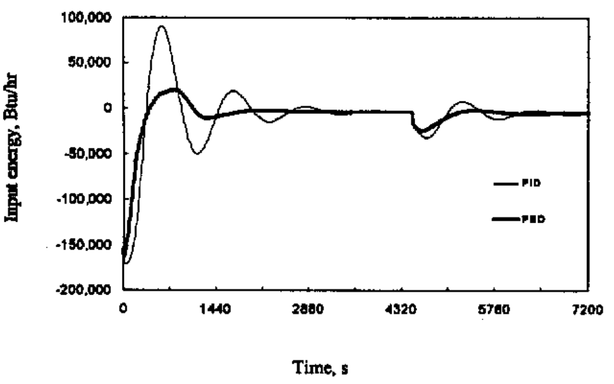

Figure 2 Transient response of heat exchanger input energy.

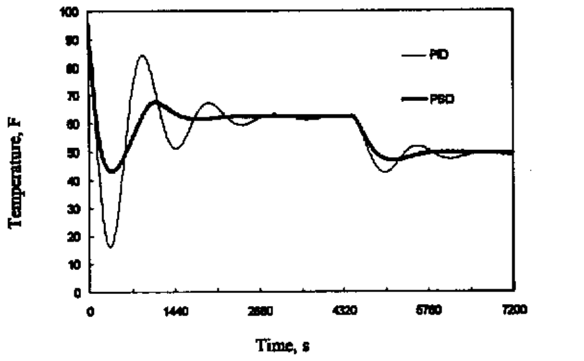

Figure 3 Transient response of heat exchanger discharge air temperature.

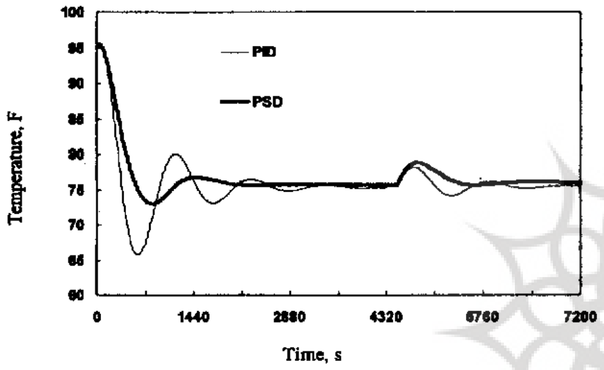

Figure 4 Transient response of zone air temperature.

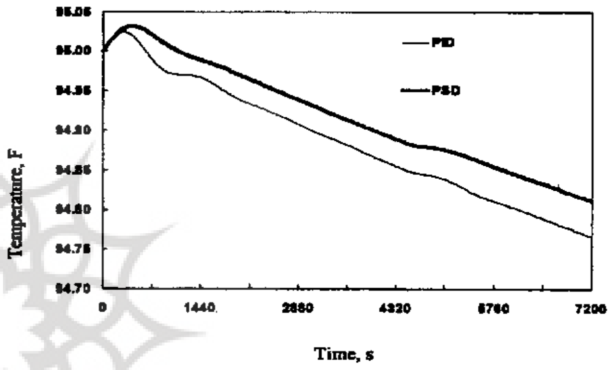

Figure 5 Transient response of zone wall temperature.

Table 1 Cost comparison for PID and PSD methodologies.

Case	Description summary	Energy Cost		Comfort-Penalty Cost, 10^{-3}	
		PID	PSD	PID	PSD
1	Cooling, Q_z=1365 Btu/h, T_0=99°F, $T_2(0)=T_3(0)=T_4(0)$=95°F K_p=109.99, $K_i=K_s$=2.0, K_d=1.01, k=19	46.5	16.7	2.87	2.73
2*	Step change in cooling load, Q_z=2730 Btu/h after 4500 s	48.4	17.8	2.96	2.90
3*	Heating, $T_0=T_2(0)=T_3(0)=T_4(0)$=60 °F	16.5	11.6	1.12	1.24

* For Cases 2 and 3, only the deviation from Case 1 is noted.

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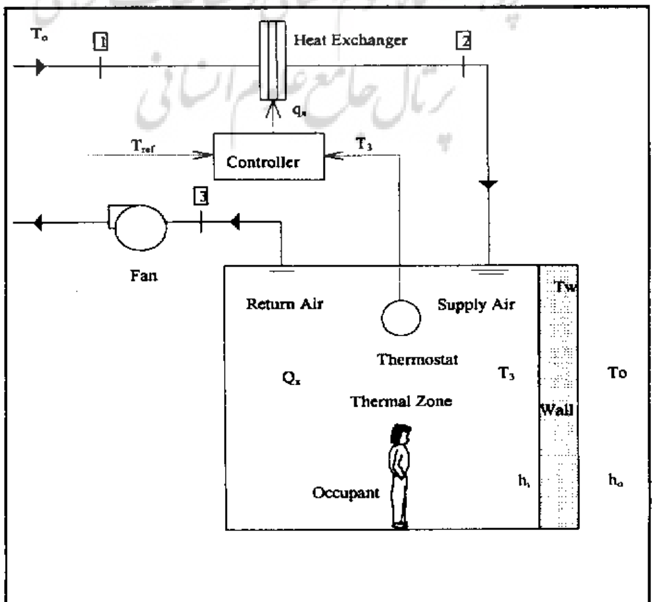

Figure 1 Building thermal energy system.

It must be noted that the general shape of the response of T_3 for the PID- controlled conforms to the response-shape recommended by Nesler and Stoecker (1984). The variation in T_2 indicates that lower peak to peak temperatures are achieved with the PSD controller than the PID controller. For the PSD controller, the shift and reduction in the response-period are observed in T_2, as compared with that for the PID controller. These phenomena are attributed to the non-linearity that is introduced by the sum controller. It must be noted that while heat exchanger freeze-up is of concern for the PID controlled heat exchanger due to the low discharge temperatures (Fig. 2), it is avoided by utilizing the PSD controller.

The temperatures at the discharge of the heat exchanger for the PSD controller influence the thermal zone temperature (Fig. 3) in a way that fluctuations are not as severe. If occupant comfort is to oe related to the fluctuations in the supply air temperature, the performance of the PSD is considered to be more desirable. However, the zone temperature steady-state value for the PSD controller is 24.31°C (75.75°F). As a benefit from the PSD controller, the zone wall temperature response reaches higher temperatures during the first few minutes because the energy consumed for cooling is less (Fig. 4). The more energy-efficient operation of the PSD controller is anticipated due to the smaller decrease in T_W. The time response for q_x is shown in Fig. 5. While the initial response for cooling from both controllers is nearly the same, the next immediate need for heating is 26,765 W (91,325 Btu/h) (after 580 s) for PID and 5,840 W (19,928 Btu/h) (after 800 s) for PSD. As a result, the PID controller peak heating requirement is 78 percent higher than that of the PSD controller. After 1440 s, the PSD controller has reached near-steady state conditions, whereas the PID controller requires almost twice the amount of time to reach similar conditions. The PSD and PID controllers result in steady-state cooling requirements of 1097 and 1103 W (3742 and 3766 Btu/h), respectively. Responses for T_2, T_3, and q_x, have fewer oscillations due to utilization of the PSD controller. The unconstrained capacity of the heat exchanger energy requirement during its operation would be less for the PSD controller thathat for the PID controller. The lower energy consumption by the PSD controller is confirmed by smaller amplitudes of T_2 during the entire time period.

The costs for deviation from the setpoint and for heat exchanger energy for both control methodologies are shown in Table 1. The energy cost for the PSD controller is 64 percent lower than that for the PID controller. This is due to lower values of q_x thereby yielding a more energy-efficient operation of the system with the PSD controller. The cost associated with the variation from the setpoint is 4.9 percent lower for the PSD controller than the same for the PID controller. Further, for T_3 (Fig. 3), the slight increase in T_3 (0.22°C) for PSD above that for PID is justifiable when the potential energy savings are 64 percent.

RESULTS AND DISCUSSION

The effects of cooling requirements are given by Ardehali et. al (1997), based on the parametric values given for Case 1 (Table 1). Time responses for T3 is shown in Fig. 2 for PID and PSD controllers. For occupant comfort, the performance of the PSD is considered to be more desirable. The time response for q_x is shown in Fig. 3. Responses for T_3, and q_x have fewer oscillations due to utilization of PSD controller. For Case 2, after 4500 s, the thermal load is increased by 100 percent to 2730 Btu/h and held constant. Similar to Case 1, energy cost (Table 1) for the PSD controller is 63 percent lower than that for the PID controller. However, unlike Case 1, the comfort-penalty costs for both controllers are nearly the same. For heating, energy cost for PSD controller is lower by 42 percent from that of PID controller.

Conclusions and Recommendations

With adaptablity to PID and less oscillations, PSD results in better comfort conditions though the steady-state zone temperature is slightly higher than that of PID control methodology. PSD requires lower energy demand and less frequent equipment switch-over for maintaining setpoint temperature, consequently, this method provides for less energy consumption while maintaining better comfort penalty cost.

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?cost? may reflect financial or energy expenditure or it may indicate deviation from some ideal physical situations. In cost functions, the quadratic forms have intuitive appeal because large errors are penalized more heavily than small errors. When maintaining comfort conditions in a thermal zone, deviations from a setpoint are uncomfortable for the occupants, and doubling the deviation from the same setpoint more than doubles the discomfort (Stengel, 1994).

The cost functions for the analysis are (1) the quadratic forms of energy input and (2) the quadratic forms of the zone air temperature from the desired value of the zone setpoint temperature as the comfort penalty, and are written as

$$J_q = \int_{t_0}^{t_f} (q_x)^2 \, dt \qquad (10)$$

$$J_{set} = \int_{t_0}^{t_f} (T_3 - T_{sp})^2 \, dt \qquad (11)$$

The costs of energy input to the heat exchanger in Eq. (10) and the offset from the setpoint temperature in Eq. (11) capture the costs from the owners and occupants points of views. The separate forms of the cost functions allow for indexing the performance of each criterion separately because quantifying costs for comfort that are difficult to state and the uncertainties that are associated with weighting factors (House et al., 1991) are avoided.

The goal of achieving energy efficiency in the operation of TES, without loss of functionality, motivates one to conduct research on control methodologies

System Description

The PID and PSD controllers are applied to the TES of Fig. 1 and the results of the simulations are presented for various responses and costs. The zone represents a typical office that has a wall adjacent to an exterior in a multi-story building. As such, these exterior zones use a large amount of energy because they typically require provisions for heating in the form of reheat to offset building-skin heat losses. Proper modulations of the local control valves for the cooling and/or heating play an important role in maintaining comfort conditions and the amount of energy expended. The time period chosen for the simulation purposes is 2 h, and the number of time intervals is chosen to be 721 so that the sampling time is 10 s as suggested by Nelson and Steocker (1984). The setpoint temperature for the thermal zone is 23.9°C (75°F). The initial temperatures for the wall, heat exchanger, and thermal zone are 35°C (95°F), and the ambient temperature is constant at 37°C (99°F). The selection of these temperature represents the design conditions for a typical summer day. The thermal load is 400 W (1365 Btu/h). The specific heat, density, and volume of air are 1 kJ/kg·°C (0.24 Btu/lbm·°F), 1.14 kg/m^3 (0.071 lbm/ft^3), and 28.3 m^3 (1000 ft^3), respectively. The zone has a floor area of 9.3 m^2 (100 ft^2) and a wall area of 9.3 m^2 (100 ft^2). The wall is 0.15 m (0.5 ft) thick with density and specific heat of 271.4 kg/m^3 (16.9 lbm/ft^3) and 3.4 13.6 kJ/kg·°C (Btu/lbm·°F). The lag-time for the zone envelope is about 2 h (ASHRAE, 1993). The convective heat transfer coefficients for inside and outside are 9.4 and 34.1 W/m^2·°C (1.65 and 6.0 Btu/h·ft^2·°F), respectively. The supply and return air flow rates are 0.047 m^3/s (100 ft^3/min) with no infiltration or exfiltration. The specific heat, density, and volume of the heat exchanger are 0.12 kJ/kg·°C (0.1 Btu/lbm·°F), 6544 kg/m^3 (409 lbm/ft^3), and 0.09 m^3 (3.3 ft^3) respectively. The tolerance error for the numerical integration is 1×10^{-4}.

Analysis

The tuning process for the PID controller for controlling the thermal zone temperature included a trial and error procedure to obtain a quarter-wave damp response as suggested by Ziegler-Nichols (1942). The gain coefficient values of 109.99, 2.0, and 1.01 apply for K_p, K_i ($=K_s$), and K_d. Also, the windup periods during which the error is integrated for the integral controller or summed for the sum controller are set at the same value of k = 19 time intervals or 190 s.

The time responses for the heat exchanger discharge air temperature, thermal zone air temperature, and wall temperature are shown in Figs. 2, 3, and 4, respectively, for PID and PSD controllers.

during transient conditions (Kuo, 1991). The derivative signal is proportional to the time derivative of the error and is

$$q_d(t_{j+1}) = K_d \frac{e_{3,j} - e_{3,j-1}}{t_j - t_{j-1}} \tag{6}$$

Proportional-integral-derivative Control

For the PID controller, the Eqs. (4), (5), and (6) are combined to yield the control signal for q_x as (7)

$$q_x(t_{j+1}) = K_p e_{j,3} + K_i \sum_{n=j-k}^{j} e_{j,3,n} + K_d \frac{e_{3,j} - e_{3,j-1}}{t_j - t_{j-1}}$$

$, \text{for } n > 0$

Stoecker and Stoecker (1989) refers to Eq. (7) as the position form and also suggests the incremental form of the PID algorithm. The incremental form is derived on the basis of the difference between successive calculations of the position form. The incremental form provides incremental control output instead of an absolute position and avoids integral windup because the controller integral term is stored by virtue of the current position of the control device and is not stored as a numerical value of the algorithm. While the implementation of the incremental form of the PID algorithm is less expensive and avoids the integral windup, the position form is used over the incremental form because the position form allows for (1) positive indication of the controlled device and (2) easy conversion from existing analog-type controls to direct digital control.

Sum Control

The concept of addressing the changing dynamic behavior of the system during the transient conditions separate from the steady-state periods by means of derivative and integral controls motivates developing a control methodology that influences the performance of the system during the entire time period. The integral controller does not see the influence of all the error that has occurred in an absolute sense because the areas under the error urve are algebraically added. The algebraic addition of positive and negative areas does not translate into the £real? error. In other words, thetotal error that has accumulated up to a certain point in time is not accounted for, and only the algebraic sum of the errors is retained. While the rational for integral control and its contribution to reducing the steady- state error is satisfactory, further examination of the integration concept proves otherwise. Intuitively, it appears that if the controller is informed of the sum of absolute errors, and not the algebraic sum of all the errors, the output from the controller would result in a more efficient operation of the system during transient and steady-state periods. As a result, a control methodology based on accumulating the total error has been developed, and its performance is examined for the system of Fig. 1.

energy saving was possible for a system operating under dynamic control

The control signal for discrete time at the end of the time interval j for the proposed control methodology is written as

$$q_s(t_{j+1}) = K_s \sum_{n=j-k}^{j} \left|e_{j,3,n}\right| \tag{8}$$

$, \text{for } n > 0$

where K_s is the sum gain coefficient and the absolute values of the error are summed over the past k times considered as the sum-up period.

Proportional-sum-derivative Control

The newly developed methodology is termed as ?sum?, S, and is examined in conjunction with other control methodologies as a proportional-sum-derivative (PSD) controller. The PSD control signal for the heat exchanger is $q_x(t_{j+1}) = K_p e_{3,j} +$

$$K_s \sum_{n=j-k}^{j} \left|e_{3,j,n}\right| + K_d \frac{e_{3,j} - e_{3,j-1}}{t_j - t_{j-1}} \tag{9}$$

$, \text{for } n > 0$

For regulating a variable volume of air to the thermal zone, a similar controller could be utilized.

Cost Functions

A cost function is commonly formulated to measure the penalty that must be paid as a consequence of the trajectory of the dynamic system. The

$$\rho_a C_a V_z \frac{dT_3}{dt} = \rho_a f\, C_a(T_2 - T_3) + h_i A_i (T_w - T_3) + Q_z \tag{2}$$

$$\rho_w C_w V_w \frac{dT_w}{dt} = h_o A_o (T_o - T_w) - h_i A_i (T_w - T_3) \tag{3}$$

with $T_2(t_0)=T_{2,0}$, $T_3(t_0)=T_{3,0}$, and $T_W(t_0)=T_{W,0}$. The lumped capacitance model is justified due to lack of radiant exchange or direct flux on the wall (Incropera and DeWitt, 1992). The solution of the problem assumes constant material properties, areas, air volumetric flow rate, and zone internal and external conditions for T_0 and h_0. The simulation period of $t=t_0$ to $t=t_f$ is discretized into NV time intervals. The initial value differential equations are solved and integrated simultaneously utilizing IMSL DIVPRK (1987) to obtain transient responses for T_3. For each time step, a value for q_x, based on a given control methodology, positive for heating and negative for cooling, is computed and the new temperature responses are found.

For the purposes of controlling the manipulated variable, q_x, in a system with feedback, various methodologies are used so that the error between the value of the controlled variable, T_3, and the setpoint, T_{sp}, is made smaller. The control objective is defined as the determination of the amount of energy input to the heat exchanger as a function of time, when the system is perturbed from an initial temperature to some setpoint temperature. When controlling TES equipment, typically, feedback controllers with sensors are used. The zone temperature tends to oscillate above and below the setpoint temperature due to the inherent and continuously varying thermal load disturbances. The controller measures T_3 and computes the difference between T_3 and T_{sp}. This difference is termed the error, e. The controller sends a signal to the control device to regulate q_x with a resultant response given by T_3. It must be noted that for implementation in a computer algorithm, the computed values for the manipulated and controlled variables are held constant during each discrete time interval.

Processes where the fluctuations of the controlled variable asa function of time are inherently higher during transient periods than those during near steady-state periods are of particular interest

Proportional Control

The proportional (P) controller has an output for the manipulated variable that is linearly proportional to the error with the constant of proportionality being K_p. The P controller output at each time discretized time j is

$$q_p(t_{j+1}) = K_p\, e_{j,3} \tag{4}$$

where the error is $e_{j,3}=T_{sp,j}-T_{3,j}$.

Integral Control

The offsets in T_3 that are caused by the use of P controllers are due to K_p (Kuo, 1991), and the variation in the thermal disturbance. By introducing the integral controller, the offset is reduced or eliminated. To enhance the performance of a P controller, the integration of the error from k past discrete times, which is commonly referred to as the wind-up period, to the end of the current time interval j to compute the I controller output is stated as

$$q_i(t_{j+1}) = K_i \sum_{n=j-k}^{j} e_{j,3,n} \quad , \text{for } n > 0 \tag{5}$$

The anti-windup for the integral controller is implemented by means of a sliding window algorithm that always accounts for error accumulation during the last k time intervals (Nesler and Stoecker, 1984).

Derivative Control

While the combination of PI control provides for adjusting the shift and reducing the steady- state error, the derivative (D) control is used to enhance the output of the system by means of influencing large slopes of the error curve of the PI gain coefficients K_p and K_i. The experimental work of Nesler and Stoecker (1984) used a pneumatic controlled hardware and focuses on investigating the influence of various combinations of the gain coefficients on the system performance. In that study, the authors suggested that the derivative (D) mode is responsible for faster or slower responses of the system and is unnecessary in TES. They also conclude that TES should be tuned during low-load conditions and that this tuning results in large values for the gain coefficients. When using PID, most studies on the effects of the dynamics focus on tuning K_p, K_i, and K_d for a specific system.

For the purpose of enhancing the performance of the PID controller, tuning strategies have been developed by Ardehali (2000), Ardehali et. al (1997), Dexter and Haves (1989), MacArthur et al. (1989), Donoghue (1977), Pinnella et al. (1986), Pinnella (1988), and Ho (1993). In a study by Clark et al. (1985), TES components with transport delays or transient responses were modeled and simulated, and the theoretical work was found to be in good agreement with experimental testing. Nesler (1986) conducted an experiment utilizing an automatic self-tuning control system for direct digital control systems that use PI control strategies. A tuning method for the PI controllers of TES was introduced and experimentally tested by Bekker et al. (1991).

In an experimental study by Virk and Loveday (1991), the performances of PID controls are compared to those of conventional on/off and predictive on/off controls. The predictive on/off control strategy utilizes a mathematical model to predict if heating or cooling is necessary based on the input and results in a smaller offset from a setpoint (Goodwin and Sin, 1984). Results for the experimental and simulation studies are comparable. The predictive technique achieved an energy saving of 17 percent over the conventional on/off method.

In an investigation by Hackner et al. (1985), models of TES equipment were developed and the optimal control setpoints were determined to minimize the use of energy. The optimal control strategies were implemented via a simulation of the TES, and the effect of system parameters were explored. Hackner et al. found that constant setpoint strategies used 19 percent more energy than the optimal strategies, and that a 3 to 4 percent energy saving was possible for a system operating under dynamic control as opposed to a system in which the control decisions were made on an hourly bass. In the study, it wanoted that optimal control strategies may be impractical to perform on-line during the operation of the TES; however, the results of the optimization could be used to develop control algorithms for an energy management control system to apply at later time. In addition to examining separately the variation of all three modes of control P, I, and D, House and Smith (1990) used an optimal PID algorithm to find optimal values of K_p, K_i, and K_d that minimize a cost function for TES. In a study by Athienitis and Chen (1993), where the performance of an electric floor heating system was investigated, the P control resulted in improved operation of the system with thermal storage compared to on/off control; on/off control causes too much cycling of the system and associated room temperature swings. Winn (1993) has also noted a reduction of energy consumption by means of utilizing PID control in TES.

Various methodologies have been developed to enhance the PID algorithm. One such method is by Huang and Nelson (1991), where PID control was combined with a fuzzy controller. Simulations indicate that better performance can be achieved. The authors noted that a possible drawback of applying fuzzy linguistic reasoning is the required additional computational time that constraints the on-line usage of such methodology.

The literature review indicates an absence of work performed on development of new on- line controllers that would employ similar concepts as those of the PID methodology, that can offer a better performance, and that are capable of being installed in existing installations without significant changes to the actual system. The objectives of this study are (1) to describe the PSD control methodology and (2) to compare the performance of the PSD and PID controllers for (a) a cooling application with a step-change in load and (b) a heating application.. The effects of the PID and PSD controllers are studied for a constant- air-volume (CAV) TES that provides heating or cooling as necessary to a thermal zone (Fig. 1). The external loads may be due to the ambient air temperature, T_o, acting through the exterior and interior convective heat transfer coefficients, h_o and h_i, across the zone envelope wall. The internal loads, Q_z, may be due to sensible heat from occupants and other sources of heat gain. The system is typical of once-through TES and was studied by House et. al (1991), Ardehali and Smith (2000), Ardehali et. al (1997). The TES is described by a set of differential equations:

$$\rho_X C_X V_X \frac{dT_2}{dt} = \rho_a f\, C_a (T_o - T_2) + q_X \qquad (1)$$

ABSTRACT

A newly-developed proportional-sum-derivative (PSD) control methodology is introduced and applied to a numerically modeled thermal energy system (TES). The need for on-line local TES controllers that can be adapted to existing proportional-integral-derivative (PID) controllers without additional effort for re-tuning motivates this work. The PSD control methodology is founded on the basis that increase in energy is achieved by means of decreasing both energy input and heating/cooling-cycle changeovers. The objectives of this study are (1) to describe the PSD control methodology and (2) to compare the performance of the PSD and PID controllers for (a) a cooling application with a step-change in load and (b) a heating application. For the case where the gain coefficients are equal, the comparison is made using cost indices based on the controller energy and the offset of temperature from the setpoint temperature. In a numerical simulation of a TES, the results show that changeover from PID to PSD algorithm results in 63 and 42 percent lower energy cost for the cooling and heating applications, respectively. Also, because of fewer oscillations in zone temperature, the change-over to PSD results in better comfort conditions though the steady-state zone temperature is 0.5 percent higher than that of PID control methodology. The PSD control methodology is adaptable to PID controllers that are currently in operation and should be considered in new installations.

KEY WORDS

Energy Efficiency; Controls; Thermal Dynamics; PID; PSD.

INTRODUCTION

The control of processes, such as temperature, flow, and pressure regulation in petroleum refinery applications, temperature regulation in manufacturing clean rooms or hospital surgery rooms, melting or solidification in manufacturing processes, landing or take-off of aircraft, fuel and oxidizer mixing in combustion processes, is an important factor, among others, that influence the final outcome of the process. In some processes, the control of system dynamics during transient periods may be of great importance. Processes where the fluctuations of the controlled variable as a function of time are inherently higher during transient periods than those during near steady-state periods are of particular interest. During transient periods, unpredicted changes in the controlled variable can be severe and may even be considered catastrophic. Therefore, the advantage of one control methodology over its counterpart is in its ability to reduce unnecessary fluctuations to avoid unpredicted changes in the controlled variable. It is desirable that any newly developed or enhanced control methodology should be adaptable to existing systems and should have a significant advantage to be considered in new installations.

The proper value for the manipulated variable must be generated so that the controller achieves a particular goal. For the purposes of performance evaluation of a controlled process, the concept of a cost is introduced. This cost is related to the energy expended to accomplish a desired goal. Additional concerns, such as deviation from certain levels of expectation described by a setpoint temperature in cooling a thermal zone or an acceptable amount of vibration during aircraft landing, can also be associated with certain costs. These costs are used in the evaluation of the controller effectiveness. In general, the controller must be able to vary the manipulated variable in accordance with two types of fluctuations in the controlled variable. One type of fluctuation is attributed to disturbances while another is due to occurrences of overshoots and undershoots in the manipulated variable that are caused by a lack of knowledge of future events. For applications of utilizing control methodologies in TES, the heat exchange in the heat exchanger for conditioning building air is considered as the manipulated variable and the thermal zone temperature is the controlled variable (Metha and Thumann, 1997). Because the manipulated variable is directly related to the energy expenditure, the energy-efficient operation of the system is achieved when the fluctuations are reduced in amplitude and in frequency. It is also important to note that the role of the type of control methodology that is employed is even more crucial because extensive dynamics due to thermal mass storage and ambient disturbances are inherent in the system.

The goal of achieving energy efficiency in the operation of TES, without loss of functionality, motivates one to conduct research on control methodologies. To achieve this goal, the TES and the building must be modeled mathematically and control strategies applied so that more energy-efficient strategies can be developed. A review of literature for TES controls indicates that one of the earliest work that examines the dynamics of a thermal system is by Shavit and Brandt (1982), where the simulation model utilized proportional (P) plus integral (I) control strategy. In that study, the system performance is analyzed as a function

Application of Proportional-Sum-Derivative Control Methodology* for Energy Efficiency Enhancement of Thermal Energy Systems

(*US Patent Pending)

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