# The Impact of Control Setpoints on Building Energy Use

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***Abstract*—This paper examines the impact of building Heating, Ventilation and Air Conditioning (HVAC) control system setpoints such as temperature and flow rate on total building energy requirements, for a typical system design and operation. Through the analysis focused on a summer and winter operating condition, the range of energy usage and the potential for minimizing building energy requirements by dynamically adjusting setpoints are presented in this paper.**

***Keywords-buildings; cooling; control systems; energy; heating; HVAC; optimization***

## I. Introduction

The increasing demand of air-conditioning and the energy crisis during the last decades have led to a surge of attention and there is no doubt that the improvement of the Heating, Ventilating and Air Conditioning (HVAC) control system is one of the effective solutions to realize sizable energy-saving for the building sector. The aim of HVAC control is to provide a comfortable, safe, healthy and productive environment for occupants using the least energy. Significant energy saving potential exists for building systems during operation with the help of current technology such as intelligent, adaptive or model predictive control. The development of this kind of technology has led to the possibility of the improvement of building operational performance. However, it is difficult to evaluate the potential or effectiveness of the new control strategies without first gaining a better understanding of the range of operating conditions possible for any particular building/HVAC system combination. That is, the amount of energy savings is a function of both the actions of the new control strategy and the fundamental capabilities of the HVAC system. In its most basic form, a building control system can do no more than monitor sensors, apply logic and manipulate actuators. Thus, the main objective of the work described in this paper is to clearly identify and define the space within which the building/HVAC combination is capable of operating in order to enable the determination of both energy saving potential and optimal setpoints and control logic. While this is not specifically an optimization effort, i.e. we are not seeking a single optimal solution since it is understood that setpoints and control logic may need to be adjusted on a dynamic basis, the primary metric utilized, namely total building energy usage, can be considered as an objective function.

The content is organized as follows. Section II reviews the recent studies. Section III presents the models adopted and simulation work. Section IV gives the results and Section V presents the conclusions.

## II. Literature Rievew

Simulation is taken as one of the oldest but very effective tools to engineers in every discipline. Building simulation began in the 1960s and became the hot topic of the 1970s within the energy research community. For nowadays, computer simulation is not only used for the building design stage like sizing and configuration design, but also adopted for system performance analysis more and more widely. Building simulation can be applied to reveal the inter-actions between the building itself and its occupants, HVAC systems, and the outdoor climate. A large amount of work has been done to show how important building simulation is in the study of energy performance and the design and operation of energy-efficient buildings [2]. For examples, Li et al. [8] and Pan et al. [12] analyzed and displayed the building energy break-down with calibrated models in 2007 and 2009, respectively; however, more effort is needed to understand how to obtain optimum operating parameters, particularly for building control systems. Simulation does provide a good opportunity to evaluate the dynamic and energy performance of HVAC system control strategy in a convenient and low cost way. The control strategy can also be pre-tuned before being utilized in the real system with the help of simulation. Recent research also showed performing building simulation analysis enabled diagnosis of malfunctioning or incorrectly commissioned equipment within the building and thus also assisted with future commissioning and tuning of the building performance [11].

Future development and application of information technology in the building industry will lead to a completely new building design philosophy and methodology [7]. In 2003, Mathews and Botha [9] conducted simulation with three cases and proved that simulation does indeed have the ability to improve the thermal and energy management of building HVAC systems. A lot of work has been done in the

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TABLE I. REFERENCE CHARACTERISTICS OF EQUIPMENT

| Components | Selected parameters values | | | |
|---|---|---|---|---|
| Chiller | 25000 | Capacity (W) | 2.75 | COP |
| | 44 | T_lcw (°F) | 85 | T_ecf (°F) |
| | 111.7 | V_chw (gpm) | 128.5 | V_cdw (gpm) |
| Natural Gas Boiler | 0.8 | Boiler Efficiency | 950 (Btu/lb) | Heat Value |
| Variable Volume Fan | 4500 (gpm) | Rated Flow rate | 1837 | Rated Power (W) |
| | 600 (Pa) | Pressure Rise | 0.7 | Fan Efficiency |
| Variable Speed Pump | 67.02 (gpm) | Rated Flow rate | 500 | Rated Power (W) |
| | 50 | Pump head (ft) | 0.66 | Pump Efficiency |

*T-Temperature, V-Flow Rate, lcw-leaving chilled water, ecf-entering condenser fluid, chw-chilled water, cdw-condenser water

TABLE II. DEFAULT PARAMETER VALUE FOR SIMULATION

| Variable | Value |
|---|---|
| Zone Area | $S=750\ ft^2$ |
| Overall Envelope Heat Transfer Rate | $UA = 0.3$ Btu/h-ft$^2$-°F |
| Ambient Temperature | $T_a = 90$ °F (summer condition) |
| | $T_a = 30$ °F (winter condition) |
| Ambient Pressure | P = 101 atm |
| Zone Air Temperature | $T_z = 75$ °F (summer condition) |
| | $T_z = 72$ °F (winter condition) |
| Outdoor Air fraction | $F_o = 70\%$ |
| Solar Heat Gain | $q_S = 1.5$ w/ft$^2$ (summer condition) |
| | $q_S = 0.8$ w/ft$^2$ (winter condition) |
| Lighting Heat Gain | $q_l = 1.0$ w/ft$^2$ |
| Equipment Heat Gain | $q_e = 1.0$ w/ft$^2$ |
| Occupants Heat Gain | $q_o = 1.0$ w/ft$^2$ |
| Ventilation Air Flow rate | $M_v = 1.5$ cfm/ft$^2$ |
| Infiltration Air Flow Rate | $M_i = 0.1$ cfm/ft$^2$ |
| Heat Exchanger Effectiveness | $U_1 = 75\%$ |
| Energy Recovery Effectiveness | $U_2 = 70\%$ |

field of building energy consumption simulation but more work remains to be done. Traditionally, less attention has been put on buildings operation compared with the design of a system and its construction/installation. What's more, the simulation software has been evolving steadily over recent years. HVAC component and subsystem models are now generally well understood and have been the subject of a number of researches [4]. Simulation has been extended to the use to the building operation process, although it has been traditionally regarded as a design tool.

## III. SIMULATIONS

The simulations that were conducted consisted primarily of quasi steady state determinations of hourly incremental and total building energy requirements for a range of setpoint combinations and exposed to a summer (cooling) or winter (heating) condition. In essence, a grid was established which represented a collection of setpoints, and annual building energy performance was determined for each grid point. The setpoints were constrained to maintain proper equipment operating conditions (e.g. temperature, mass flow). The primary objective of the simulations was to quantify the range of possible operating points and the maximum potential savings, assuming that the control logic could direct the HVAC system to the optimal operating conditions. Equipment performance was modeled as described below.

Total building energy was determined utilizing performance characteristics of the each component, the chiller, the cooling tower and chiller water pump and the supply air fan plus the energy input value related to lighting and other electrical equipment. The evaluation metric:

$$E_{total} = E_{lighting} + E_{equipment} + E_{chiller} + E_{pump} + E_{fan} \quad (1)$$

where:
$E_{Total}$ = total energy power density
$E_{Lighting}$ = lighting power density input
$E_{Equipment}$ = Equipment power density input
$E_{Chiller}$ = chiller power density input
$E_{Pump}$ = pump power density input
$E_{Fan}$ = fan power density input

The first two terms are specified as follows, according to ASHRAE Standard 90.1 IP [1]:
$E_{Lighting} = 1.0$ w/ft$^2$
$E_{Equipment} = 1.5$ w/ft$^2$

The system schematic is presented in Figure 1. As the diagram shows, one zone of a multiple zone Variable Air Volume (VAV) system with energy recovery ventilator was studied for this simulation analysis. For HVAC component energy consumption analysis, polynomial fits were used with representative coefficients, with the important variables being chilled water supply temperature, coil loads, chilled water flow rate, outdoor air fraction, supply airflow rate, supply air temperature and room temperature [6]. These component mathematical equation models are commonly used in similar applications. For the simulation software, Engineering Equation Solver (EES) [5] was selected because of its built–in high-accuracy thermodynamic and heat transfer parameters and capability for solving design problems in which the effects of one or more parameters must be determined. Previous research work shows that the simplicity of the models and the use of an equation solver to run the simulation ensure good robustness and full transparency [3]. Table I summarizes the model parameters.

To minimize the effect from the building itself on the simulation results, the zone is simplified as much as possible. The case that is used in this simulation is assumed to be an office zone has a dimension of 25ft ×30ft with a 9ft high ceiling. An overall envelop thermal transfer rate is given. The U value is assumed to be 0.3 Btu/h-ft$^2$-°F. The infiltration rate through the exterior walls is set at 0.1cfm/ft$^2$, which is based on information from [10]. This infiltration occurs 24 hours a day. The ventilation rate is assumed to be 1.5 cfm/ ft$^2$. For the lighting, equipment and occupants heat gain are all assumed equal to 1w/ft$^2$. Also, the effectiveness of the energy wheel is assumed to be constant throughout the year while it is not true in real word. It should change as the outdoor temperature and humidity change throughout the year. For this case, the effectiveness is set at 70% and the effectiveness for the heat exchanger is assumed to be 75%.

Two representative outdoor conditions were analyzed, namely 1) summer condition, and 2) winter condition. And the latent load, which is produced when moisture in the air goes from a vapor to a liquid state, is not calculated in this paper but will be discussed in the future work. In order to evaluate the objective function as defined, it is necessary to specify some parameters first (Table II).

$$Q_z = q_s + q_i + q_t + q_o + q_e + q_l \tag{2}$$

where:
$q_s$ = solar load
$q_i$ = infiltration air load
$q_t$ = envelope thermal load
$q_o$ = occupants load
$q_e$ = equipment load
$q_l$ = lighting load

As shown above, the zone load is made up of solar load, lighting load, equipment load, occupants load, infiltration air load and envelope thermal load (heat gains to zone were assumed as positive). The zone heating and cooling loads are met by supplying conditioned air to the zone such that the product of the mass flow rate of the supply air, the specific heat of air and the temperature change of the air from supply ($T_s$) to return ($T_r$) are equal to the zone thermal load:

$$q_i = m_i \cdot cp_{air} \cdot (T_z - T_a) \tag{3}$$

$$q_t = UA \cdot (T_z - T_a) \tag{4}$$

Since the heat gain from lighting, equipment occupants and solar was already set up, the load values of infiltration and envelope thermal conduct can be determined from the thermodynamic relationships as described above, the zone load can be figured out for the energy consumption simulation.

For the summer condition simulation, five parameters: condenser entering temperature, chilled water supply temperature, chilled water mass flow rate, supply air temperature and flow rate are set as variables. Ten different values are selected for each parameter so there are 50 different scenarios in total. As only hot water supply temperature and mass flow rate, supply air temperature and flow rate were changed in the winter condition, 40 group of total power density resulted from the simulation. The component energy consumption was simulated with polynomials, as described below:

$$E_{chiller} = \frac{Q_{avail} \cdot ChillerEIRFTemp \cdot ChillerEIRFPLR}{COP_{ref}} \tag{5}$$

$$E_{pump} = v_{water} \cdot \frac{Pump_{Head}}{Total_{Efficiency}} \tag{6}$$

$$E_{fan} = f_{pl} \cdot m_{design} \cdot \frac{P_{rise}}{e_{tot} \cdot r_{air}} \tag{7}$$

where:
$Q_{avail} = Q_{ref} \times ChillerCapFTemp$
$V_{water}$ = mass flow rate of chilled/hot water
$f_{pl}$ = air part load factor
$m_{design}$ = fan design flow rate
$P_{rise}$ = fan pressure rise
$E_{tot}$ = fan total efficiency
$\rho_{air}$ = density of air

In the heating situation, the fuel input was calculated with this equation [13]:

$$F_{boiler} = m_{hw} \cdot cp_{water} \cdot \left[\frac{T_{hws} - T_{hwr}}{BE \cdot VHI}\right] \cdot 3600 \tag{8}$$

where:
BE = boiler efficiency
VHI = fuel heat value
$m_{hw}$ = hot water mass flow rate
$cp_{water}$ = specific heat capacity of water
$T_{hws}$ = hot water supply temper
$T_{hwr}$= hot water return temperature

TABLE III. CASE DESCRIPTION FOR THE TWO CONDITIONS

| **Cases(summer)** | **Simulation Description** | **Results Range** |
|---|---|---|
| 1 (group 1-10) | Increase condenser entering temperature (50-68 °F) | 4.87-5.21 w/ ft$^2$ |
| 2 (group11-20) | Increase chilled water supply flow rate (0.4-0.7 lbm/s ) | 4.70-5.40 w/ ft$^2$ |
| 3 (group 21-30) | Increase chilled water supply temperature (41-59 °F) | 5.14-5.12 w/ ft$^2$ |
| 4 (group 31-40) | Increase supply air flow rate (0.4-0.7 lbm/s) | 4.57-5.27 w/ ft$^2$ |
| 5 (group 41-50) | Increase supply air temperature (59-68 °F) | 5.24-4.75 w/ ft$^2$ |

| **Cases(winter)** | **Simulation Description** | **Results Range** |
|---|---|---|
| 1 (group 1-10) | Increase hot water supply temperature (176-194 °F) | 5.95-5.61 w/ ft$^2$ |
| 2 (group11-20) | Increase hot water supply flow rate (0.4-0.7 lbm/s) | 5.60-5.36 w/ ft$^2$ |
| 3 (group 21-30) | Increase supply air flow rate (0.6-0.8 lbm/s) | 5.36-5.16 w/ ft$^2$ |
| 4 (group 31-40) | Increase supply air temperature (85-92 °F) | 5.12-4.88 w/ ft$^2$ |

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## IV. RESULTS

Figure 2 illustrates the power density for five different cases from largest to the smallest in the summer condition. The different colors indicate the breakdown of the electricity usage. Lighting and equipment represent fixed loads, while chiller, pump and fan energy, respectively, vary in response to the each specific combination of setpoints. Variation in total building energy for the summer condition is 18%. This indicates that use of the best setpoint combination could achieve an 18% reduction in total building energy compared to the worst setpoint combination. As we can see, HVAC system (including chiller, cooling tower pump, chiller water pump and supply air fan) is the biggest electric consumer in the building, which accounts for around 45% of total energy consumption, while lighting and equipment account for around 22% and 33% of the total electricity consumption, respectively. According to Table III, the maximum power density can reach 5.40w/ft$^2$ when the chilled water flow rate at the biggest value and a small supply air flow rate can decrease the energy consumption to 4.57w/ft$^2$. These calculations could be repeated at any desired interval to enable the continuous reassessment and adjustment of setpoints.

Figure 3 illustrates the power density for four different cases from the largest value to smallest value in winter condition. In this case, the tradeoff is between boiler fuel inputs, pump and fan power. As the natural gas boiler replaced the electrical chiller for conditioning the zone temperature, the electricity usage is decreased, because cooling tower pump is not needed, so the pump energy percentage is also reduced. As a result, the HVAC system (including pump and fan) only accounts about 30% of total electricity consumption.

The best and worst scenarios happened when the hot water flow rate is the highest and when the supply air flow rate is lowest respectively, which was similar to the results for the summer condition. The largest power density is 5.95w/ft$^2$ and the smallest value is 4.88 w/ft$^2$ based on Table III. The maximum potential savings due to setpoint manipulation for the winter condition was 22%. As before, this process can be repeated at any desired time interval to allow continuous dynamic adjustment of setpoints to achieve maximum energy efficiency.

The energy performance of this particular building/HVAC system combination was evaluated for typical summer (cooling) and winter (heating) scenarios in order to illustrate the methodology and the energy saving potential of dynamic setpoint manipulation. While the magnitude of the potential energy savings would be expected to vary for different buildings and locations, the methodology would still be applicable and useful provided the proper information was available to accurately model the HVAC system and its components. The methodology could also be used to evaluate the effectiveness of advanced control strategies by comparing the energy savings predicted or realized by those methods to the maximum potential savings identified using the approach described here.

## V. CONCLUSION

A methodology was developed and demonstrated for determining the impact of HVAC control system setpoints on the total building energy requirements for different building operation situations in the cooling and heating seasons in order to quantify the maximum potential energy savings due to dynamic setpoint adjustment. According to the simulation result, the energy saving potential through possible optimum control is substantial and more noticeable in winter season. The potential saving can be as high as 18% and 22% for cooling and heating, respectively, when comparing the best performance with the worst one. Different control system setpoints provide different degree of energy savings. Minimizing the supply air flow rate is shown to be the most effective measure to save electricity usage in both cooling and heating season, while a large chilled/hot water flow rate will consume the most power. The results suggest that control strategies that are capable of dynamically adjusting setpoints in response to environmental and occupant conditions can potentially save a substantial amount of energy as compared to fixed setpoints.

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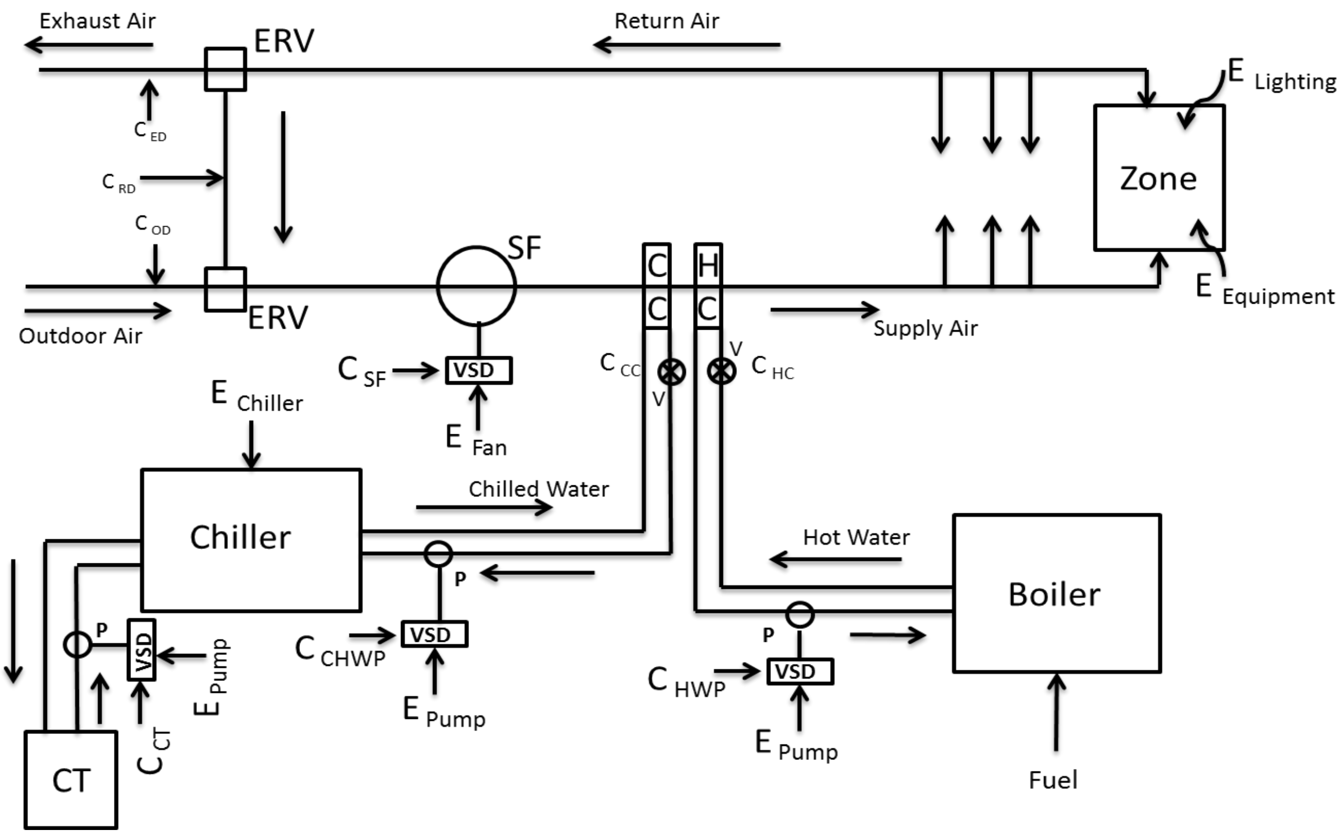


**Nomenclature**

| | |
|---|---|
| ERV | Energy Recovery Ventilator |
| SF | Supply Fan |
| CC | Cooling Coil |
| HC | Heating Coil |
| VSD | Varied Speed Driver |
| P | Pump |
| V | Valve |

| Control Point Lists | |
|---|---|
| $C_{ED}$ | Exhaust Air Damper |
| $C_{RD}$ | Return Air Damper |
| $C_{OA}$ | Outdoor Air Damper |
| $C_{SF}$ | Supply Fan Driver |
| $C_{CC}$ | Cooling Coil Valve |
| $C_{HC}$ | Heating Coil Valve |
| $C_{CT}$ | Cooling Tower Pump |
| $C_{CHWP}$ | Chilled Water Pump |
| $C_{HWP}$ | Hot Water Pump |

Figure 1. System Schematic

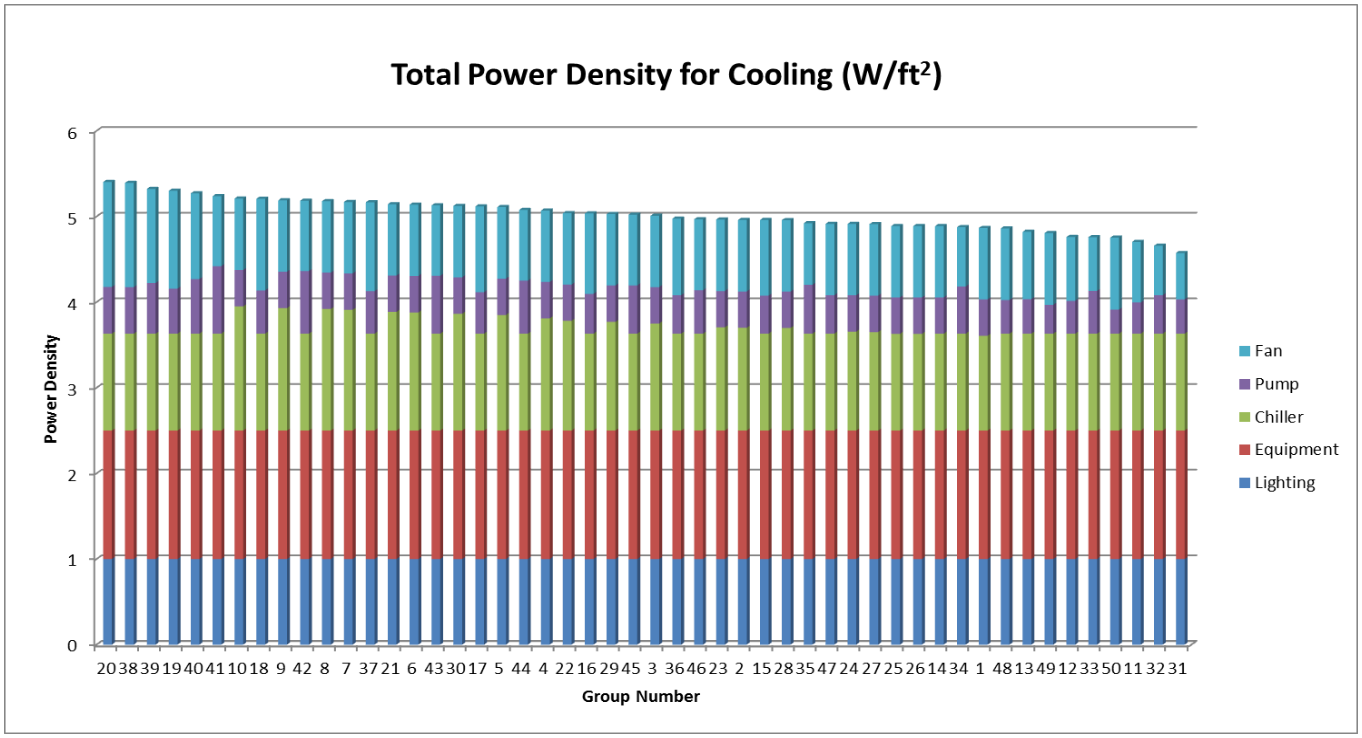


Figure 2. Total Power Density for Summer Condition

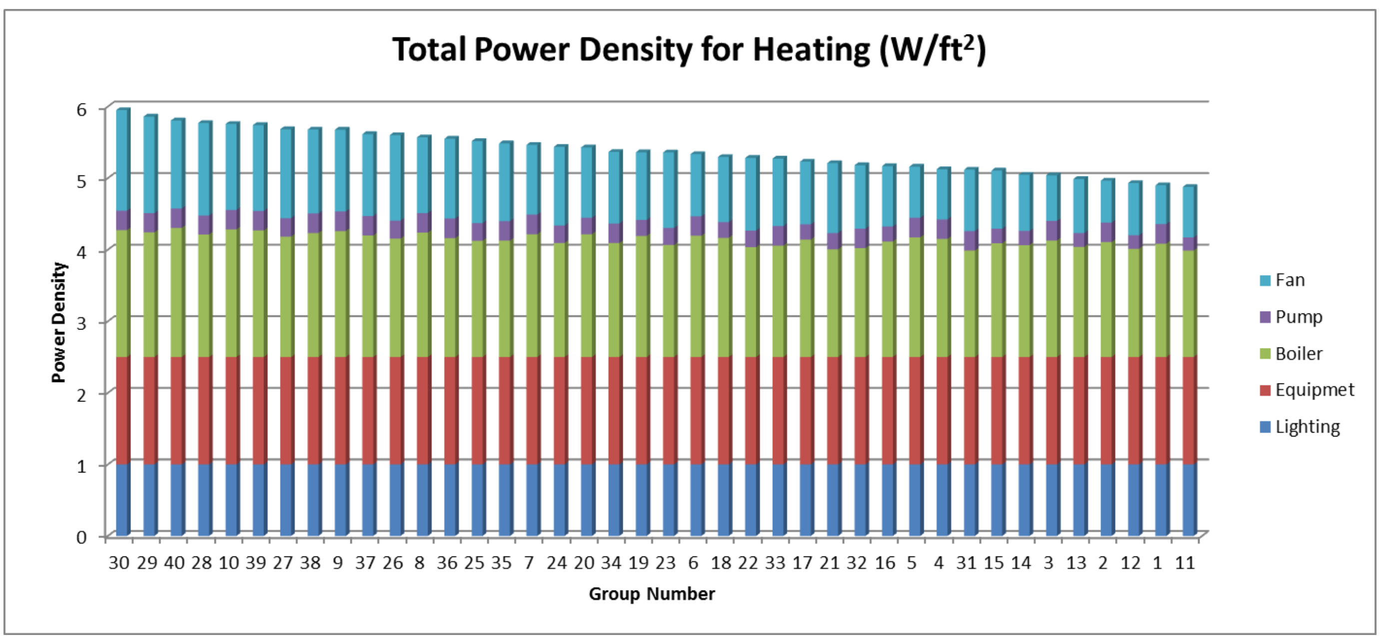


Figure 3. Total Power Density for Winter Condition