

EXPERIMENTAL VALIDATION OF A METHODOLOGY FOR DETERMINING HEATING SYSTEM CONTROL STRATEGIES

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ABSTRACT

Methodologies developed for the optimization of centralized cooling facilities (Braun et al. 1987; 1989a,b) have been applied to a simple heating system consisting of a fan and a gas-fired furnace. The optimal strategy has been determined for each of the two components separately and for the system as a whole. It is shown that the optimal control laws for each component may or may not apply to the system as a whole. The optimal strategy for the fan is applicable when considering the entire system. In contrast, the strategy for minimizing the furnace energy only is not relevant to the system as a whole.

Minimum total energy for the entire system is obtained by operating the furnace at the highest possible water temperature, even though this is the lowest furnace efficiency. For this system, the increase in fan power with airflow rate is more costly than the decrease in furnace energy with flow rate. The results of this study demonstrate the need to consider the energy use of all components of a system when performing an optimization procedure.

INTRODUCTION

Heating and cooling facilities consist of many components, each of which may have several control variables. Optimal control of such plants involves determination of the settings for these variables that minimize the cost of energy consumption at any instant of time. The optimal settings change with time in response to changes in the external forcing functions such as loads and weather.

Recent work by Braun et al. (1987; 1989a,b) has led to generalized methodologies for the determination of optimal control strategies for centralized cooling facilities. Two approaches are described in this series of papers. In one, detailed expressions for the dependence of the performance of each component on the control variables and forcing functions are formulated. These component relations are then combined and the control strategy that optimizes the performance of the entire system is determined. In the second approach, the system is considered as a whole. A single relation is developed that describes the dependency of the total energy supplied to the system upon the control variables and forcing functions. Both of these methodologies have been applied successfully to the chilled-water facility at the Dallas/Fort Worth Airport.

In the approach developed by Braun et al., the operating cost, J , for any system is a function of the forcing functions, continuous control variables, and discrete control

variables or modes. The optimization problem is to minimize the cost function at any time:

$$J = J(f, u, M) . \quad (1)$$

The forcing function, f , is a vector containing all uncontrolled variables, such as the load on the system and the outdoor wet-bulb temperature. The continuous control variables are those such as chilled-water set temperature that can be varied continuously over the control range. The discrete control variables are those controls that have discrete settings, such as high and low fan speeds. In addition, each of the control variables may have limits, such as a minimum allowable temperature or a maximum possible flow rate. These constraints must be considered in the optimization process.

A simple function for which an optimum exists that can be determined analytically is a quadratic function. Braun et al. (1987) have shown that power requirements for chillers, fans, and pumps can be adequately expressed as quadratic relationships. Similarly, they have shown that the cost of energy for an entire cooling plant can also be represented as a quadratic. As a result, the cost function represented by Equation 1 can be written in general as

$$J(f, M, u) = u^T A u + b u + f^T C f + d f + f^T E u + g , \quad (2)$$

where A , C , and E are coefficient matrices, b and d are coefficient vectors, and g is a scalar constant. The coefficients depend on the operating modes of the system and must be determined for each discrete control mode, M .

An advantage of the formulation of the cost function as given by Equation 2 is that the solution for the optimal control vector that minimizes the total cost can be determined analytically. The minimum cost is determined by applying the first-order condition for a minimum, i.e., differentiating the cost with respect to the control variables and setting the derivative equal to zero. Solving for the optimal control vector yields control laws for which the settings depend linearly on the forcing functions. These laws are of the form

$$u = K f + k . \quad (3)$$

This formulation and the resulting linear control laws are the result of unconstrained optimization. When con-

order, time-delayed systems. The desirability of critical damping is, of course, a matter of preference. In the case of humidity control, however, the threat of overshoot to the point of saturation makes a critically damped response one of the best choices. The simplicity of the root locus tuning method, along with its improved performance compared to trial-and-error and quarter-amplitude decay methods, makes it useful in other HVAC applications as well.

Problems with the root locus method are those that plague other tuning methods. Most obvious is system identification; the tuning method requires some means of determining the physical system parameters of time constant, time delay, and actuator operating time. Another difficulty encountered during experimental trials was hysteresis in the actuator. This had to be compensated for to achieve the results shown in Figure 8.

These problems notwithstanding, the root locus tuning method is a new approach that uses classical techniques to answer a need in the control field. It is applicable to the first-order, time-delayed systems in most HVAC control situations. The method also shows promise in initiating further research. Work is already under way to extend the method to higher-order systems and couple it with on-line system identification techniques.

CONCLUSIONS

A tuning method was developed that enables PI controllers used with first-order, time-delayed systems to be tuned to a critically damped response. The method was developed in two stages. The first stage, based on root locus and pole-zero cancellation techniques, provides the proportional gain, K_p , and the integral gain, K_i , for a PI controller used with a quick actuator.

The second stage modified the controller gains to accommodate constant-velocity actuators such as electric gearmotors. Gain reduction is required to compensate for a constant-velocity actuator's inability to follow a control signal instantly. The need for, and degree of, gain reduction depends on the relative size of the system's time constant and time delay and on the speed of the constant-velocity actuator. Because these actuators are difficult to model with classical, linear control theory, empirical curve fitting provided a gain reduction factor that allows the tuning method to be used with constant-velocity actuators.

The tuning method was tested in computer simulation and in an experimental humidity control loop. Both simulation and experiment demonstrated the ability of the tuning method to produce critically damped responses with first-order, time-delayed systems that contained constant-velocity actuators. Since most temperature and humidity control

systems are first-order, time-delayed systems, this tuning method is expected to find use in applications where a PI controller is used with electric valves and damper controls.

ACKNOWLEDGMENTS

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DISCUSSION

Philip Haves, Professor, University of Oxford, Department of Engineering Science, UK: Your stated objective in tuning the controller was to avoid overshoot in the duct as well as the room, yet your slide showed large overshoot in the duct humidity. How do you tune the controller to avoid overshoot in the duct humidity when the controlled variable in your single-input, single-output controller is room humidity?

J. Bekker: The stated objective of the tuning method as described in the paper is to achieve a critically damped response to the controller, which, in this case, controls room humidity. The system and controller require overshoot in the duct to achieve a critically damped response in the room. The real danger is not overshoot per se, but overshoot to the point of saturation. This condition was addressed in the research that led to this paper but is too complex to treat thoroughly in the paper itself. In the context of the paper, the single humidity loop was used merely as an example of the tuning method.

straints on the control functions are present, the laws may not be linear in the control variables. The utility of this optimization process has been established for central chilled-water systems. In this paper, the application to an air-heating system with a gas-fired furnace will be demonstrated.

EXPERIMENTAL FACILITY

The facility employed is an HVAC test system at a Belgian university. It provides a controlled environment and was originally constructed for testing of HVAC components. It is well instrumented with temperature, flow, and power sensors. A direct digital control (DDC) controller has been installed to allow automatic control of conditions throughout the facility.

The facility is shown schematically in Figure 1. Airflow through a centrifugal fan exits into a duct. A damper helps control the flow. The air then flows sequentially through several conditioning units. The preheater is an electrically heated water loop and heat exchanger combination that heats the air entering the humidifier. The humidifier supplies water and is controlled to maintain a set humidity in the airstream. The cooling coil is supplied with service cold water and can cool and condense water from the airstream. The main heater element is a heat exchanger supplied with hot water from a natural-gas-fired boiler.

In the experiments described in this paper, optimal control strategies were developed for the fan and the main heating system.

The fan is a centrifugal unit equipped with a three-speed motor. In addition to damper control, the fan is fitted with variable-inlet vanes to allow control of the flow rate. Both inlet vane settings and damper settings can be varied from fully closed to wide open. The flow ranges of the motor overlap slightly for the three different speeds. Power is measured with a three-phase wattmeter.

The natural-gas-fired furnace heats water to a temperature between 40°C and 90°C. The desired outlet temperature of the furnace is maintained between the high and low limits of the deadband by the two-position action of the aquastat. A three-way valve allows regulation of the flow through the exchanger. In the experiments conducted in this study, the valve was set fully open and the temperature of the hot water loop was maintained within the deadband by the controller.

The experiments were conducted at steady-state conditions in order to determine the performance of each component. The desired conditions were set and the test facility operated until the measurements were stable; this usually took on the order of two hours. The desired experimental conditions were established using the energy management system, and the sensor outputs were recorded using a personal computer. The performance relations obtained in this manner are usable in control strategies in which changes in settings occur at long times relative to the response of the equipment.

COMPONENT OPTIMIZATION

The approach followed in this study is to first treat each of the two components (fan and furnace) as a system. For each component, the optimal control strategy that minimizes the cost of operation of that component alone will be determined. This will yield useful performance relations and demonstrate the optimization process. Following the individual unit optimization, the two units will

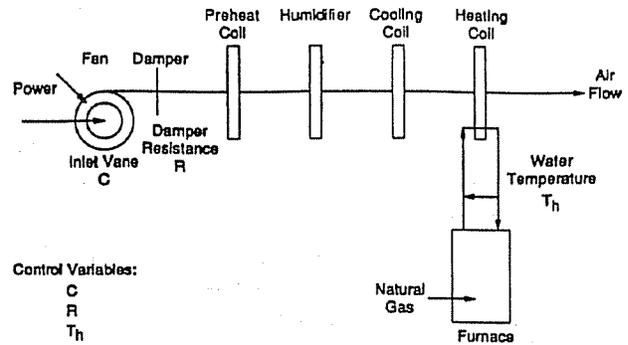


Figure 1 Schematic of test facility

be combined and treated as a system. The optimal strategy for the combination then will be determined. This will illustrate the trade-offs in energy use between the two components and the need to consider the operation of the entire system. The optimal strategy for the fan will be considered first.

Optimization of Fan Operation

The fan power is a function of two continuous control variables—a discrete control variable and a forcing function. The two continuous control variables are the inlet vane setting and the setting of the damper at the outlet of the fan discharge. The damper settings can be varied over a wide range from fully open to fully closed. The discontinuous control setting is the speed mode; three modes of low, medium, and high are possible. The forcing function for the fan is the mass flow rate.

The relation between the power, control variables, and forcing function is, in general,

$$P = P(u, M, f) \quad (4)$$

where, for the fan,

- $u_1 = C =$ inlet vane setting (10% to 100%)
- $u_2 = R =$ outlet damper resistance (0% to 100%)
- $M =$ speed mode (1, 2, or 3)
- $f = m =$ mass flow rate.

There appear to be two independent control variables, C and R . However, these are related to the mass flow rate. For any desired flow rate and a given inlet vane setting, C , there is only one possible damper resistance, R , that will allow the desired flow rate to be obtained. This is given mathematically by a constraint equation of the form:

$$m = m(C, R) \quad (5)$$

The presence of the constraint equation (Equation 5) reduces the number of degrees of freedom. In this situation, the resistance, R , is taken as a dependent variable—dependent on the flow, m , and the inlet vane setting, C . The problem then can be phrased as minimizing the power, P , with respect to the inlet vane setting, C , for each mode, M , and providing a desired flow rate.

The relations given by Equations 4 and 5 were formulated as bi-quadratic equations following Braun et al. (1987). The power was written in the following form:

$$P = a_0 + a_1 C + a_2 C^2 + a_3 m + a_4 m^2 + a_5 C m \quad (6)$$

where the coefficients a_0 to a_5 are determined for each M . The optimization of the power in Equation 6 with respect to C at a given flow rate, m , may not yield a physically realizable condition. The optimal C may not be within the physical bounds. Further, even if the optimal C does lie within the allowable range, the necessary damper setting, R , also may not be physically realizable.

The constraint equation for flow rate in terms of the damper setting, R , and the inlet vane setting, C , was formulated as

$$m = b_0 + b_1 C + b_2 C^2 + b_3 R + b_4 R^2 + b_5 C R \quad (7)$$

The constraint equation (Equation 7) is used in conjunction with the optimization of the power to determine the combinations of R and C that will produce the desired flow rate, m .

The coefficients for each mode M were determined experimentally. The fan was operated at many combinations of C and R over their respective ranges for each of the three modes. Over the range of operation, the measured power agreed with that given by Equation 6 within $\pm 2\%$. The measured flow rate agreed with that given by Equation 7 within $\pm 10\%$.

The minimum power for each load is determined by differentiating the power with respect to C at a constant flow rate in Equation 6 and setting the derivative to zero. This yields the control strategy for optimal power consumption. The relationship is

$$\frac{\partial P}{\partial C} = a_1 + 2 a_2 C + a_5 m \quad (8)$$

The second derivative of power with respect to C is

$$\frac{\partial^2 P}{\partial C^2} = 2 a_2 \quad (9)$$

The coefficients a_2 for all three modes were found to be less than zero. Thus, the condition given by Equation 8 would yield the maximum, not the minimum, power consumption. This means that the solution for minimum power consumption lies on a boundary of the surface. To determine the minimum power at a desired flow rate, Equation 6 must be evaluated for the two conditions of C equal to 10% and 100%. Further, if the value of R at the conditions of C of 10% and 100% is out of the range of 0% to 100%, the power at the limits of R must then be determined.

The relationship between power, inlet vane setting, and damper resistance is shown in Figure 2 at a flow rate of 2.6 kg/s. The value of the resistance, R , must be greater than zero and thus there are no possible values in the middle of the figure (for inlet vane settings between 40% and 95%). This figure shows that for this particular flow rate, the minimum power consumption is obtained at $C = 10\%$. The corresponding R that yields the desired flow is 46%.

The minimum power consumption for the fan for all three modes is shown in Figure 3. For each mode, the power increases slowly with increasing flow rate. There are jumps in power at the change from one mode to the next. The corresponding control strategy for the inlet vanes and damper is shown in Figure 4. The optimal control strategy to follow as the flow increases from the lowest value at any mode is to set the inlet vanes to the lowest setting ($C =$

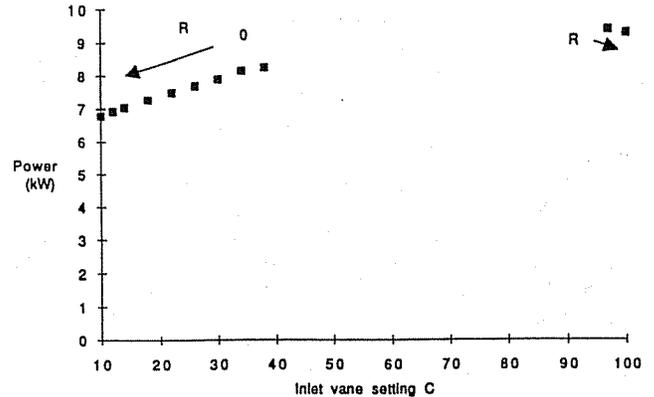


Figure 2 Fan power as a function of C and R at an airflow rate of 2.6 kg/s

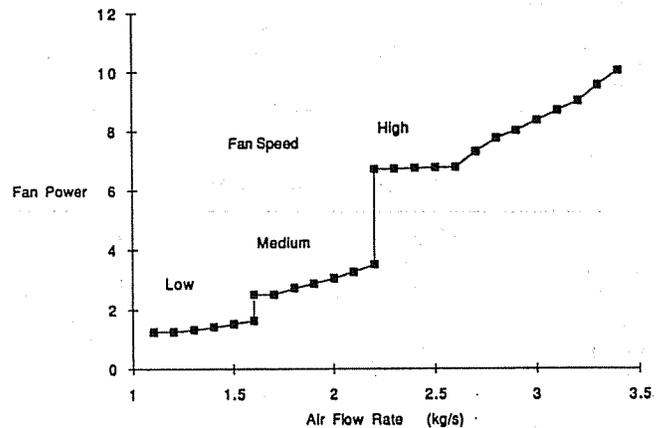


Figure 3 Minimum fan power as a function of airflow rate

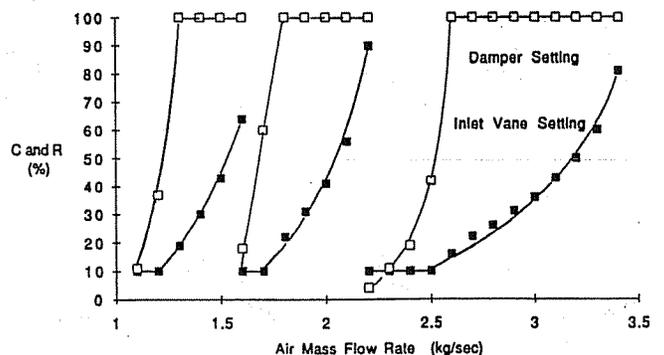


Figure 4 Control parameters C and R as functions of flow rate

10%) and increase the damper resistance from low ($R = 0\%$) to high ($R = 100\%$) to increase the airflow. When R is at its maximum value, the inlet vanes are then opened. When both damper resistance and inlet control vanes are at maximum values, a change to the next mode is made.

Figure 4 shows that the control relations between C and R are not linear with respect to flow. The nonlinearities are due to the constraints imposed on the damper and inlet vane control. Figure 4 also shows that the optimal mode selection is to always use the lowest-speed mode until that mode can no longer meet the flow requirement. The shift to the next highest mode is then required.

This procedure has determined the optimal control strategy for the fan alone. This strategy produces the minimum electrical energy consumption necessary to provide a given flow rate. It does not consider any trade-offs between any of the other energy uses in the system. These trade-offs will be considered in the following sections.

Optimization of Furnace Operation

The fuel energy required for the furnace is a function of one continuous control variable and a forcing function. The continuous control variable is the temperature of the circulating hot water, T_H , which can be varied over the range of 40°C to 90°C . The forcing function is the thermal load, Q_L , required by the furnace. This relationship is expressed as

$$Q_F = Q_F(T_H, Q_L) \quad (10)$$

There are two additional constraint relations that are appropriate. One is the energy balance relation between the load, temperature, and airflow rate, given by

$$Q_L = m c_p (T_o - T_i) \quad (11)$$

where T_o is the temperature of the air leaving the coil and T_i is the inlet temperature to the heating coil. In the experiments performed on this facility, the inlet temperature was maintained constant.

An additional constraint is the heat transfer relation for the heating coil. This can be expressed in terms of the heat exchanger effectiveness, which is defined as

$$\epsilon = (T_o - T_i) / (T_H - T_i) \quad (12)$$

The effectiveness is, in general, a function of the mass flow rate of air through the coil. During these tests, the water flow rate was held constant. Equations 11 and 12 can be combined to express the dependency of the load, Q_L , on the airflow rate, m , and coil temperature, T_H , as

$$Q_L = \epsilon m c_p (T_H - T_i) \quad (13)$$

The load is a function only of the airflow rate and the coil temperature for a fixed inlet temperature. This is expressed as

$$Q_L = Q_L(m, T_H) \quad (14)$$

The minimization of the furnace energy is determined by minimizing the energy using Equation 10 subject to the

constraint equation (Equation 14) for the load. Experiments taken on the furnace yielded the following functional relations for Equations 10 and 14:

$$Q_F = c_0 + c_1 T_H + c_2 Q_L \quad (15)$$

and

$$Q_L = d_0 + d_1 T_H + d_2 m \quad (16)$$

The forms for these relations are less complex than those for the fan due to the physical mechanisms involved. The relation between the furnace energy requirement and the load is basically linear. There would be no effect of the temperature of the hot water on the fuel requirement if the piping losses and efficiency were constant. For this furnace, an increased water temperature increases the losses from the pipes and reduces furnace efficiency. Over the range of the experiments (40°C to 90°C), these effects are relatively small and can be approximated by a linear relation.

The minimization of furnace energy with respect to water temperature at a desired load is given by differentiating Equation 15 and setting the differential to zero. The form of Equation 15 does not yield a minimum since the first derivative (c_1) is a constant. Thus, the optimal solution lies on a boundary; the optimal water temperature is either 40°C or 90°C . Since the coefficient a_1 is positive, the furnace energy is minimized at the lowest water temperature that can meet the load. This is physically reasonable, since the lowest water temperature minimizes the losses and maximizes furnace efficiency.

The resulting control laws for the furnace are shown in Figure 5. The optimal strategy is to maintain the water temperature at the lowest value (40°C) and vary the flow rate to meet the load. When the flow rate reaches its maximum value, further increases in load are met by increasing the water temperature.

SYSTEM OPTIMIZATION

The minimization of the energy consumption for the fan and furnace separately does not yield the minimum energy consumption of the two in combination. Minimizing the total energy requirement for the two units requires that the trade-off between fan power and furnace energy be considered. Increased flow rate reduces the necessary hot water temperature and lowers the furnace fuel requirement.

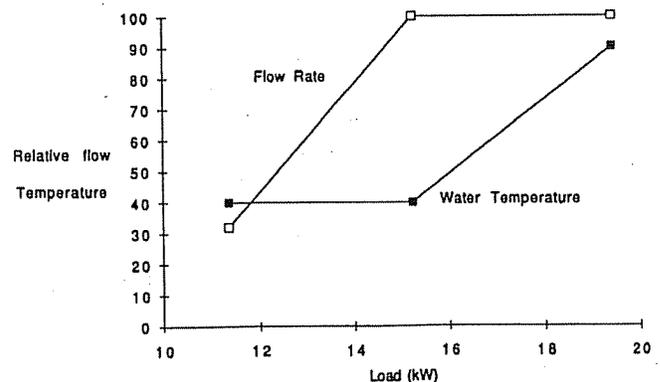


Figure 5 Control strategy for the furnace as a system

However, increased flow rate increases the fan energy required. The optimal strategy minimizes the sum of these two energy terms.

Two approaches will be used to determine the optimal strategy for the system. First, the models for the fan and furnace separately will be combined to determine an optimal strategy for the system as a whole. Second, a model for the system of the fan and furnace together will be developed and the optimal strategy for the system determined.

The fan energy is electric power and the furnace energy is natural gas. The values of these two forms are different; in general, electricity has a higher value than gas. This is indicated by a weighting factor, w , that is typically between two and three. The total energy for fan plus furnace energy is

$$E = wP + Q_F \quad (17)$$

Optimization of Fan and Furnace Using Component Strategies

The functional relationships for the fan and furnace separately establish that there are two independent control variables—inlet vane setting, C , and water temperature, T_H . The forcing function is the load, Q_L . The total energy is then a function of these variables as follows:

$$E = E(Q_L, C, T_H) \quad (18)$$

The minimization can be simplified, since the fan power does not interact with the thermal load. From Equation 6, fan power is a function of flow rate and inlet vane setting only. For any required flow rate, the fan power can be minimized without affecting the thermal load. Equation 18 can be simplified to minimize the total energy, E , using a relation of the form

$$E = E(Q_L, T_H) \quad (19)$$

The process of determining the optimal strategy for a given load, Q_L , involves applying an optimization method for finding the values of T_H that minimize the total energy over the range of 40°C to 90°C. For each water temperature, the following calculations must be performed:

- Determine the required flow rate from the constraint equations (Equation 16 or Equation 13).
- Determine the optimal inlet vane setting, C , from Figure 4 or the corresponding equations, and determine the power consumption from Equation 6.
- Determine the furnace requirement at that flow and water temperature from Equation 15.
- Determine the total energy, E , using the fan power and furnace energy relations.

The minimum total energy consumption is shown in Figure 6 as a function of flow rate. There are three segments of the curve corresponding to each of the three fan-speed modes.

The corresponding optimal strategy for water temperature and airflow rate is shown in Figure 7. The strategy is to operate at the low limit of airflow (1.1 kg/s) and vary the temperature as long as the load can be met. When the load increases and the low flow rate and maximum water

temperature (90°C) are insufficient, the strategy is then to maintain the maximum water temperature and increase the airflow rate. This strategy is exactly the opposite of the strategy for the furnace alone, as shown in Figure 5.

The strategy of first increasing the water temperature and then increasing the airflow rate implies that the fan power is more critical than the furnace energy. This is shown in Figure 8, where the furnace energy, fan power, and total energy are given as a function of water temperature at a given load. It is seen that the slope of furnace energy with temperature is less than that of the fan power.

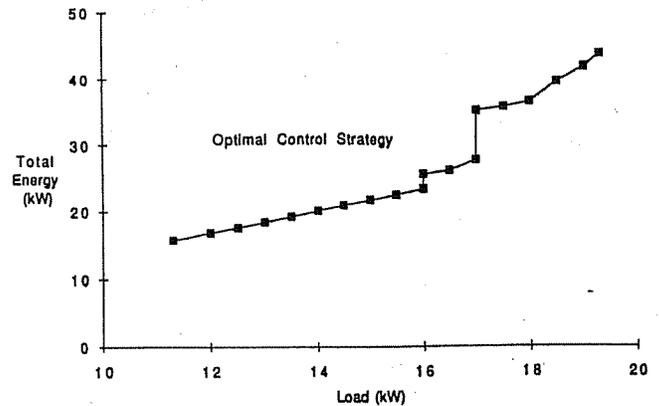


Figure 6 Minimum total energy consumption for fan and furnace

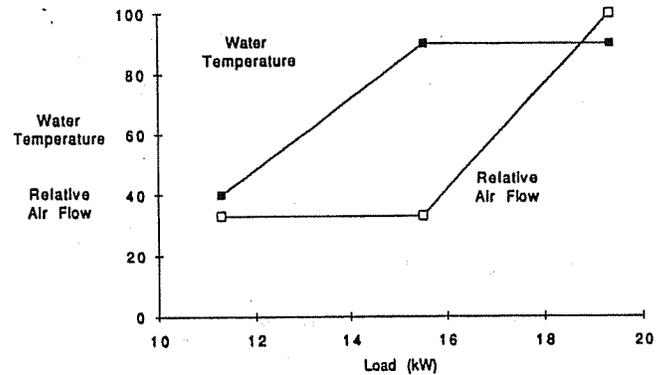


Figure 7 Optimal control strategy for fan and furnace

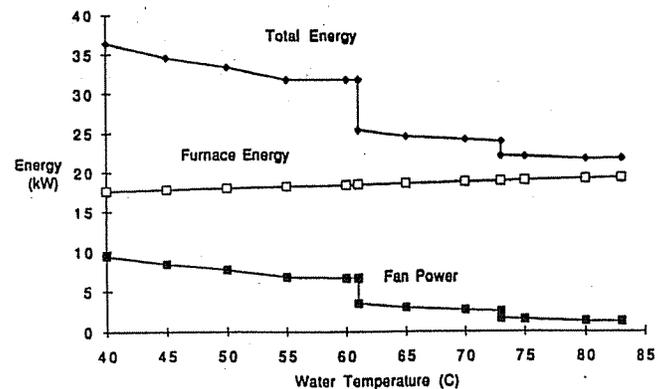


Figure 8 Furnace, fan, and total energy as a function of furnace water temperature

Even though the magnitude of the furnace energy is greater than that of the fan, it is the variations of energy with the control variable that are important.

Optimization of Fan and Furnace as a System

The optimization described earlier uses the component relations to determine the optimal strategy. As discussed by Braun et al. (1987), it is also possible to develop the optimal strategy considering the system as a whole. This requires developing a relation for the total energy, E , as a function of the control variables and forcing function. As with the component-based optimization, the control for the mass flow rate is to operate the fan at minimal power consumption at each flow, which presumes that this control strategy has been determined.

The total energy is then a function of the single control variable, T_H , and the forcing function, Q_L . The relationship is

$$E = E(T_H, Q_L) \quad (20)$$

Equation 20 is of the same form as Equation 19. The difference is in concept. In the component-based optimization, there are several constraint equations and relations. For the system-based optimization, there is only one equation, represented by Equation 20. The specific form of the relation is

$$E = f_0 + f_1 T_H + f_2 T_H^2 + f_3 Q_L + f_4 Q_L^2 + f_5 T_H Q_L \quad (21)$$

The coefficients in Equation 21 were determined empirically. The previously developed relations for the fan power and furnace energy were used to generate values of the total energy, E , for each fan mode. The coefficients for the total power were then obtained by linear regression. The optimal strategy was then determined by equating the derivative of E with respect to T_H to zero to find the control relation,

$$T_H = (-f_1/2f_2) + (-f_5/2f_2) Q_L \quad (22)$$

subject to the constraint that

$$40^\circ\text{C} < T_H < 90^\circ\text{C} \quad (23)$$

The control strategy for the lowest fan-speed mode yields a variation in hot water temperature from 40°C at the lowest flow rate to 90°C at the same flow rate and then an increase in flow rate to the maximum for the mode. The control relations for the second and third modes yield operation at the temperature limit of 90°C . These relations are plotted in Figure 9.

Comparison of Figure 9 with the control strategy determined from the component-based approach (Figure 7) shows that the strategies determined by both approaches are the same. This establishes that it is possible to consider the fan and furnace as a system using the relations suggested by Braun et al. (1987) and determine the optimal strategy.

The reduction in energy consumption between optimal and non-optimal strategies can also be illustrated. A possible non-optimal strategy would be one based on the

optimal strategy for the furnace alone, as given in Figure 5. Following this strategy, the water temperature would be maintained at the lowest possible temperature to maximize furnace efficiency, and the airflow rate would be increased to meet increasing loads. The power requirements under this strategy are compared to those with the optimal strategy in Figure 10. It is seen that the power requirements are the same at the lowest and highest loads but are significantly different at intermediate loads. The use of the nonoptimal strategy can increase the power requirement by up to 50% at loads of 14 kW to 16 kW. This illustrates the importance of considering all of the components of the system when developing the optimal strategy.

CONCLUSIONS

The methodologies developed for the optimization of centralized facilities (Braun et al. 1987; 1989a, b) have been applied to a simple system consisting of a fan and a gas-fired furnace. The optimization procedure has been applied first to each of the two components separately and then to the two in combination. The optimal strategies have then been determined.

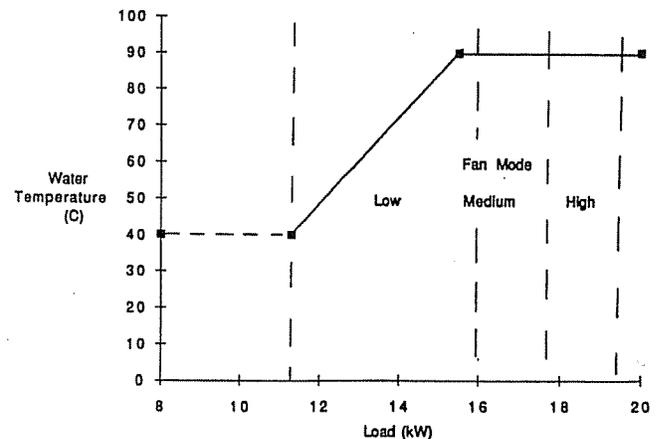


Figure 9 Optimal control strategy for water temperature using system-based methodology

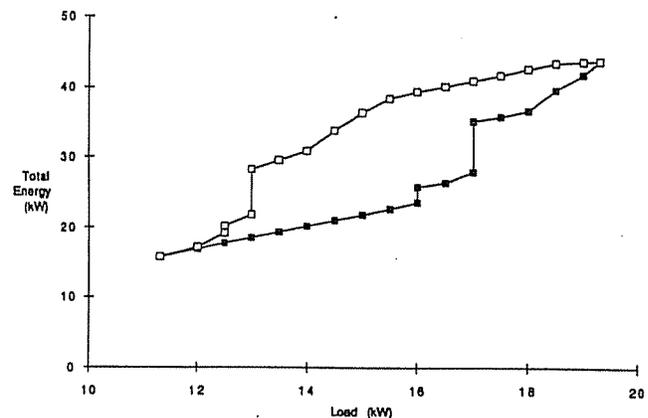


Figure 10 Comparison of total energy consumption using optimal control strategy for the furnace alone compared to that for the fan and furnace as a system

Considering each unit separately produces control laws that may or may not apply to the system as a whole. For the fan, the optimization process produces the conditions for minimum fan power at a given flow rate. This is a useful result when considering the entire system. For any desired flow rate through the system, it is desirable to minimize fan power.

In contrast to the fan result, the process of minimizing the furnace energy is not applicable to the system as a whole. Minimum furnace energy is obtained by operating the hot water loop at the lowest possible temperature. This result does not account for the fan power. Minimum total energy for the entire system is obtained by operating the furnace at the highest possible water temperature. For the system, the increase in fan power with flow is more costly than the decrease in furnace energy with flow. This demonstrates the need to consider all aspects of the system when performing an optimization procedure.

For the two components considered here, the optimal strategy is always at a limit. The constraints on temperature, vane, and damper settings do not allow operation over all values of the variables. Thus, it is necessary to evaluate the operation at the limits to determine optimal operation.

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