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Calibration of HVAC equipment PID coefficients for energy conservation

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ABSTRACT

The combination of proportional–integral–derivative (PID) coefficients for a set of equipment in a heating, ventilating, and air conditioning (HVAC) system has an impact on the overall system energy consumption as well as the ability for the system to maintain temperature setpoints. A simple calibration methodology is discussed where successive optimization on the set of proportional, integral, and derivative coefficients is performed to reduce the energy consumption of the system. The calibration methodology discussed here is applied to a two-zone building for a summer design day. The results show that calibration of proportional coefficients can reduce the system energy consumption by up to 29% and can improve meeting temperature setpoints by up to 45%. Successive calibration of integral coefficients can increase the energy savings up to 35% and can improve meeting temperature setpoints by up to 52%. The successive calibration of derivative coefficients has a negligible impact on the energy conservation and the ability to meet temperature setpoints. The re-calibration of proportional coefficients with the new values of integral and derivative coefficients yields an additional 2.3% increase in energy savings.

1. Introduction

The Department of Energy states that household heating, ventilating, and air conditioning (HVAC) systems consumed 356 billion kWh in 2001 [1]. Traditional HVAC design focuses solely on attaining spatial comfort requirements regardless of the energy cost. Recently, more attention has been focused on the reduction of HVAC energy costs while maintaining comfort requirements, whether it be in the form of more efficient equipment [2–4], novel approaches to HVAC energy storage [5], or supervisory control techniques [6–8]. In addition, novel computational techniques in HVAC design have emerged for energy usage estimation, data collection techniques, and data visualization [9].

Supervisory control techniques often involve significant alterations to existing controls technology (e.g. [10–12]), so the ability to implement these techniques may be difficult. Therefore, numerous optimization studies (e.g. [13–24]) have been performed to develop a means to calibrate the existing input parameters for various HVAC equipment. The majority of these studies involve the use of an optimization technique to determine the parameters that minimize energy usage while maintaining constraints related to comfort requirements. Some of these studies have focused on the use of an automated approach to tuning the parameters in a proportional–integral–derivative (PID) controller to achieve optimum performance and to avoid the inherent instabilities in HVAC systems [25]. Early work by Brandt [26] discusses the need for this optimization as advantageous over the use of manual tuning of parameters, and Nesler [27] states that computational approaches are advantageous to completing this task. Pinnella et al. [28] determine the optimal integral-only control parameters for the system by achieving a critically damped system response to a step input change in load. Bi et al. [10] later advance auto-tuning methods by offering multiple tests and different tuning methods depending on the number of unknown parameters in the system. In addition, Wang et al. [29] discuss the incorporation of new auto-tuning PID design rules for optimization on a system of multiple parameters. They achieved improved HVAC performance experimentally when their scheme was applied.

Generally, controllable HVAC equipment follows a PID-type control schematic that adjusts equipment settings (output) based on current system conditions (input). For example, in a variable-air-volume (VAV) control method that is commonly implemented in modern office buildings, the damper opening is modulated based on difference between the room and setpoint temperatures [30]. The PID control implementation for these dampers may be expressed mathematically as

$$C = C_{prev} + \alpha_P \Delta T + \frac{\alpha_I}{t}\int_0^t \Delta T(\tau)\,d\tau + \alpha_D\left(\frac{d(\Delta T)}{dt}\right) \quad (1)$$

where the control output C is based on the temperature deviation $\Delta T = T_{room} - T_{sp}$, where T_{room} and T_{sp} are the room and setpoint temperatures, respectively. The value of C_{prev} is the control output determined from the previous PID calculation. Eq. (1) shows that

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the PID coefficients α_P, α_I, and α_D impact how C is updated based on ΔT.

PID control settings may also be implemented on a variable-speed fan or pump. Hartman [31] uses variable speed pumps to improve the chiller efficiency relative to constant chiller flow. The efficiency of a variable-speed chiller is more affected by changes in the condenser water temperature than a constant speed chiller, and therefore the adjustment of a variable-speed chiller allows for better energy savings. This result is in agreement with Teitel et al. [3], who showed that variable speed fans provide better energy efficiency than simple on/off control. In addition, Koh et al. [4] showed that a modular varying refrigerant flow system can provide up to 70% in energy savings by eliminating duct losses, improving control over the air supply temperature, and by providing simultaneous heating and cooling in different regions of the system.

The work presented here discusses a computational tool that determines how to tune the PID control coefficients such that the setpoint temperature is best maintained while near-optimal energy conservation is achieved. The concept of applying simple adjustments to existing equipment for reducing energy consumption has been already performed for cooling towers [32], and a brief sensitivity analysis of the Proportional Control Coefficient has been performed by Krarti and Al-Alawi [33]. Here, the concept is applied as a general scheme to a system of dampers, fans, and chillers with the goal of reducing the energy consumption for a summer design day. In this study, the damper setting is expressed as an added minor loss coefficient in the supply duct. In addition, the fan speed applies Eq. (1) where an effective temperature deviation $(\Delta T)_e$ is calculated as

$$(\Delta T)_e = \max(\Delta T_1, \Delta T_2, \ldots, \Delta T_{N_r}) \quad (2)$$

where N_r is the number of rooms served by supply air from the fan.

In this study, the values of PID coefficients may be set independently for each control or shared among common devices. In previous studies [14,34], only proportional control was considered, and their set of α_P values were chosen arbitrarily provided that they allowed the room temperatures to converge to their respective setpoint values. In this study, the set of PID coefficients are calibrated towards reducing energy consumption, yet it will be shown that the reduction of energy consumption also improves the ability for the system to meet setpoint conditions.

2. Modeling

This study applies the PID coefficient calibration method on a simple system with predictable loads. Transient load distributions may be ascertained through either experiments or via commercial software. In this study, DesignBuilder [35] software was used to develop the loads for the two-room building shown in Figs. 1 and 2. This software has been used in other studies and adheres to the European Parliament Board of Directive (EPBD) Standard [36]. The building contains office (18.8 m^2) and restroom (10.9 m^2) spaces. The transient activity in these spaces follows templates provided by DesignBuilder. The building contains one internal door between the two spaces, and one door connects the office space to the outside environment. All external walls contain 30% glazing. Each external wall contains 0.1 m brick, 0.0795 m extruded polystyrene, 0.1 m concrete block, and 0.013 m gypsum board. The internal partition contains two 0.025 m gypsum boards surrounding a 0.1 m airgap. The flat roof contains 0.01 m asphalt, 0.145 m glass wool, an 0.2 m air gap, and 0.013 m plasterboard. The loads and outside air dry-bulb temperature were calculated in subhourly increments for a summer design day (July 5) in Philadelphia, PA. The load calculations included climate, occupancy, infiltration, solar, lighting, and equipment loads. The resultant transient loads and external temperature for this system are shown in Fig. 3.

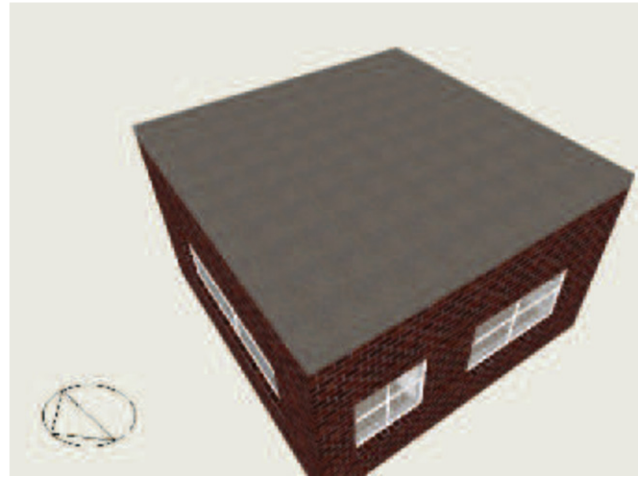

Fig. 1. An external view of the two-room building in this study. The image was created using DesignBuilder software.

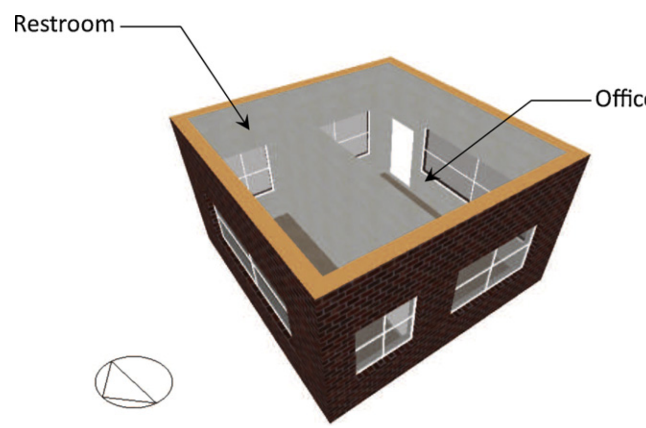

Fig. 2. An open view of the two-room building in this study. The image was created using DesignBuilder software.

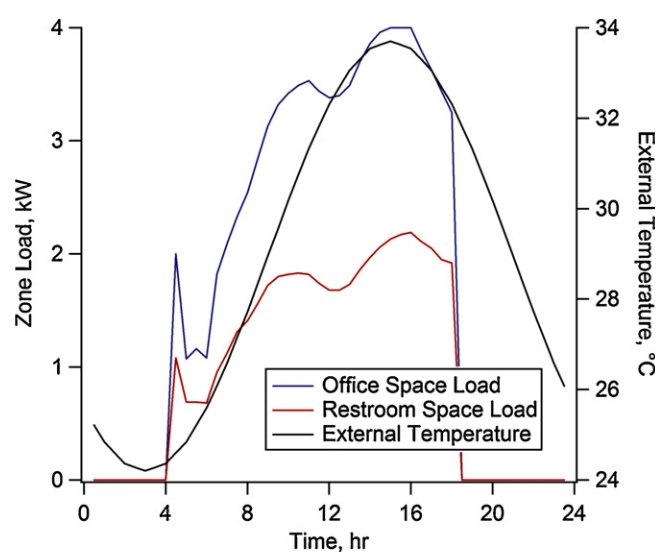

Fig. 3. The transient system loads and external temperature calculated by DesignBuilder. Time zero corresponds to midnight.

This study applies the in-house code Lumped HVAC (L-HVAC) [37] on the above system. L-HVAC implicitly solves the coupled