modeling and computational fluid dynamics simulations are used in addition to the experimental, on-site testing. Building energy modeling involves generating a detailed building model and numerically simulating its performance over one year using real weather data. Building simulation is used to estimate the energy savings of the other three retrofits (not XChanger) as well as to provide justification for the measured energy savings. Also, building simulation provides a means to investigate the effect of pressure drop of the XChanger ducting on energy savings. Computational fluid dynamics involves solving for the detailed airflow and temperature field inside a room. The model was validated using experimentally measured temperature profiles, then used to investigate stratification and thermal comfort under various duct configurations.

The utility bill analysis shows that the XChanger system saves 28.8% HVAC electricity yearly. Gas savings were not investigated for one year, but show 22.3% savings in January through February. Differences in outdoor temperature between the periods before and after the XChanger installation will influence the building's energy consumption and thus, the measured energy savings. For example, summer 2015 had 832 cooling degree days while summer 2014 had 682 cooling degree days. Since summer 2015 was warmer than the baseline period, the measured energy savings can be considered conservative.

Measured temperature profiles track the temperature distribution in the test facility over the course of one year. In summer, XChanger increases the air temperature near the ceiling and reduces the temperature in the occupied zone. This implies that the set point may be able to be increased, thereby saving energy. In winter, the room with the largest heating load saw a reduction in stratification from 7.1 K to 3.5 K. The manufacturing rooms were in cooling mode during portions of winter (due to their large cooling load) and therefore show little difference during winter.

Building energy simulation shows that the XChanger system saves 19.3% HVAC electricity and 25.2% gas yearly with its extra pressure drop and 37.4% HVAC electricity and 7.7% gas without added pressure drop. These results bracket the measured energy savings determined through the utility bill analysis and show that the pressure drop of the extended ducting has the potential to reduce the energy savings significantly. Computational fluid dynamics shows that, in cooling season, supplying air near the floor increases stratification while maintaining thermal comfort. All cases considered show that thermal comfort is maintained in the occupied zone according to ASHRAE thermal comfort metrics.

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1   Introduction

This document is a final report on the effects of four energy saving technologies in a manufacturing facility located in the Mid-Atlantic region of the United States. The energy saving technologies were a replacement of the existing metal halide lighting fixtures with light emitting diode (LED) lights, replacement of existing rooftop units (RTUs) with more efficient units, replacement of an existing air compressor with a higher efficiency unit, and the installation of a new air control device called the XChanger. The test facility and regional weather are described in Section 4 of this report. This report concentrates on the energy savings of the XChanger. The energy savings of each retrofit are estimated through simulation and engineering calculations, with the XChanger savings taken as the difference between observed energy savings and savings due to other retrofits.

The goals of the XChanger system are to improve occupant thermal comfort, improve indoor air quality, and save energy used for space heating and cooling in forced-air systems. The XChanger aims to achieve this goal by strategically supplying and returning air to create thermal stratification. The traditional, overhead, mixing ventilation system is replaced with ducting that changes the location of supply and return air based on the season. In cooling, air is supplied low, near the occupants, and returned just above the occupied zone. This configuration will supply cool air directly to building occupants, cooling them through convection. In heating, air is supplied near the ceiling and drawn down to low return ducts. This configuration will draw hot air down, breaking the significant stratification found in overhead systems.

The impact of such a system on a facility's energy use was measured in a test facility. The test facility is Holmatro Inc, located in Glen Burnie, Maryland, a global manufacturer of state of the art rescue equipment. Vertical temperature profiles and duct temperatures were measured for use in building simulation software and computational fluid dynamics (CFD). Data was collected for over one year to understand the system's performance during each season.

The objective of this study is to provide an estimate of the monthly energy savings of each building retrofit, including new LED lights, new rooftop-units, a new air compressor, and the XChanger system. This objective will be achieved through a utility bill analysis, measured temperature profiles, building energy simulation, and CFD simulation.

2   Building Energy Simulation in EnergyPlus

EnergyPlus is a state-of-the-art, whole-building energy simulation software developed by the U.S. Department of Energy. A building model consists of building geometry, geographical location, orientation, actual weather data from Baltimore-Washington Airport, HVAC (heating, ventilation, and air conditioning) systems, lighting, plug loads, occupant load, thermal mass, and a number of other parameters. The model is geo-located to import the angle and intensity of solar radiation.

The software calculates heating and cooling load, subsequent HVAC system performance, and space conditions by time-stepping through one year with actual weather data. The time step used in this simulation is 10 minutes. Models can be calibrated by comparing the predicted electricity and gas use to utility bills or measured hourly or daily electricity consumption.

Holmatro's building consists of five zones – the offices, large manufacturing room, small manufacturing room, storage, and assembly room as shown in Figure 1. Each zone is heated and cooled by packaged RTUs (rooftop-units). XChanger was installed in the large manufacturing room, small manufacturing room, and storage room. The offices and assembly room do not have the XChanger system. Although this fact is not captured by the utility bill analysis due to difficulty in estimating the thermal load of each individual room, the EnergyPlus simulation will be able to estimate energy savings by room.
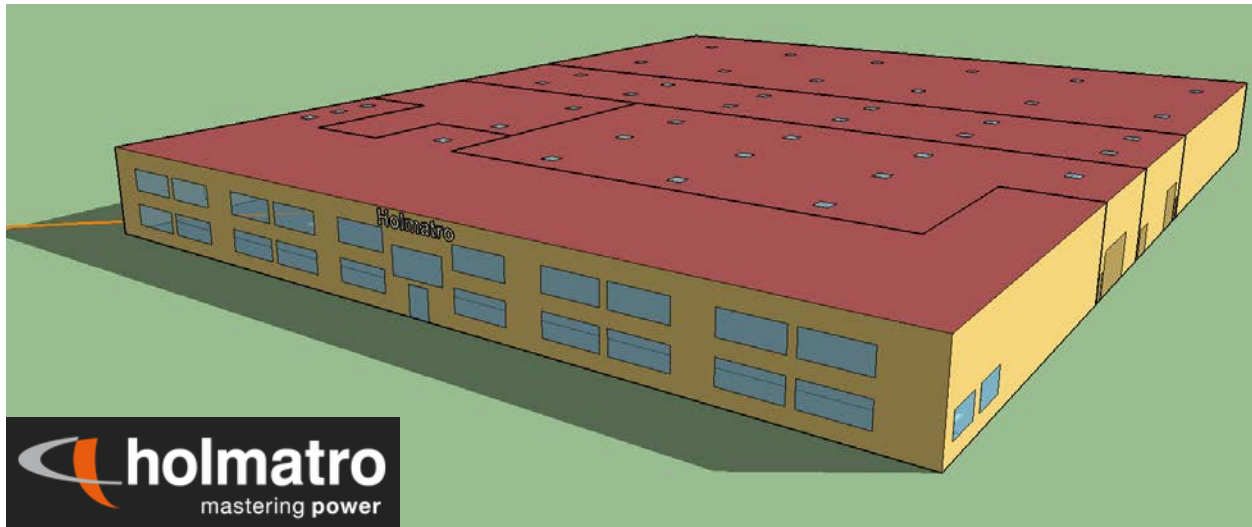

Figure 1: Test facility, Holmatro, model exterior and zones

EnergyPlus was chosen as the simulation software because it is capable of incorporating vertical temperature profiles into the heat balance calculation. Traditionally, building energy simulations assume that a space is well-mixed, meaning that the room has uniform temperature and other properties. This well-mixed assumption produces inaccurate results for tall spaces, such as atria, hangars, or warehouses with significant vertical stratification.

Measured vertical temperature profiles were included in the building model for both manufacturing rooms, the storage room, and the assembly room. Temperature profiles were averaged by season. Since XChanger was installed in July, it was convenient to break the year into six seasons – three before XChanger and three after, assuming spring is comparable to fall.

To enter a temperature profile, the room height is non-dimensionalized such that 0 represents the floor and 1 represents the ceiling. Nodes are defined at a given height with a corresponding temperature offset from mean air temperature (MAT). Along with node temperatures, two additional inputs are required: return air offset and thermostat offset. The return air offset specifies the temperature of the return air, which for this study was assumed based on the height of the return duct. The thermostat temperature is the temperature near where the thermostat would be located, about 1.5 m from the floor. An example temperature profile is shown in Figure 2.

$$T(z) = T_{air,i} - T_{MAT}$$
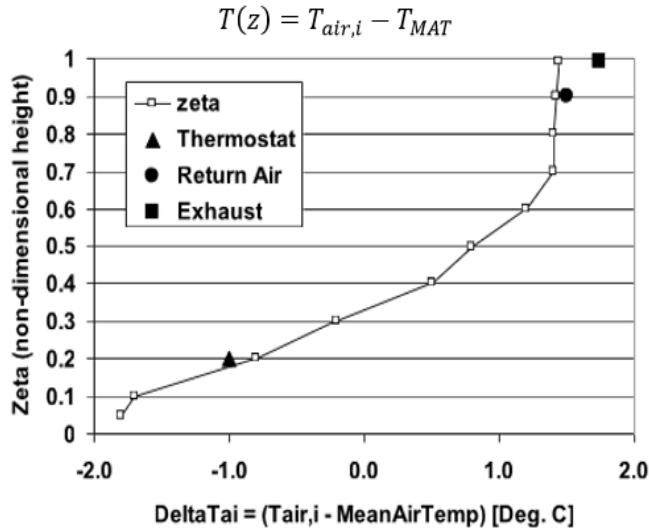
Figure 2: Example temperature profile (source: EnergyPlus reference manual [1])

For any building simulation, the model must be calibrated to measured data. Typical building simulations have hundreds of inputs, some of which are unknown or could reasonably assume a wide range of values. In this study, the plug load and schedule is unknown due to the manufacturing machines. Therefore, it is varied to 'tune' the model to monthly gas (from utility bills) and daily net facility electricity consumption. The results are shown for monthly electricity in Figure 3 and gas in Figure 4.
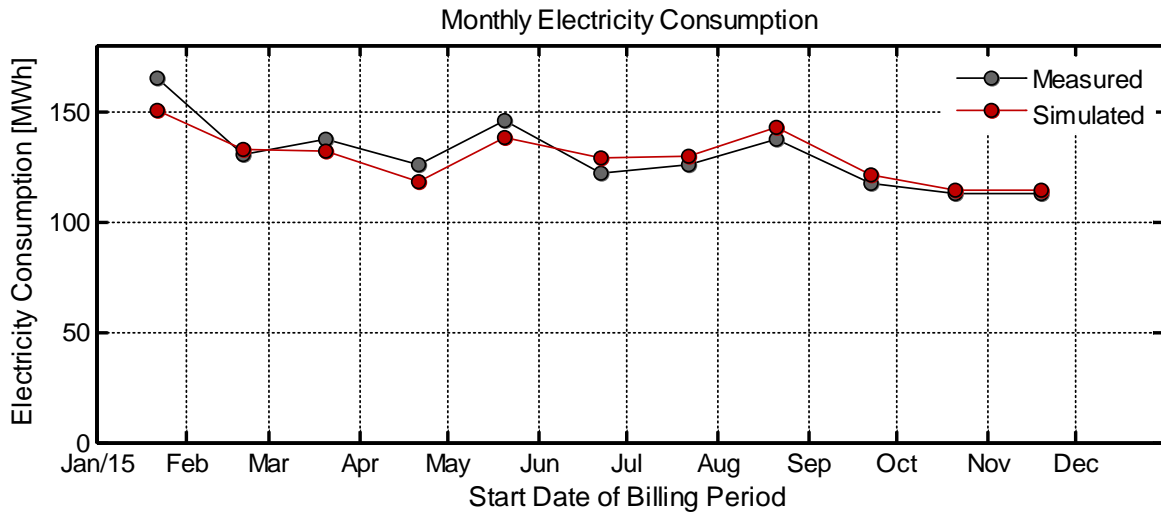
Figure 3: Building simulation monthly electricity calibration
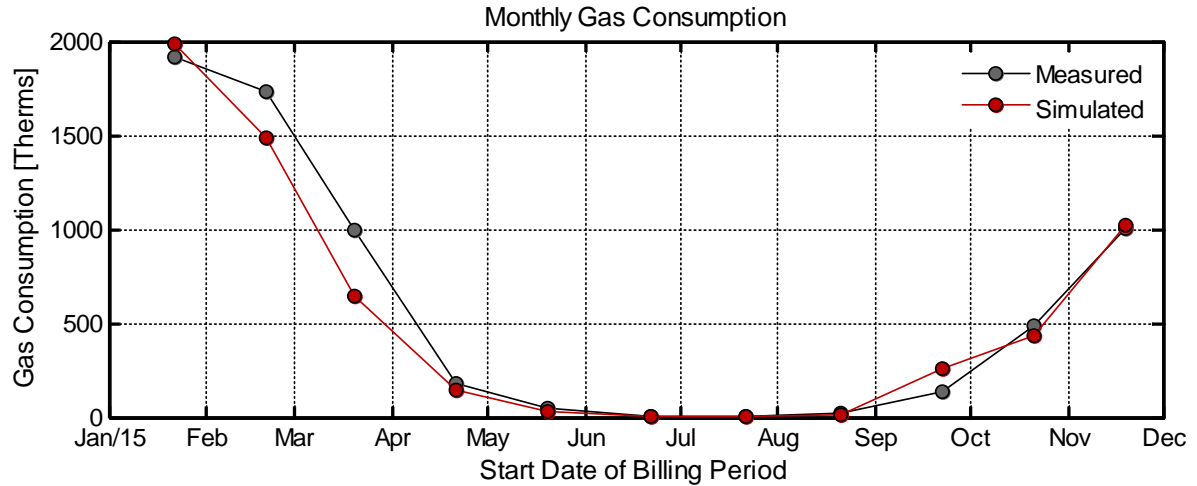
Figure 4: Building simulation monthly gas calibration

ASHRAE recommends two statistics to judge the calibration of a building model [2]. The normalized mean bias error (NMBE) characterizes whether the model over- or under-predicted measured data. The coefficient of variation of the root mean squared error (CVRSME) characterizes overall how accurately the simulated data compares to measured data. A value closer to zero means the model is more accurate. Based on Table 1, the electricity is calibrated very well, but the gas is not. The model under-predicts gas use during spring, but over-predicts gas use during fall. It is possible that this is due to malfunctioning economizers or that the economizer is not simulated properly. Figure 5 shows that the model predicts daily electricity to within about 15%, though deviations of up to 50% exist for low-load days (Sundays).

Table 1: ASHRAE Guideline 14 statistics

	NMBE [%]	CV(RSME) [%]
Electricity (daily)	-0.27	7.75
Electricity (monthly)	1.74	5.01
Target (hourly)	< 10	< 30
Target (monthly)	< 5	< 15

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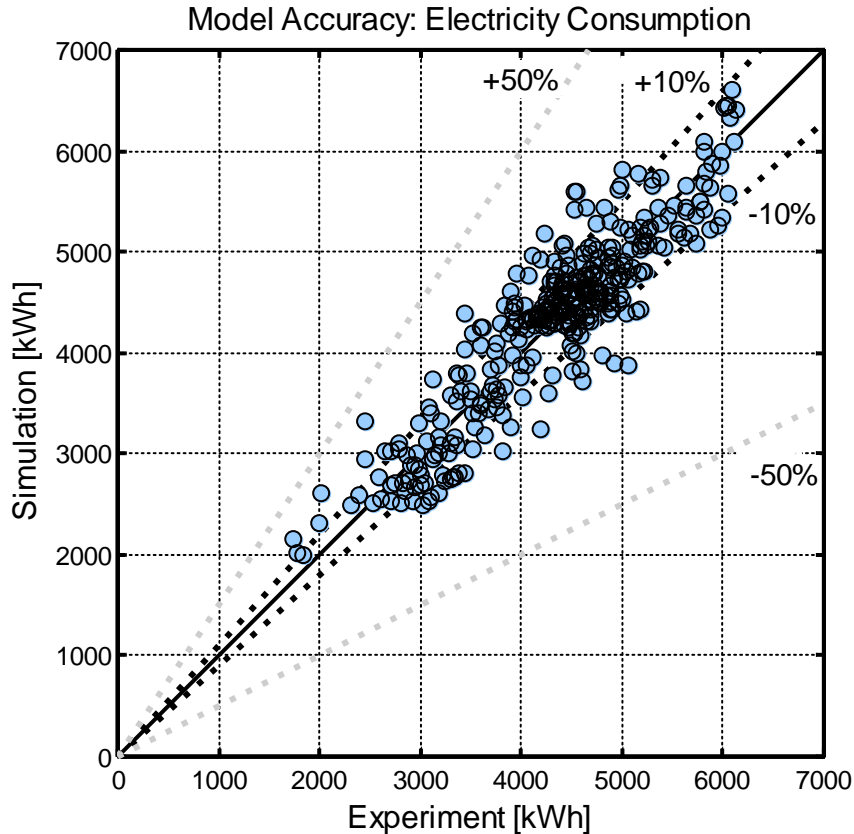
Figure 5: Percent difference between model and measured electricity (daily)

3 Computational Fluid Dynamics Simulation

Computational Fluid Dynamics (CFD) is a method of solving fluid flows numerically. The governing partial differential equations of mass conservation, momentum conservation, and energy conservation are solved numerically, with constitutive relations and turbulence models closing the set of equations. CFD is used in this study for cooling applications to understand the room air velocity and temperature distributions under different duct conditions. This information can be used to study stratification and thermal comfort.

Typical workflow for a CFD simulation includes generating a mesh, defining boundary conditions, setting up the solver to simulate the relevant physics, running the simulation, and processing the results. A mesh is a discrete representation of the flow domain. In this study, the flow domain is a room, which is broken into about 750,000 cells. These cells compose the mesh. As the simulation progresses, the properties (temperature, velocity, etc.) in each cell are solved more accurately and the flow approaches its solution.

CFD simulations require a lot of computing power, taking about two days per simulation in this study. Since the test facility is so large, it was chosen to simulate one-sixth of the large manufacturing room to cut down on the computational cost. The portion of the room that was simulated is shown in Figure 6. This portion is conveniently served by one RTU.
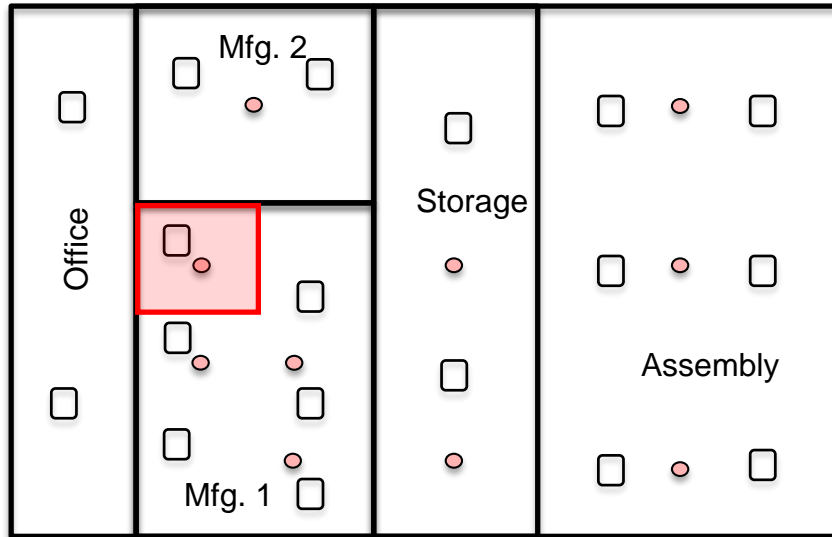
Figure 6: Chosen region for CFD simulation
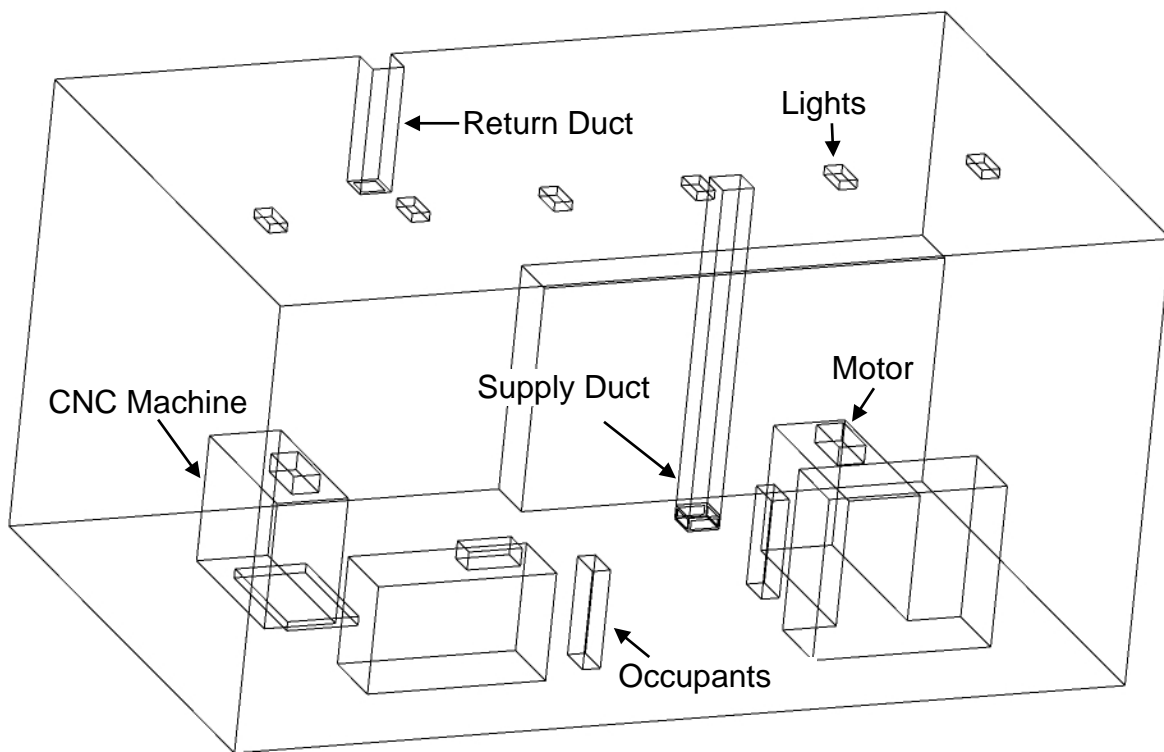
Figure 7: Example CFD model

An example room with its appliances and features is shown in Figure 7. It was more important to create a room with similar heat sources to the real room than go for photo-realism. The supply and return duct heights are varied in the parametric study while all other features remain the same.

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Important input variables are the supply air temperature, flow rate, diffuser face area, and height, the return duct height, the heat flux of the heat sources (lights, occupants, CNC machines), and the floor, wall, and ceiling temperatures. Measured average summer temperature profiles are used for the boundary conditions along the walls. The temperature of the floor and ceiling are obtained from the measured floor and ceiling temperatures. The occupants have a constant heat flux boundary condition equal to a typical metabolic rate. The lights have a constant heat flux boundary condition equivalent to the heat generated by the actual lights in the test facility. All other surfaces have specified temperature boundary conditions. Radiation is ignored for this study because it does not influence stratification directly. An unstructured tetrahedral mesh was used. Its cross section for the baseline case is shown below.
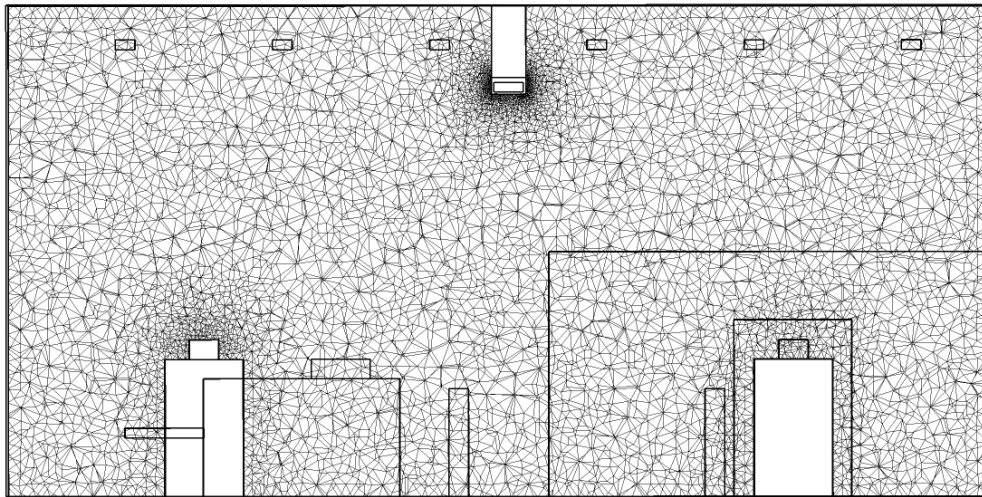
Figure 8: CFD mesh for the baseline, overhead system

It is important to validate a CFD model using experimental data to show that the model predicts accurate results. For this study, measured temperature profiles were compared with several temperature profiles from the CFD simulation for both the baseline and XChanger cases. The results are shown in Figure 9. The deviation between the CFD model prediction and the average measured temperature profile are within experimental uncertainty. The model can therefore be considered validated.

It was found that the same boundary conditions could not be used for both the baseline and XChanger cases. The average supply air temperature from 8/30 to 9/2 was measured to be 12.7 °C. Flow rate is not measured, though this RTU has two nominal speeds: 2.6 m^3/s and 1.4 m^3/s. Since there are times that the RTU is not supplying air, it is expected that the average supply air velocity is somewhere between 0 and 2.6 m^3/s. It was found through trial and error that 12 ºC and 0.56 m^3/s (2 m/s face velocity) provide a very accurate prediction of room air temperature for the baseline case. For the XChanger case, it was found that 14 ºC and 1.12 m^3/s (4 m/s face velocity) was best.
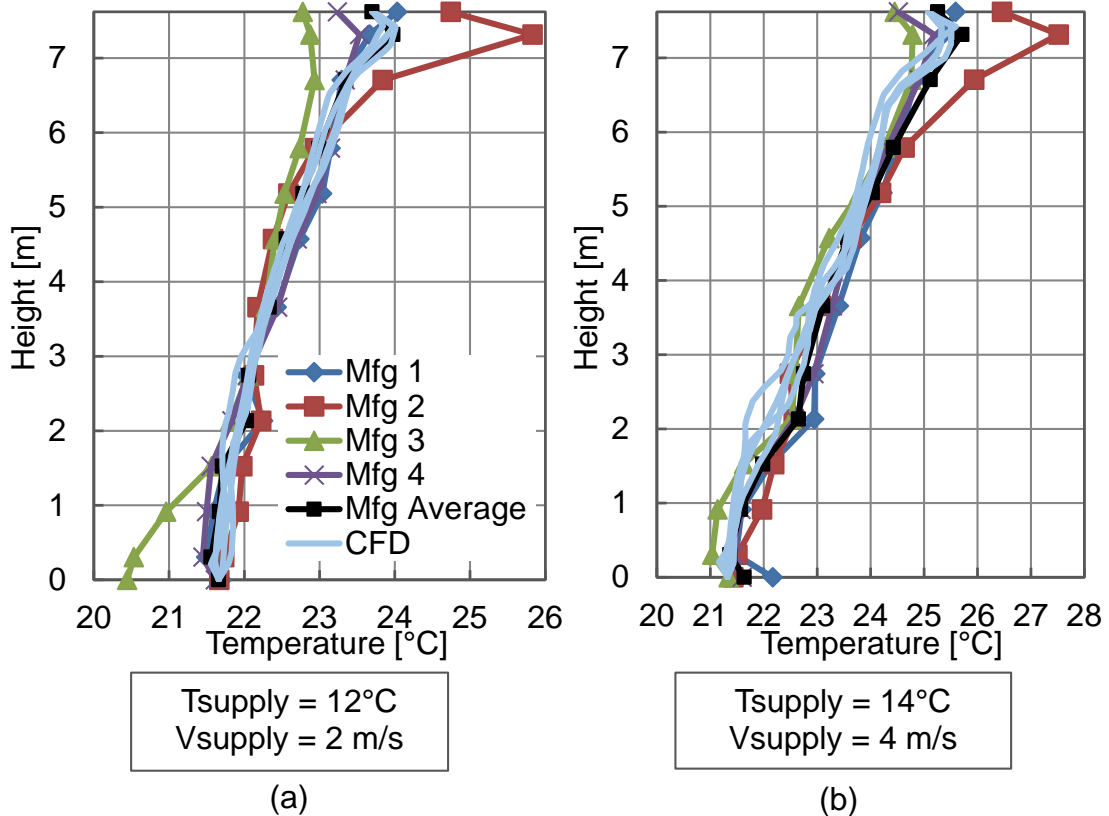
Figure 9: CFD model validation for (a) baseline and (b) XChanger cases

"Thermal comfort is the condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation" [3]. Experiments were conducted in the 1970s to create a probability distribution for the occupants' satisfaction under various thermal conditions. The HVAC and building science industry uses an adaptation of these original experiments, as specified in ASHRAE Standard 55 [3].

Thermal comfort is calculated from six inputs: temperature, velocity, humidity, radiant temperature, metabolic rate, and clothing insulation. By means of a heat balance, these six inputs produce two outputs: the particular mean vote (PMV) and the percent people dissatisfied (PPD). The PMV is a nominal scale consisting of -3, -2, -1, 0, 1, 2, 3. For example, a PMV of -3 means the occupant is very cold, a PMV of 0 means the occupant is neutral, and a PMV of 1 means the occupant is slightly warm. A PPD of 50% means that, on average, 50% of the occupants would be dissatisfied with their surroundings. The lowest value that PPD can obtain is 10% - meaning that 10% of occupants will always be dissatisfied. Thermal comfort is investigated later in this report.

4   Utility Bill Summary

Monthly utility bills for the test facility are summarized in this section. Each billing period begins near the 20th of each month and lasts for approximately 30 days. Over each 30 day period, electricity consumption, electricity demand, and gas consumption are provided and plotted in Figure 10. Vertical dashed lines indicate the date the energy saving technology was implemented in the test facility. The red box indicates a period

when the utility provider estimated the electricity use, rather than reading the meter. The utility provider's time-of-use pricing definitions are defined in Table 2. Additionally, the utility provider reports daily electricity consumption by using a watt-meter. Daily power consumption is plotted in Figure 11 with the exception of one week in June when the utility provider reported a communication error.

Table 2: Utility Provider's Time-of-Use Pricing Definitions

Utility Provider's Time-of-Use Pricing Definitions		
Peak	Summer	Weekdays 10 am - 8 pm
	Winter	Weekdays 7 am - 11 am and 5 pm - 9 pm
Intermediate	Summer	Weekdays 7 am - 10 am and 8 pm - 11 pm
	Winter	Weekdays 11 am - 5 pm
Off-Peak	Summer	Weekday 11 pm - 7 am
	Winter	Weekdays 9 pm - 7am, Weekends
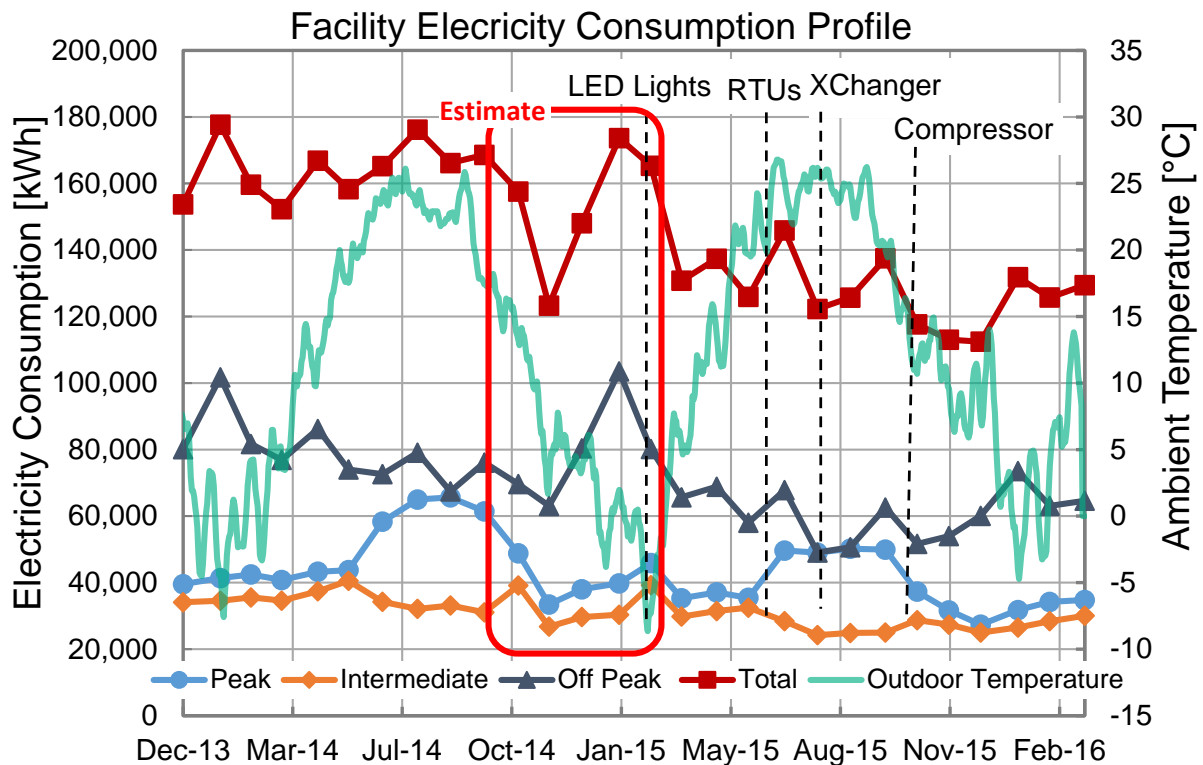
Figure 10: Facility net monthly electricity use profile

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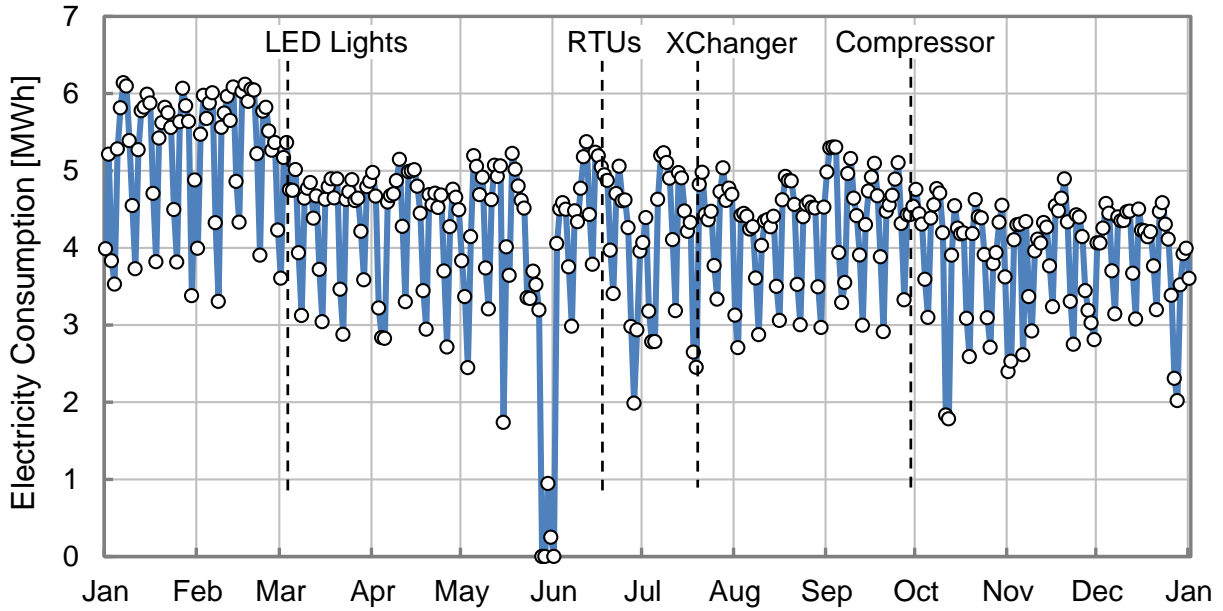
Figure 11: Facility net daily electricity use for 2015
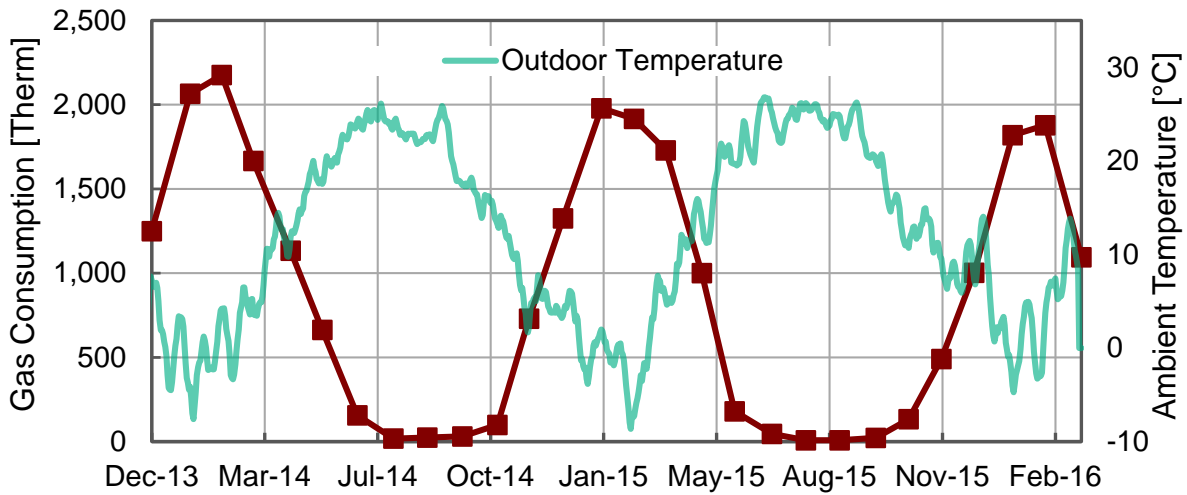
Figure 12: Facility gas consumption profile

Monthly total electricity consumption for the entire facility from 2014 through 2015 is shown in Table 3, below. It is shown that the energy conservation measures, including a lighting retrofit, rooftop-unit retrofit, new air compressor, and XChanger installation, have saved the facility an average of 15.9% electricity each month. The months displayed here are actually billing periods. For example, January really means December 19 – January 22. These savings do not take into account the effect of outdoor air temperature variation between years. Degree days are a reflection of outdoor air temperature, where more heating/cooling degree days means more required space heating/cooling, respectively.

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Table 3: Electricity consumption and savings (in MWh)

	Jan.	Feb.	Mar.	Apr.	May.	Jun.	Jul.	Aug.	Sep.	Oct.	Nov.	Dec.	Total
2014	177.7	159.7	152.3	166.9	158.3	165.2	176.2	166.1	168.6	157.6	123.3	148.1	1915.7
2015	173.6	165.3	130.8	137.4	125.9	145.9	122.3	125.7	137.5	117.7	113.0	112.4	1611.5
Difference	4.0	-5.7	21.5	29.5	32.4	19.3	53.9	40.5	31.1	39.9	10.2	35.6	304.2
Reduction	2.3%	-3.6%	14.1%	17.7%	20.5%	11.7%	30.6%	24.3%	18.5%	25.3%	8.3%	24.1%	15.9%
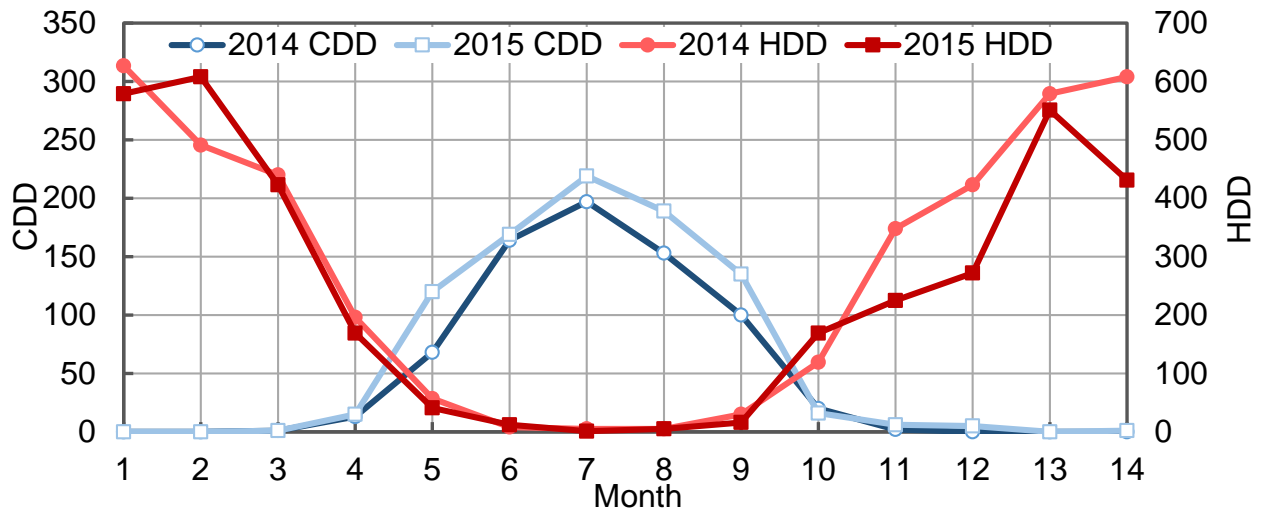
Note: Month 13 and 14 denote January and February of the following year

Figure 13: Monthly degree days (base 18 °C)

Since 2015 had 11.2% more cooling degree days, the cooling load in the facility was higher in 2015 than in 2014. Therefore, under the same weather conditions, the energy savings from the XChanger would be even higher. Table 4 shows the number of degree days in each billing period and the facility's monthly energy savings. It is clear that the facility saves energy despite the elevated temperature in summer 2015. One would expect that, all else held constant, energy savings would decrease as the percent difference in CDD between 2014 and 2015 increases. This is illustrated in the period of 6/20-7/22, where 2015 was slightly cooler than 2014. A slight linear relationship was found between the percent difference in CDD and energy savings ($R^2 = 0.32$), implying that the energy savings are weakly dependent on degree day differences.

Table 4: Energy savings and cooling degree days

2014 - 2015		2014 CDD	2015 CDD	CDD Percent Difference	Savings
Begin	End				
1/23	2/20	0.0	0.0	-	-3.6%
2/20	3/20	1.9	1.0	-	14.1%
3/20	4/22	26.3	30.4	15.6%	17.7%
4/22	5/20	68.4	114.5	67.4%	20.5%
5/21	6/20	203.9	250.8	23.0%	11.7%
6/20	7/22	289.1	277.2	-4.1%	30.6%
7/22	8/21	232.9	287.0	23.2%	24.3%
8/21	9/21	214.8	238.4	11.0%	18.5%
9/22	10/21	70.7	64.8	-8.3%	25.3%
10/21	11/19	13.9	27.2	95.7%	8.3%
11/19	12/18	4.5	7.2	60.0%	24.1%
12/18	1/21	0.1	9.3	-	24.1%
2015 - 2016		2015 CDD	2016 CDD	CDD Percent Difference	Total
Begin	End				
1/21	2/19	0.0	0.4	-	24.0%
2/19	3/22	1.0	20.3	-	1.0%

Heating and cooling energy consumption is expected to vary with outdoor air temperature. Therefore, to make an accurate prediction of energy savings, the effect of outdoor temperature needs to be accounted for. ASHRAE Guideline 14-2002 recommends regression models that normalize total facility energy use by degree days [2]. Figure 14 shows a regression model where natural gas and electricity from the facility's utility bills are a function of degree days. Since natural gas is only used for space heating, gas consumption varies linearly with heating degree days ($R^2 = 0.969$). Electricity consumption does not vary linearly with ambient temperature ($R^2 = 0.019$). This correlation cannot account for differences in latent load, plug load, or the fact that the facility has small amounts of electric heating in the winter. This same lack of correlation was observed for daily data as well as the monthly data shown here.
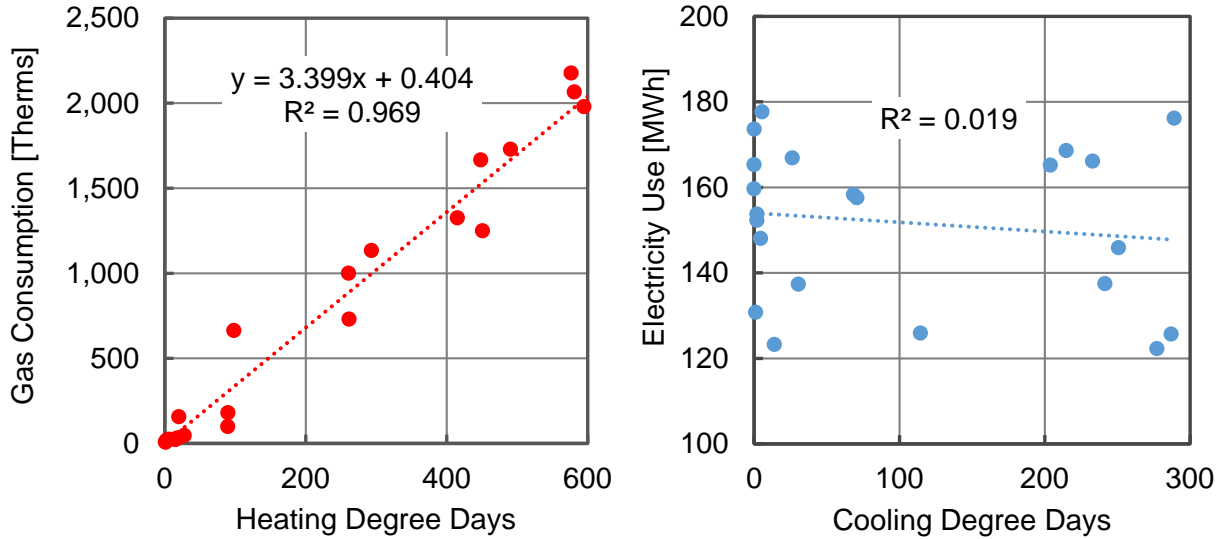
Figure 14: Degree day normalization

5 Effect of component replacement

5.1 Lighting Replacement

The number of shop lights used in the factory is summarized in Table 5. The shop lights were replaced over a period from February 22, 2015 to March 15, 2015. The old lights, 400 W metal halide lights, were replaced by 190 W LED light fixtures. The metal halide light fixtures require 58 W for the ballast, so the entire fixture consumes 458 W.

The electricity use of a single light can be calculated by Equation 1. From Table 6, the new lights consume 150,248 kWh less electricity per year. However, this calculation does not take into account the additional cooling load caused by the heat generation of the light fixtures. Since the facility has few windows and no ceiling exhaust, each light fixture outputs heat equal to the power it consumes as electricity. This extra heat adds to the cooling load and the time that the RTUs and fans must run. Therefore, the net electricity use seen by the facility during cooling season as a result of the lighting retrofit is given by Equation 2.

$$E_{light} = Ptf \qquad (1)$$

where:

E_{light} = light's electricity use [kWh]
P = light's rated power consumption [kW]
t = time period of interest [h]
f = Use Factor, average value of light schedule

Table 5: Number of lights in each space

Room	Large Mfg.	Small Mfg.	Storage	Assembly	Total
Number of Lights	33	25	21	66	145

Table 6: Electricity consumption of lights

Light Fixture	Power [W]	Use Factor	Number of Lights	Energy per Bulb [kWh/bulb/year]	Energy per 30 Days [kWh]	Energy per Year [kWh]
Old	458	0.41	145	1,771	21,104	256,767
New	190			735	8,755	106,519

$$E_{facility} = E_{light} + \frac{E_{light}}{COP} \qquad (2)$$

where:

$E_{facility}$ = facility's extra electricity use due to lighting [kWh]

COP = coefficient of performance for vapor compression cycle [-]

Table 7 shows the electricity consumption of the entire facility due to the lights, as calculated using Equation 2. Fan electricity and heat generation have been neglected in this calculation. Assuming the facility is in constant cooling (so that (2) is valid), the facility will save 192,748 kWh per year by retrofitting from metal halide lights to LEDs. In the winter time, the reduced heat load of the LED lights will have to be supplied by extra natural gas heating.

Table 7: Cooling season net electricity consumption due to lights

Light Fixture	Power [W]	Use Factor	Number of Lights	Average RTU COP	Facility Yearly Electricity Consumption due to Lighting [kWh]
Old	458	0.44	145	3.44	331,409
New	190				137,484
Summer Monthly Energy Savings					16,160

5.2 Air Compressor Replacement

The test facility uses an industrial air compressor to supply compressed air for manufacturing purposes. This compressor was replaced on October 14, 2015 with a unit that uses 10 hp less electricity (7.4 kW). The efficiency of the old compressor was not listed, so its heat output was calculated assuming the same efficiency as the new compressor. The energy consumption of industrial compressors using load/unload-type control can be calculated in the manner provided by the Department of Energy [4]. In the 35 days since installation, the air compressor was fully-loaded for 263 hours, or 31.3% of the time. To make the energy consumption calculation possible, it is assumed that the unit operates either fully-loaded or fully-unloaded, the unloaded capacity is 25% fully-loaded capacity, and the old compressor run-time was the same as the new compressor. The result of this calculation is time averaged electricity consumption.

Since the compressor is air-cooled with space air, but the space air is exhausted outside, the added electricity can be calculated directly without considering the cooling load. The facility's yearly electricity consumption due to the air-compressors is estimated at 189.8 MWh for the old compressor and 158.4 MWh for the new compressor. Based on this calculation procedure, the air-compressor replacement saves 31.4 MWh each year.

Table 8: Air compressor replacement

	Installation Date	Electricity [kW]	Heat Output [kW]	Time Averaged Electricity [kW]	Monthly Electricity [kWh]	Facility Yearly Electricity Consumption due to Compressors [kWh]
Old	2/14/2008	44.7	42.2	21.7	15,601	189,815
New	10/14/2015	37.3	35.2	18.1	13,018	158,391
				Monthly Energy Savings		2,619

5.3 Rooftop Unit Replacement

The RTUs were retrofitted according to Table 9. The units serving the manufacturing rooms, the rooms with the highest cooling demand, had an average COP improvement from 2.56 to 3.58. The entire facility average COP improved from 2.92 to 3.44. An additional 151 kW of gas heating capacity and 35 kW of cooling capacity were added to the facility. Upgrading the RTUs also introduces two stages of cooling through the use of dual compressors. Both old and new units use air-side economizers with enthalpy control. It is difficult to calculate savings due to the RTU retrofit, so the savings are simulated.

Table 9: Test facility RTU summary

Test Facility RTU Summary [NEW] [OLD] [UNCHANGED]

Unit	Manufacturer	Cooling [Tons]	Heating [Btu/h]	EER	COP	Description	Area Served	Manufactured	Refrigerant	Model #	Serial #	Status
1	TRANE	30	486,000	11.0	3.22	DX, VAV	Office	Mar-2010	R-410A	YCD360B4HC2B6DD	C10001446	
2	TRANE	4	243,000	9.7	2.84	DX, CV	Mfg 2	Sep, 2005	R-22	YHC048A		
3	TRANE	12.5	250,000	12.1	3.55	DX, CV	Mfg 1	May, 2015	R-410A	YHD150G4RHA01H00	152010096D	
	York		161,500	8.75	2.56			May-1997	R-22	D4CG150N 16546JSC	NEFM058047	
4	TRANE	12.5	250,000	12.1	3.55	DX, CV	Mfg 1	May, 2015	R-410A	YHD150G4RHA01H00	152010096D	
	York		161,500	8.75	2.56			May-1997	R-22	D4CG150N 16546JSC	NEFM058043	
5	TRANE	10	161,500	12.4	3.63	DX, CV	Mfg 2	May, 2015	R-410A	YHC120E4RMA11H00	151311104L	
	York			8.75	2.56			May-1997	R-22	D3CG120N 16546JSD	NEFM064821	
6	TRANE	12.5	250,000	12.1	3.55	DX, CV	Mfg 1	May, 2015	R-410A	YHD150G4RHA01H00	152010096D	
	York		161,500	8.75	2.56			May-1997	R-22	D4CG150N 16546JSC	NEFM067221	
7	TRANE	15	300,000	12.1	3.55	DX, CV	Mfg 1	May, 2015	R-410A	YHD180G4RLA01H00	15210139D	
	York			8.75	2.56			Jun-1997	R-22	D2CG180N 24046ECE	NFFM067221	
8	TRANE	10	161,500	12.4	3.63	DX, CV	Mfg 2	May, 2015	R-410A	YHC120E4RMA11H00	151311104L	
	York			8.75	2.56			May-1997	R-22	D3CG120N 165461SD	NEFM064803	
9	TRANE	12.5	250,000	12.1	3.55	DX, CV	Storage	May, 2015	R-410A	YHD150G4RHA01H00	152010096D	
	York		161,500	8.75	2.56			Apr-1997	R-22	D4CG150N 16546JSC	NDFM045467	
10	TRANE	8.5	129,400	12.5	3.66	DX, CV	Machine	May, 2015	R-410A	TYSC102F4EMA001SC	-	
	York			8.75	2.56			May-1997	R-22	D3CG102N 130461SF	NEFM064209	
11	TRANE	10	161,500	12.1	3.55	DX, CV	Machine	May, 2015	R-410A	YHD150G4RHA01H00	152010096D	
	York			8.75	2.56			Apr-1997	R-22	D4CG150N 16546JSC	NDFM045467	
12	TRANE	30	283,500	11.0	3.22	DX, VAV	Office	Sep-2005	R-22	YCD330A4LR2A6DE4	CO5H07554	
3W	TRANE	6	64,000	11.5	3.37	DX, CV	Assembly	Sep-2005	R-22	YHC072A4RLA1R	5361026081	
4W	TRANE	6	64,000	11.5	3.37	DX, CV	Assembly	Sep-2005	R-22	YHC072A4RLA1R	5361028351	
5W	TRANE	6	64,000	11.5	3.37	DX, CV	Assembly	Sep-2005	R-22	YHC072A4RLA1R	5361029071	
6W	TRANE	6	64,000	11.5	3.37	DX, CV	Assembly	Sep-2005	R-22	YHC072A4RLA1R	536102761L	
7W	TRANE	6	64,000	11.5	3.37	DX, CV	Assembly	Sep-2005	R-22	YHC072A4RLA1R	536102610L	
8W	TRANE	6	64,000	11.5	3.37	DX, CV	Assembly	Sep-2005	R-22	YHC072A4RLA1R	536102981L	
13	TRANE	6	96,000	11.2	3.28	DX, CV	Storage	May, 2015	R-410A	TYSC072F4EMA001ST	-	
14	TRANE	4	65,600	13.0	3.81	DX, CV	Mfg 2	May, 2015	R-410A	TYSC048E4EMA001ST	-	

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5.4 Simulated Energy Savings of Lighting, RTU, and Compressor Retrofits

Several building energy simulations were run based on the calibrated 2015 EnergyPlus model. To simulate the energy savings of the retrofits, two models were created for each retrofit – one with the retrofit and one with the baseline system. Then, the monthly energy uses were compared. Using the RTUs as an example, a model was created with all baseline RTUs. Then, a model was created with the upgraded RTUs. Each model assumes that the retrofit considered was the only retrofit installed in the facility. The results are shown in Table 10. The percent savings are shown in parenthesis and in Figure 15 for visualization. The 'months' are actually the month in which each billing period began. Although these retrofits were not installed in the test facility for all of 2015, this simulation gives insight into how the retrofit would influence the building's energy use as if it were the only retrofit installed.

Table 10: Simulated electricity and gas savings due to retrofits

Month	Electricity [kWh]			Gas [Therms]		
	Lights	RTU	Compressor	Lights	RTU	Compressor
Jan.	13,084 (8.5%)	386 (0.3%)	2,901 (2.0%)	-182 (-9.3%)	-1 (-0.1%)	-55 (2.7%)
Feb.	13,760 (9.6%)	741 (0.6%)	2,436 (2.0%)	-133 (-9.0%)	-4 (-0.3%)	-15 (-0.8%)
Mar.	15,875 (10.9%)	1,756 (1.4%)	2,747 (2.2%)	-42 (-5.4%)	0 (0%)	-1 (-0.2%)
Apr.	15,516 (11.7%)	3,492 (3.0%)	2,455 (1.9%)	-8 (-3.5%)	6 (2.5%)	-1 (-0.7%)
May.	17,769 (11.6%)	6,553 (4.6%)	2,696 (2.0%)	-2 (-3.7%)	1 (1.6%)	0 (-1.0%)
Jun.	14,798 (10.6%)	6,825 (5.1%)	2,602 (1.9%)	-1 (-5.5%)	0 (-0.6%)	0 (-0.3%)
Jul.	16,839 (11.9%)	4,095 (3.1%)	2,640 (2.0%)	-1 (-3.5%)	-1 (-3.8%)	0 (-1.3%)
Aug.	16,637 (11.1%)	3,713 (2.7%)	2,668 (2.1%)	-4 (-12.7%)	-1 (-2.0%)	1 (3.1%)
Sep.	15,189 (11.8%)	1,564 (1.3%)	2,496 (2.0%)	-23 (-5.5%)	-1 (-0.1%)	0 (0%)
Oct.	14,673 (12.1%)	771 (0.7%)	2,506 (2.2%)	-39 (-6.8%)	0 (0%)	0 (0%)
Nov.	14,383 (11.5%)	868 (0.8%)	2,544 (2.3%)	-65 (-6.6%)	-1 (-0.1%)	0 (0%)
Dec.	13,734 (10.0%)	627 (0.5%)	2,723 (2.2%)	-123 (-7.9%)	-1 (-0.1%)	-27 (-1.4%)
Total	182,258 (11.0%)	31,391 (2.1%)	31,415 (2.1%)	-620 (-5.5%)	-3 (-0.3%)	-99 (-0.3%)
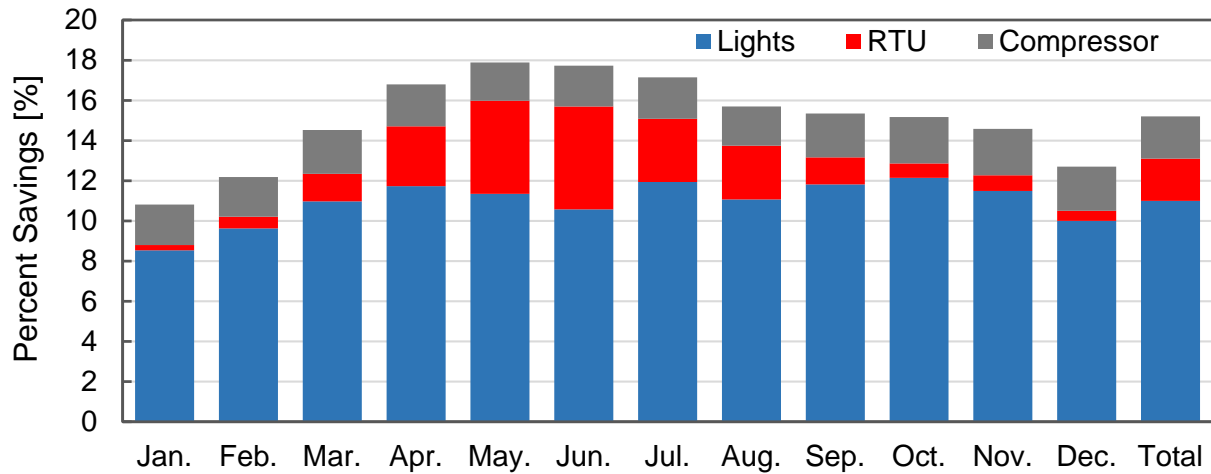
Figure 15: Simulated percent electricity savings of retrofits

The simulated lighting electricity savings, 182 MWh, were very close to the calculated savings using a hand calculation, 193 MWh. The compressor retrofit electricity savings is exactly the calculated value, 31.4 MWh. The RTUs are expected to save 6% electricity during the summer and 0.3% during winter. During winter months, the facility will use about 7-9% more gas due to the lighting retrofit.

6   Effects of XChanger

6.1   XChanger Energy Savings

Based on the baseline (Jan 2014 – July 2015) and XChanger (July 2015 – March 2016) utility bills and the analysis of other retrofits, the XChanger summer energy savings can be calculated as:

$$XChanger\ Energy\ Savings = E_{2014} - E_{2015} - \Delta E_{lights} - \Delta E_{RTU} - \Delta E_{Compressor} \qquad \textbf{(3)}$$

where E is the measured or simulated electricity consumption.

The simulated values of the percent energy saved each month is used here because it was close to the calculated values. The remaining energy savings are assumed to come from XChanger, which was installed during July.

XChanger provided the facility an average monthly savings of 6.4% electricity. Of course, XChanger was only installed in three of five rooms and this analysis does not take into account weather conditions. For example, the period of 10/21-11/19 had twice as many cooling degree days in 2015 compared to 2014 and therefore diminished net energy savings (8.3%). Therefore, the calculated XChanger energy savings were small, even negative. The fluctuations in monthly XChanger energy savings are caused by weather conditions, plug loads, and other uncontrolled factors that influence a building's energy consumption.

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Table 11: XChanger electricity savings

Utility Bills					Electricity Savings due to:			
Start	Stop	2014 [MWh]	2015 [MWh]	Energy Savings	Lights	RTU	Compressor	XChanger
1/23	2/20	160	165	-3.6%	-	-	-	-
2/20	3/20	152	131	14.1%	9.6%	-	-	-
3/20	4/22	167	137	17.7%	11.0%	-	-	-
4/22	5/20	158	126	20.5%	11.7%	-	-	-
5/21	6/20	165	146	11.7%	11.4%	-	-	-
6/20	7/22	176	122	30.6%	10.6%	5.1%	-	14.9%
7/22	8/21	166	126	24.3%	11.9%	3.2%	-	9.3%
8/21	9/21	169	137	18.5%	11.1%	2.7%	-	4.7%
9/22	10/21	158	118	25.3%	11.8%	1.4%	-	12.1%
10/21	11/19	123	113	8.3%	12.1%	0.7%	2.3%	-6.9%
11/19	12/18	148	112	24.1%	11.5%	0.8%	2.3%	9.5%
12/18	1/21	174	132	24.1%	11.0%	0.3%	2.0%	10.8%
Start	Stop	2015 [MWh]	2016 [MWh]	Energy Savings	Lights	RTU	Compressor	XChanger
1/21	2/19	165	126	24.0%	10.5%	0.6%	2.0%	10.9%
2/19	3/22	131	129	1.0%	10.2%	0.6%	1.9%	-11.6%

The lighting energy savings appear to be an underestimate because the savings of the lighting do not account for all observed energy savings during 2/20-6/20. The power consumption, number, and schedule of lighting are fully known and accurately simulated. Therefore, it is unclear why such a large difference exists.

The gas used for space heating correlates very well with heating degree days. Therefore, it makes sense to compare gas usage per heating degree day. However, to compare the gas used in 2015 against 2014, the heat load of the lighting and compressor retrofits must be added back because the lower electrical loads seen in 2015 mean that the facility needs more gas for space heating. Using the validated 2015 EnergyPlus model, it was found how much heat would have been added to the space had each retrofit not occurred. This heat was converted to a "therm equivalent" by converting Joules (EnergyPlus output) to therms, then dividing by the efficiency of the gas furnace, 0.81. Since the heat output difference due to the compressor retrofit is less than six therms equivalent for each period, it was ignored for simplicity. Summer months were included for completion, but the results are omitted because a negative therms-per-degree-day was calculated.

One source of error in this analysis is that 'adding back' the heat due to the lighting retrofit is correct only when the facility is in heating operation during the entire period considered. This is the reason that the shoulder seasons and summer had negative therms-per-degree-day in 2015. This is also the reason that the calculated 'Difference' increases from 1/23/14 to 3/20/14. A better approach would include the fraction of time the facility spent in heating. However, this is not readily available from the experiment or as an EnergyPlus output variable.

The XChanger was switched from cooling operation (supply low, return high), to heating operation (supply high, return low) on January 4, 2016. Therefore, half the period from 12/18/15 to 1/21/16 and the whole period from 1/21/16 to 2/19/16 are direct comparisons between baseline and XChanger systems. During these periods, the XChanger saved 0.7 therms per heating degree day. This is a savings of 22.3% gas. However, this analysis is clearly subject to large uncertainties, as shown in the first row when no retrofits were installed, yet the facility observed a difference in therms-per-degree-day of 0.58.

Table 12: Heating energy savings

| Start	Stop	2014 Therms	2015 Therms	2015 Adjusted	HDD 2014	HDD 2015	2014 Therms / HDD	2015 Therms / HDD	Difference
1/23	2/20	2,176	1,916	-	576.6	600.3	3.77	3.19	0.58
2/20	3/20	1,666	1,729	1149	448.5	490.7	3.71	2.34	1.4
3/20	4/22	1,133	1,000	331	293.4	266.9	3.86	1.24	2.6
4/22	5/20	663	180	-474	98.1	86.8	6.76	-	-
5/21	6/20	156	46	-703	20.1	26.8	7.76	-	-
6/20	7/22	18	8	-616	2.5	1.1	7.20	-	-
7/22	8/21	24	8	-702	6	1.7	4.00	-	-
8/21	9/21	31	22	-679	17.7	14.5	1.75	-	-
9/22	10/21	99	133	-507	89.6	119.5	1.10	-	-
10/21	11/19	730	490	-128	261.5	166.5	2.79	-	-
11/19	12/18	1,325	1,002	396	415.2	294.4	3.19	1.34	1.9
12/18	1/21	1,979	1,819	1268	594.8	475.7	3.33	2.66	0.7
Start	Stop	2015 Therms	2016 Therms	2016 Adjusted	HDD 2015	HDD 2016	2015 Therms / HDD	2016 Therms / HDD	Difference
1/21	2/19	1,916	1,878	1327	600.3	535.4	3.19	2.48	0.7
2/19	3/22	1,729	1,094	579.8	514	490.7	3.52	1.63	1.89

6.2 HVAC Energy Use

Since the characteristics of this particular facility may differ from other commercial buildings, it is beneficial to view the energy savings as a percentage of total HVAC energy use. A watt-meter was placed on one electric panel to measure the power consumption of a collection of rooftop-units serving both manufacturing rooms and the storage room (where XChanger was installed). The office and assembly spaces were not instrumented. 52.4% of installed cooling capacity was instrumented starting on July 31. Between July 31 and September 24 (roughly the end of cooling season), the instrumented RTUs consumed 41,089 kWh. Assuming that the cooling load is distributed evenly throughout the facility, the entire facility will use 78,415 kWh over the same period (41,089/0.524). Based on Table 11, the XChanger system saved the facility 7.0% energy in August and September. Since the entire facility consumed 237,592 kWh during this period, the XChanger system saved the facility 16,631 kWh (237,592*0.07).

Had the storage and manufacturing rooms not been installed with XChanger, they would have used 41,089 + 16,631 kWh, or 57,720 kWh. Compared to what they actually used, 41,089 kWh, XChanger saved these rooms 28.8% HVAC electricity. Implicit in this

figure is the assumption that the cooling load is distributed evenly throughout the space. This assumption is necessary to estimate the energy consumption of the RTUs in the office and assembly spaced.

Based on this analysis of the utility bills, the XChanger system is expected to save 6.4% total electricity, which is 28.8% of HVAC electricity during cooling season. During heating season, XChanger is expected to save 21.2% gas during heating season, though the analysis for heating season had large uncertainties due to the influence of the other retrofits.
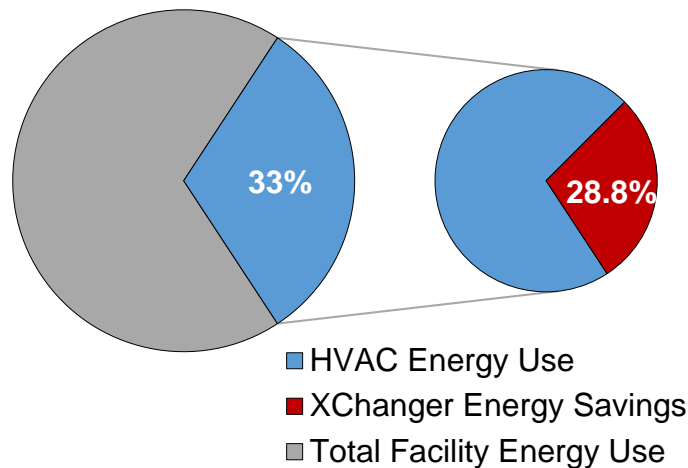
Figure 16: XChanger energy savings as a percentage of HVAC Energy

6.3   Temperature Profiles

Temperature profiles were measured from floor to ceiling in ten locations across four rooms in the test facility. On each measurement pole, there are 12 thermocouples measuring air temperature, one thermocouple measuring ceiling temperature, and one thermocouple measuring floor temperature. Each thermocouple has an uncertainty of 0.5 °C. To see trends across different seasons, each thermocouple value was averaged over the entire season and shown in Figure 17, where baseline is defined as the time before XChanger's installation. Figure 18 shows the average temperature profile by room.

During the summer, the manufacturing rooms have almost continuous cooling and therefore realized the greatest change under the XChanger retrofit. In the large manufacturing room, XChanger maintained the same temperature in the occupied zone while increasing the stratification in the upper zone by 1.8 K. In the small manufacturing room, the temperature in the occupied zone decreased while the temperature in the upper portion of the space increased. Since the storage room has a low cooling load, the RTUs are not on very often. Therefore, the storage room had no significant change in temperature profile during cooling season.

During the winter, however, the storage room realized the greatest benefit from XChanger. The baseline winter had the greatest observed stratification in the building – 7.1 K. Supplying air high and returning air above the occupied zone reduced the temperature stratification to 3.5 K, a reduction of 50%. In particular, the sharp gradient near the ceiling was reduced. The manufacturing rooms saw a negligible difference during heating season. One possible explanation for this is that the manufacturing rooms were

in cooling for portions of the winter. One piece of evidence to justify this claim is that the XChanger winter profile resembles baseline summer (which both had high supply ducts) in their shape, especially near the ceiling.

During spring and fall, XChanger produced different effects in each room. In the large manufacturing room, the temperature below 2 m was reduced by 1 K, while the upper zone remained the same. In the storage room, the effect was the exact opposite, with 1 K stratification near the ceiling and the same temperature in the occupied zone. The small manufacturing room had no change whatsoever.
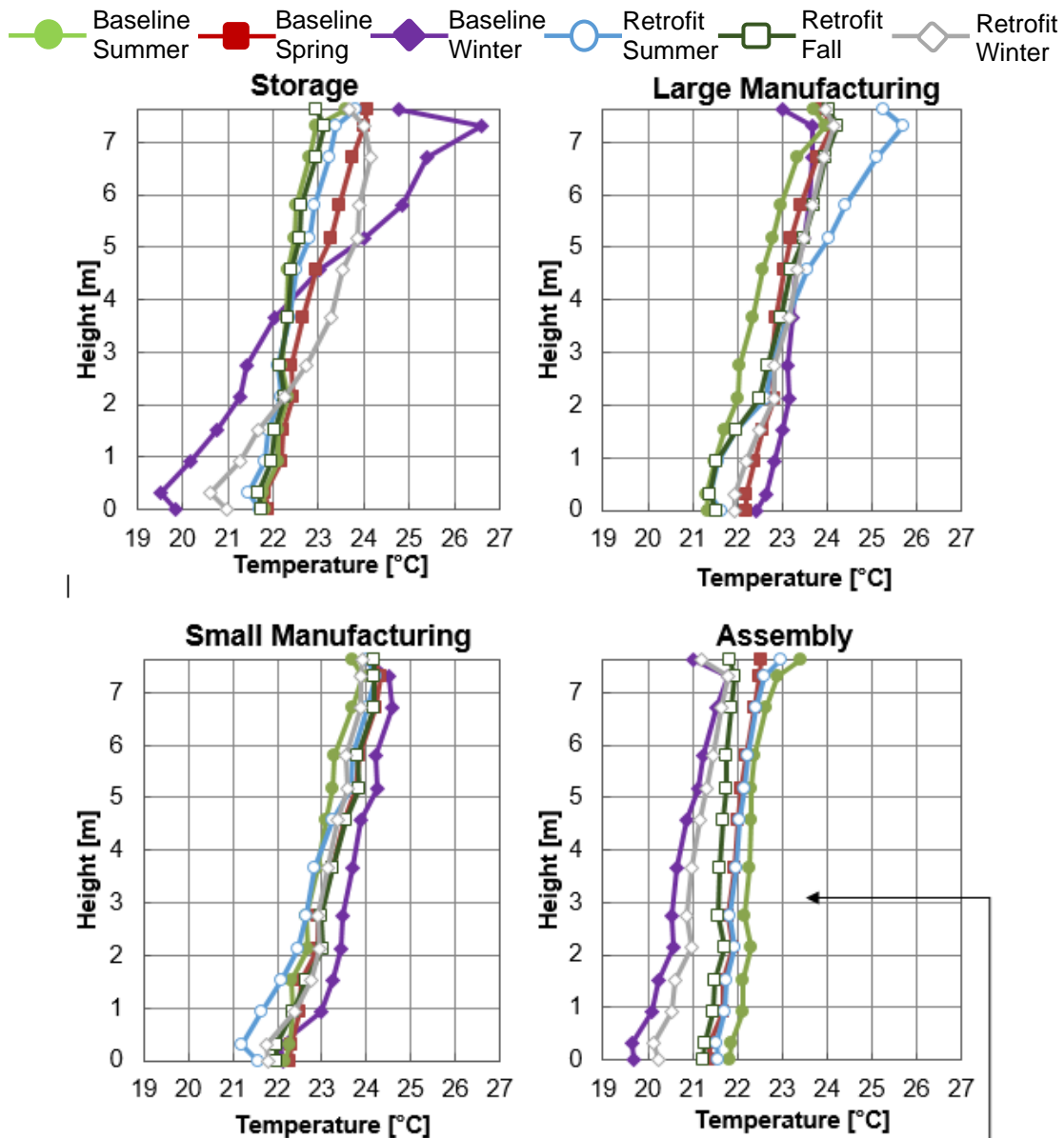

Figure 17: Temperature distribution in each space

Note: XChanger was not installed in the Assembly room. This profile is the measurement of the conventional system after the XChanger was installed in the other rooms.
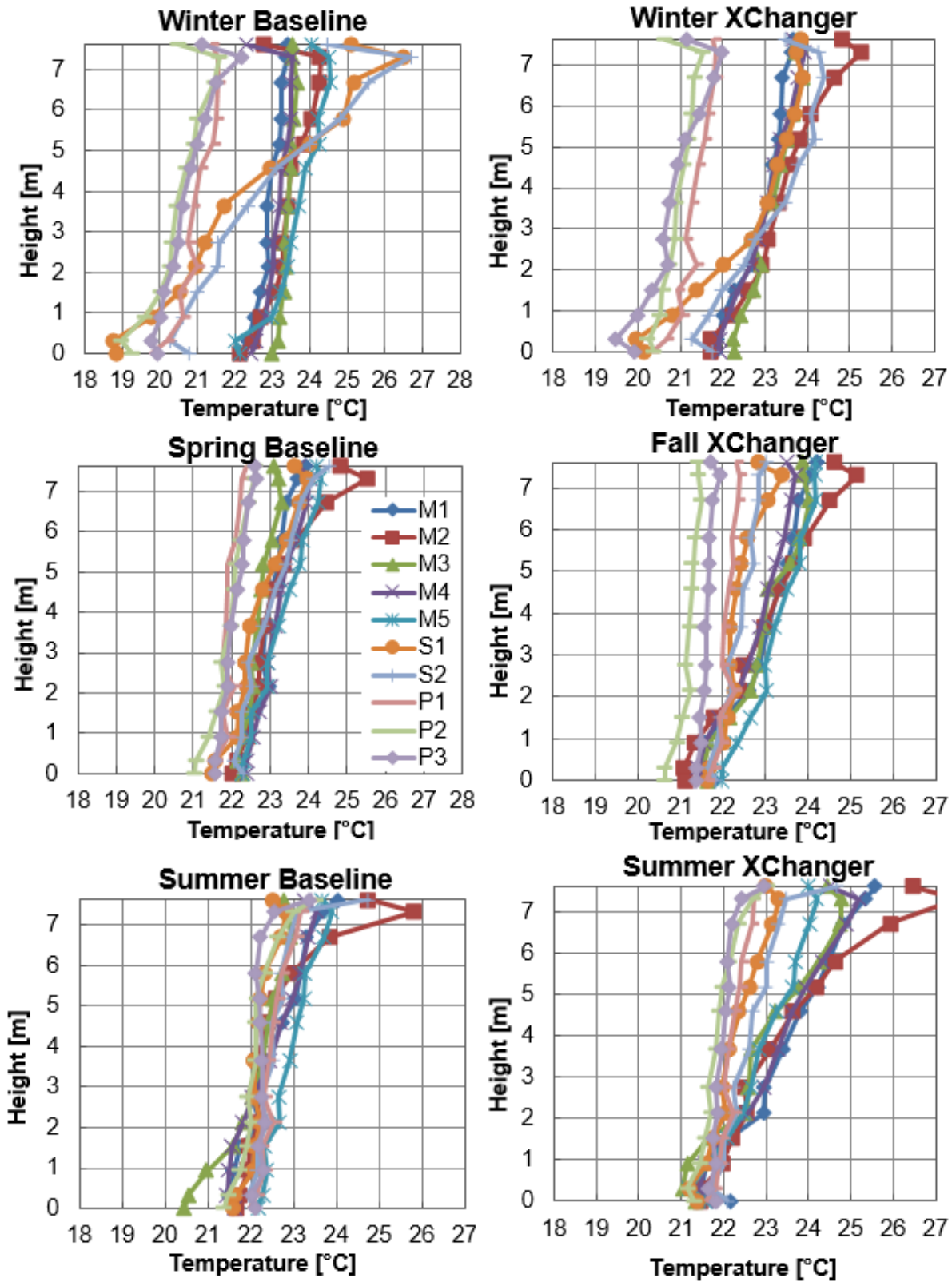
Figure 17: Temperature profile of each measurement column

6.4 Pressure Drop

The XChanger ducting will require that the RTU indoor fan operates with a higher static pressure rise. Pressure drop can be broken between static, or friction pressure, and dynamic pressure and calculated according to Equations (4-5). Using the friction pressure-loss diagram in McQuiston et al. (2004), a 0.45 m^2 duct providing 2.36 m^3/s (5,000 CFM) produces a 4.5 Pa pressure drop per meter of duct [5].

Table 13: Fan static pressure drop based on RTU specifications

RTU Fan Static Pressure Drop (Pa)	
Filter	7.46
Economizer (10% outdoor air)	6.71
Baseline Ducting (2 m)	9
XChanger Ducting (8 m)	36
Loss Coefficients, K (dimensionless)	
90° Elbow (un-vaned)	1.2
90° Elbow (vaned)	0.33
Supply Grille	0.36

$$P_v = \frac{1}{2}\rho v^2 = 77.76 \ Pa \qquad \textbf{(4)}$$

$$P_{loss} = P_{static} + \sum K * P_v \qquad \textbf{(5)}$$

As shown in Table 13, the baseline ducting contains one un-vaned bend, while the XChanger contains an average of three vaned bends and one un-vaned bends. Table 14 shows calculated total pressure drop of baseline and XChanger cases. Based on the manufacturer's specification for the average 12.5 ton RTUs used at the test facility, each fan with an XChanger unit will use nominally 1.47 kW while the baseline fans will use 0.78 kW. This is important for the building energy simulation results, as shown later.

Table 14: Pressure drop in ducts

ΔP	Baseline [Pa]	XChanger [Pa]
Static	23.2	50.2
Dynamic	121.3	198.2
Total	144.5	248.4

7   CFD Simulation Results

Five cases were simulated to study the effect of duct configuration on stratification and thermal comfort. The baseline and XChanger cases are simulated as installed in the test facility. These were validated using measured temperature profiles to within measurement uncertainty. Case 1 is how XChanger Co. would install units when there is no obstructions on the floor – supply near the floor and return just above the occupied zone. Case 2 has the supply duct at the same height as the installed XChanger, but the

return duct is located at two-thirds height (floor to ceiling), instead of one-third height. Case 3 investigates the effect of doubling the diffuser face area and decreasing the velocity by half (to maintain the same flow rate as the other cases).

Table 15: Parametric study test matrix (for cooling only)

Case	Description	Supply Height [m]	Return Height [m]
Baseline	Overhead, mixing ventilation	7.0	7.0
XChanger	As installed	1.8	2.4
Case 1	"Preferred" installation	0.46	2.4
Case 2	Higher Return	1.8	5.5
Case 3	Double face area	0.46	2.4

Increasing the return duct height in Case 2 compared to the XChanger case has the effect of reducing the average temperature in the space, reducing stratification (particularly in the upper zone), and increasing the return air temperature. Case 1, supplying at floor-level and returning at the same height as the installed XChanger, shows an average temperature reduction of 0.4 K, a reduction in stratification of 0.8 K, and an increase in return air temperature comparable to Case 1. Supplying air near the floor while increasing the supply diffuser area has the effect of greatly increasing stratification and slightly increasing average air temperature.

Table 16: Parametric study results for the entire room

Whole Volume Averages						
Case	Temp. [ºC]	Velocity [m/s]	PMV [-]	PPD [%]	RAT [ºC]	Stratification [K]
Baseline	22.5	0.11	0.6	14.2	22.0	2.3
XChanger	23.2	0.12	0.7	18.8	21.8	4.5
Case 1	22.8	0.11	0.7	16.7	22.3	3.7
Case 2	22.1	0.13	0.5	12.8	22.6	3.9
Case 3	23.1	0.07	0.7	20.9	22.6	6.3

It would be expected that Case 2 would perform better than the XChanger case (as-installed) in terms of increasing the average temperature and reducing the return air temperature. In this simulation, however, the opposite is observed. The reason for this is that, when air is supplied near the floor, the jet of cold supply air traveled further along the floor, hit the wall and one CNC machine, and traveled up the wall/machine to mix with the air in the upper zone. This phenomenon is due in part to the Coanda effect, where a jet 'attaches' to a surface, preserving its momentum for longer distances. If obstructions are placed in the way of the jet of supply air (like a desk, cabinet, etc.), then the jet will hit this object and spread into the room and cause unnecessary mixing.

Energy savings during cooling operation can generally be achieved by increasing the average air temperature of the space, which decreases heat gain through the building envelope. It would be misleading to simply point out which case in Table 16 saves the most energy based on this criteria because these simulations have no concept of thermostat control. In other words, a thermostat would control the room temperature such that the temperature near the thermostat is at set point. In the simulation, the temperature near the thermostat is not controlled.

A better way to express the effect of average air temperature is the difference between mean air temperature and temperature near the location of a typical thermostat: 1.5 m height on the wall. This value expresses the increase in mean air temperature from set point that a room would expect to see under each case. Based on Table 17, Case 3 would have the least heat gain from outside due to the excessive stratification and the baseline case would have the most heat gain from outside due to more uniform vertical temperatures.

Table 17: Thermostat offset

Case	$T_{avg} - T_{thermostat}$
Baseline	0.64
XChanger	1.81
Case 1	2.00
Case 2	1.31
Case 3	3.73

According to ASHRAE Standard 55-2013, the occupied zone is the region that is 10 cm off the ground, below 1.8 m, and 0.3 m away from the walls. Table 18 shows the average values of several parameters over this region. The average PMV and PPD are within acceptable ranges for each case considered. The baseline, mixing ventilation had the most dissatisfied occupants, but by a negligible amount. ASHRAE Standard 55 specifies that the head-to-ankle temperature difference be less than 3 °C for seated occupants and 4 °C for standing occupants. Each case satisfied this criteria. The temperature in the occupied zone decreased as the supply duct height was decreased. For the low supply cases, it may be possible to raise the set point of the thermostat and achieve the same average temperature in the occupied zone as the baseline case. For every degree a thermostat is raised above 72 °F, the building will use 1-3% less electricity [6].

PPD contour plots are shown in Figure 19. Twenty percent is typically the upper recommended limit for PPD. Each case meets this limit in the occupied zone. The baseline case has uniform PPD less than 20% in the entire space. Clearly, each case has PPD approaching 100% near the supply diffusers, yet acceptable PPD outside the vicinity of the diffuser. Therefore, care should be taken during installation to aim the diffuser away from occupants. It was discussed previously that Case 1, supplying air near the floor, actually reduced the stratification and the average temperature because the air mixed more than the XChanger case. This is visible in Figure 19 when compared to Case 3 or the XChanger case.

Table 18: Parametric study results for the occupied zone

Case	Temperature [ºC]	Velocity [m/s]	PMV [-]	PPD [%]	Head-to-Ankle [ºC]
			Occupied Zone		
Baseline	21.8	0.13	0.49	10.45	0.3
XChanger	21.5	0.20	0.39	9.12	0.4
Case 1	20.9	0.24	0.26	8.42	0.9
Case 2	20.9	0.20	0.27	7.40	0.2
Case 3	19.4	0.18	0.00	7.20	1.5
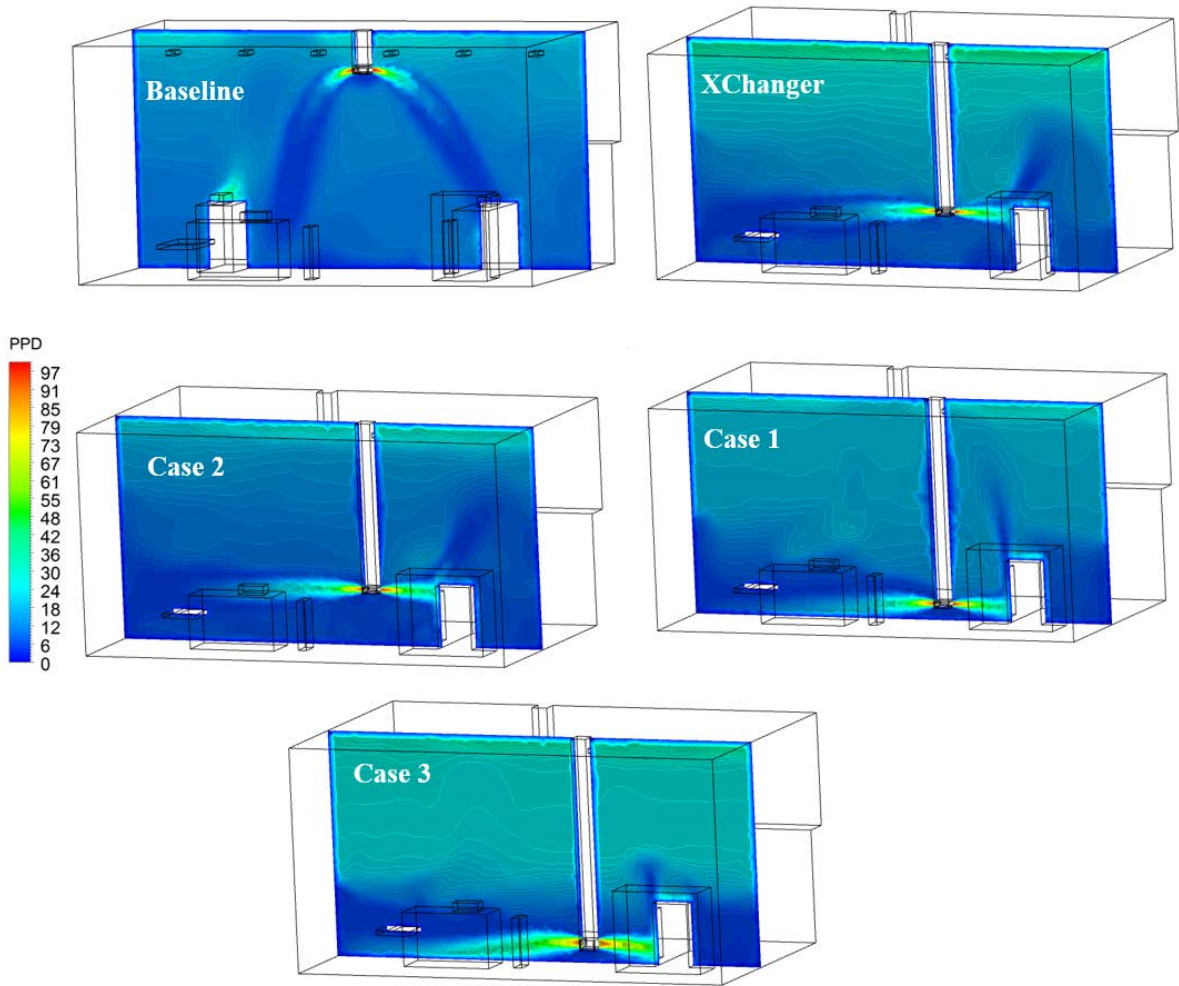
Figure 18: PPD contour plots

8   Building Energy Simulation Results

An EnergyPlus model was created for 2015 that contains each building retrofit – lighting, RTUs, XChanger, and the new air compressor. This model has been calibrated using daily site electricity use and monthly gas use, though the accuracy for gas consumption is less than ASHRAE Guideline 14 recommends.

Two models were created based on the calibrated 2015 model – one with all measured baseline temperature profiles and one with all measured XChanger temperature profiles. The temperature profile used for each time period in each model is shown in Table 19. The names of the temperature profiles correspond to the legend in Figure 17. To calculate the energy savings due to XChanger, the energy use per month or per day can be computed for each model and compared.

Table 19: Temperature profile corresponding to model and time period

Period	Mixing ventilation model	XChanger model
1/1 – 3/23	Baseline Winter	XChanger Winter
3/24 – 5/31	Baseline Spring	XChanger Fall
6/1 – 9/22	Baseline Summer	XChanger Summer
9/23 – 11/31	Baseline Spring	XChanger Fall
12/1 – 12/31	Baseline Winter	XChanger Winter

To show the sensitivity of the building's energy use to fan power consumption, two models were created – one with equal pressure drop between baseline and XChanger and another with the calculated pressure drop from section 6.4. When fan power consumption is the same in baseline and XChanger models, stratification alone saves 37.4% electricity and 7.7% gas. In other words, these savings are realized by the room air modeling temperature profiles, return air offset, and thermostat offset. When factoring fan pressure drop into the model, the savings dropped to 19.3% electricity. The gas savings increased because the added heat of the fans required less gas to be burned and should therefore not be considered a benefit. As a percentage of the total facility electricity (i.e. not just HVAC energy), Model 1 shows a 5.0% electricity savings and Model 2 shows a 2.2% electricity savings.

Table 20: Building energy simulation results

Model	Description	HVAC Electricity Savings [%]	Facility Electricity Savings [%]	Gas Savings [%]
1	$\Delta P_{Baseline}$ = 144.5 Pa, $\Delta P_{XChanger}$ = 144.5 Pa	37.4	5.0	7.7
2	$\Delta P_{Baseline}$ = 144.5 Pa, $\Delta P_{XChanger}$ = 248.4 Pa	19.3	2.2	25.2*

9   Inherent Uncertainties

There are several inherent sources of uncertainty in an energy analysis that is based on monthly utility bills. The following is a list of factors that are difficult to quantify or are assumed to have little influence.

Outdoor air temperature and humidity: The effect of outdoor air temperature on the calculated energy savings was discussed earlier in this report. Facility electricity consumption did not correlate with degree days because the building is dominated by

internal loads. Although gas consumption did correlate with degree days, the results are clouded by the removal of a significant heat load due to the lighting retrofit.

Manufacturing output: The test facility uses large, automated manufacturing equipment to process raw material and produce their products. Any variation in manufacturing electricity consumption would manifest as a variation in electric load from utility bills.

Plug loads: Like manufacturing output, the amount of electricity used for operations other than space cooling and manufacturing will vary from month to month. The number and capacity of VAV boxes in the offices is unknown for the building energy simulation.

Thermostat set point: The set point was informally inspected several times since the beginning of 2015. It varied between 21.1 °C and 22.2 °C (70-72 °F). This variation is small and can be assumed to affect the energy calculations negligibly.

10 Conclusions

A full-scale experiment was conducted in a high-bay test facility to measure the effects of a new air delivery strategy. The experimental study shows that, in cooling, supplying and returning air directly to the occupied zone produces an acceptable temperature gradient in the occupied zone, while increasing stratification in the upper portion of the space. In heating, supplying air from the ceiling vertically downward while returning air in the occupied zone reduced stratification, though not eliminating it entirely. In both heating and cooling, the effects of the installed system are more pronounced in rooms that have high cooling and heating loads.

CFD modeling reveals the effects of supply and return duct location on room air flow and stratification generation. Supplying air lower generally creates more stratification and an acceptable temperature gradient in the occupied zone. Increasing the supply diffuser face area while holding flow rate constant produces lower velocities in the occupied zone, creates large temperature gradients in the occupied zone, and produces the largest stratification observed. The room temperature gradient is less sensitive to return duct height than supply duct height, though increasing the return duct height slightly reduced stratification and the average room temperature.

Building energy simulation is used to show how stratification influences a building's energy use. A novel approach used room air modeling with 24 measured temperature profiles over one year to capture the effects of return air temperature offset, thermostat offset, and stratification. It was found that increased fan energy consumption due to the XChanger system's larger pressure drop can depreciate the electricity savings by 50%.

Based on the utility bill analysis and measured combined RTU power consumption, XChanger saves the facility 28.8% HVAC electricity over the course of one year. The EnergyPlus models bracket this result – not accounting for pressure drop resulted in 37.4% HVAC electricity savings while accounting for pressure drop resulted in 19.3% HVAC electricity savings. Building simulation results would be closer to experimental results if the fans in EnergyPlus were simulated more realistically (i.e. had pressure drop as a function of flow rate, not specified at the fan's nominal flow rate).

Gas consumption results for the experiment are inconclusive because the same therm-per-degree-day value was calculated before and after XChanger was switched to heating mode. Also, "savings" in terms of therm-per-degree-day were observed before any retrofits took place. These unexpected results are observed because the heating load

in the space changed due to the other three retrofits over the measurement period. Attempts were made to 'add back' the heat of these retrofits but this approach adds uncertainty. Building energy simulation shows that XChanger saves 7.7 – 25.2% gas, depending on fan power consumption.

11 References

[1] *Input Output Reference.* EnergyPlusTM Documentation. U.S. Department of Energy

[2] *ASHRAE Guideline 14-2002 Measurement of Energy and Demand Savings.* American Society of Heating and Refrigeration Engineers (2002).

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[5] McQuiston, Parker, and Spitler. 2004. Heating, Ventilating and Air Conditioning Analysis and Design. 6th edition. Hoboken, N.J.: Wiley.

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