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Field Validation of a Short Time Step Model for Vertical Ground-Loop Heat Exchangers

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ABSTRACT

The field validation of a short time step temperature response factor model is presented using actual operational data from an elementary school building in Lincoln, Nebr. The short time step model is used to predict hourly temperature changes of vertical ground-loop heat exchangers as used in ground-source heat pump applications. The model was based on an analytically validated numerical borehole model that simulates the heat transfer in and around the ground heat exchanger, using its thermal response to unit heat pulses. The short time step model was cast as a modular component and used to simulate the thermal behavior of the borefield using the TRNSYS environment. In order to assess the potential influence of errors associated with predicted temperatures, a series of sensitivity analyses are also presented. The sensitivity analyses focus on the impact of the heat pump entering fluid temperature on the system energy consumption, since the uncertainty in the predicted temperature is expected to have a corresponding uncertainty in the system energy use. System energy consumption, based on predicted and measured heat pump entering fluid temperatures, is compared and discussed.

INTRODUCTION

The heat transferred by the ground-loop heat exchanger of a ground-source heat pump (GSHP) system to the soil varies continuously due to variations in the space loads of the building it serves. As a result, corresponding variations are observed in the temperature of the heat transfer fluid that circulates in the ground heat exchanger. Since the temperature of the heat transfer fluid returning from the ground loop impacts the coefficient of performance (COP) of the heat pump directly, the short time step (hourly) temperature fluctuations thus affect the overall GSHP system performance. In order to be able to assess the system performance fluctuations, knowledge of heat pump entering fluid temperatures, along with information on system energy consumption and system energy demand, is required in hourly or less time intervals. The impact on system economics can be significant, especially for commercial buildings that use hybrid ground-source heat pump systems (Yavuzturk and Spitler 2000; Ramamoorthy et al. 2001) and/or that are on time-of-day electricity rates. A short time step model of the ground heat exchanger may also be useful to replace approximations for peak temperatures, which, in some loop design methodologies, are currently estimated as the response to a single load with a user-specified duration.

The accurate and efficient prediction of the short time step behavior of a ground loop heat exchanger is therefore desirable for detailed building energy analysis and may be useful for the design of hybrid GSHP systems. Yavuzturk and Spitler (1999) developed a short time step model based on the temperature response factors of Eskilson (1987). Although the short time step model uses an analytically validated numerical model of the ground heat exchanger borehole (Yavuzturk et al. 1999), a comparison of model predictions to actual field data is desirable in order to assess further the validity of the model. This paper focuses on field validation of the short time step model developed for vertical ground-loop heat exchangers by Yavuzturk and Spitler (1999).

Unfortunately, suitable field data for model validation are scarce. A suitable data set would include a high-quality independent measurement of the ground thermal properties at the site. In addition, monitoring of the system, which would include accurate measurements of the loop flow rate and inlet

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and outlet temperatures, would have to commence at the beginning of the system operation. Although not entirely satisfactory, measured data are available for the operation of a vertical ground-coupled heat pump system from Maxey Elementary School in the Lincoln, Nebr., school district. The data set is fairly recent (Carlson 1998; Shonder 1999) and was used by McLain and Martin (1999) to investigate the validity of a new ground-source heat pump subroutine that is part of the DOE-2 building simulation software. Shonder et al. (2000) utilized the Maxey field data to calibrate an energy use model of the Maxey elementary school to form a benchmark with which four commercially available vertical borehole heat exchanger design software were compared. The data set consists of instantaneous ten-minute measurements of entering and exiting heat transfer fluid temperatures and flow rates on the loop for a period of several years. The building was placed in service in August 1995. The collection of ten-minute operating data started in November 1995 (Carlson 1998). For the field validation of the short time step model by Yavuzturk and Spitler (1999), data from 12:00 a.m. on January 1, 1996, to 11:55 p.m. on December 31, 1996, were used.

SHORT TIME STEP RESPONSE FACTOR MODEL

The short time step ground-loop heat exchanger model (Yavuzturk and Spitler 1999) is based on dimensionless temperature response factors, g-functions of Eskilson (1987), which are unique for various borehole field geometries. The short time step temperature response factor model was cast as a TRNSYS (Klein et al. 1996) component model and includes a flexible load aggregation algorithm that significantly reduces computing time by averaging less recent building loads over a user-defined block of hourly loads and treating it as a single pulse while more recent hourly loads are not averaged. For the simulations presented here, a load aggregation block of 730 hours is used. The parameters of the component model are provided in Figure 1.

For each time step, the entering fluid temperature to the ground-loop heat exchanger and the mass flow rate of the heat transfer fluid are inputs into the model. The component model computes the average borehole temperature and the exiting fluid temperature of the loop (entering fluid temperature to the heat pump) using the model parameters that describe the ground-loop heat exchanger geometry, heat transfer fluid and ground thermal properties, and the g-functions. It should be noted that the fluid flow rate is a model variable and not a model output, although it is changing for each time step. Thus, for each time step the fluid flow rate does not change.

The average temperature of the borefield for each time step is determined by decomposing the time-dependent building loads profile into unit pulses. Each heat pulse is then superimposed in time using the corresponding g-functions. The following equation is used.

$$T_{borehole}(t) = T_{ground} + \sum_{i=1}^{n} \frac{(Q_i - Q_{i-1})}{2\pi k} g\left(\frac{t_n - t_{i-1}}{t_s}, \frac{r_b}{H}\right) \quad (1)$$

where

t	=	time, s
t_s	=	time scale = $H^2/9\alpha$
H	=	borehole depth, ft (m)
k	=	ground thermal conductivity, Btu/h·ft·°F (W/m·°C)
$T_{borehole}$	=	average borehole temperature, °F (°C)
T_{ground}	=	undisturbed ground temperature, °F (°C)
Q	=	step heat rejection pulse, Btu/h·ft (W/m)
r_b	=	borehole radius, ft (m)
i	=	index to denote the end of a time step t
α	=	ground diffusivity
g	=	dimensionless temperature response factor (g-function)

Yavuzturk and Spitler (1999) provide a detailed discussion of the development and use of the short time step g-functions and the load aggregation algorithm of the short time step component model.

IN-SITU GROUND THERMAL CONDUCTIVITY TEST

The thermal conductivity of the ground formation and the undisturbed far field temperature of the ground are significant parameters in the design and simulation of the thermal behavior of ground-loop heat exchangers. Austin et al. (2000) present a series of analyses that demonstrate the strong influence of the far field temperature and the thermal conductivity of the ground on the heat transfer between ground-loop heat exchangers and the surrounding soil formation.

In order to estimate the thermal conductivity of the ground formation at the Maxey site, a series of in-situ tests were conducted, using the testing and parameter estimation method suggested by Austin et al. (2000), on a test borehole at the Maxey borefield site. The analyses yielded an average undisturbed far field ground temperature of 54.34°F (12.41°C) and an effective thermal conductivity of 1.36 Btu/h·ft·°F

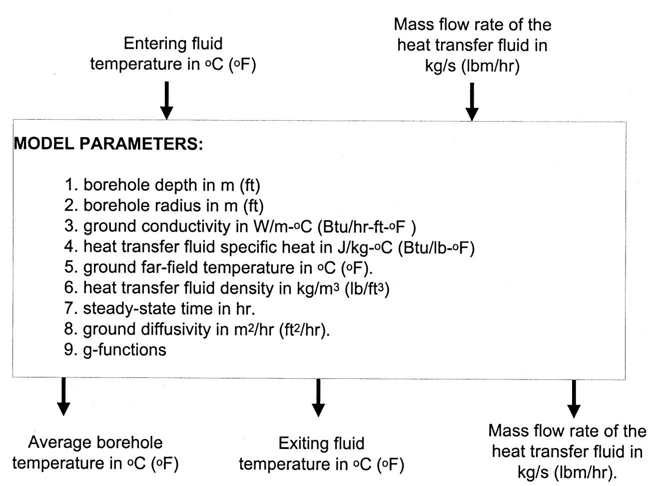

Figure 1 *Short time step ground-loop heat exchanger model component parameters.*

(2.353 W/m·K). These values are then used as input parameters in the validation of the short time step model.

DESCRIPTION OF THE EXAMPLE BUILDING SYSTEM

The Maxey Elementary School has about 70,000 ft^2 (6500 m^2) of floor area that is served by 54 heat pumps distributed throughout the single-story building. The borehole field of the ground-source heat pump system consists of 120 vertical boreholes arranged in a 10 × 12 rectangular configuration. Each borehole is about 240 ft (73.2 m) deep and has a diameter of 4.5 in. (114.3 mm). The ground-loop heat exchangers are 1 in. (25.4 mm) nominal high-density polyethylene U-tube pipes. The boreholes are spaced approximately 20 ft (6.1 m) apart from each other, center-to-center. The heat transfer fluid circulated in the U-tubes of the ground heat exchanger is a 22% aqueous propylene glycol solution. The heat transfer fluid circulation pumps are variable speed pumps and have a maximum rated capacity of 575 gpm (36.2 L/s). The boreholes are backfilled with a mixture of sand and fine gravel up to 10 ft (3 m) below the ground surface. A bentonite plug is used in the top 10 ft (3 m) to prevent surface water from contaminating the groundwater. Fluid flow to and from the borehole field is through buried horizontal piping that is connected to the circulation pumps and the HVAC system in the mechanical room of the building. Data acquisition on the system is accomplished in the mechanical room of the school building.

MONITORED FIELD DATA

The Maxey system is designed to maintain a minimum flow through the ground loop during the winter months, even when there may be only minimal or no heating demand. During the summer months, when the school building is used at minimal capacity due to the summer break, the circulation pumps are turned off when there is no demand on the borehole field. At these times, the entering fluid temperature is observed to drift above 70ºF (21.1ºC) during weekday nights and above 80ºF (26.7ºC) during weekends as the temperature sensors come into equilibrium with the air temperature of the mechanical room, where they are located. Figures 2 and 3 show the heat pump entering fluid temperature and the heat transfer fluid flow rate measured at the Maxey Elementary School during 1996. The temperature “spikes” in Figure 2 are representative of no or low fluid flow conditions.

The data collection on the Maxey system started in November 1995, although the system was placed in service about three months earlier. This time gap is significant, since there is no way of accurately adjusting for the unrecorded thermal disturbances in the ground during this time. However, in order to account for this thermal disturbance “history,” data from the same time period (from August 1996 to November 1996) are assumed to have occurred for the same period in 1995, for which data are missing. Although this approach is clearly not ideal, it seems to be the best approximation available under the circumstances.

Data acquisition was performed by the building energy management system. Unfortunately, data for a number of time intervals were lost or not obtained by the system. The cause of this is unknown, although building energy management systems often do not make the most reliable long-term building monitoring systems. The missing data were filled after the fact (Carlson 1998), using best engineering estimates for the time intervals in question. Nevertheless, reliability of the filled data points must be questioned.

In addition, the no or low-flow conditions on the system are somewhat problematic in the monitored field data. During some time intervals, the data set contains flow rates and flow rate changes without corresponding effects in the fluid temperatures. Also, there are significant temperature changes even though the fluid flow rate is zero. An example of such

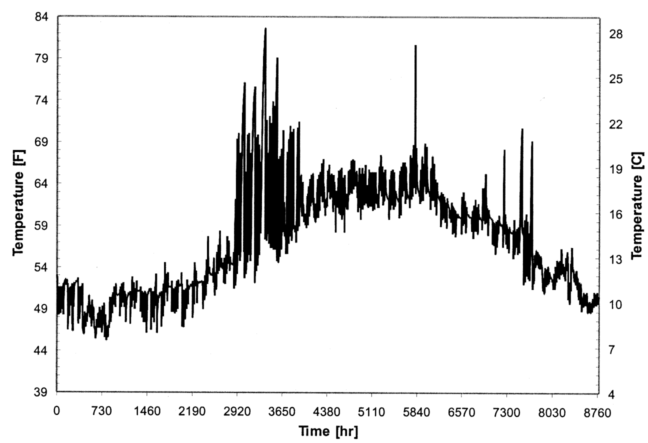

Figure 2 *Heat pump entering fluid temperatures from 12 a.m. January 1, 1996, to 11:55 p.m. December 31, 1996, plotted hourly.*

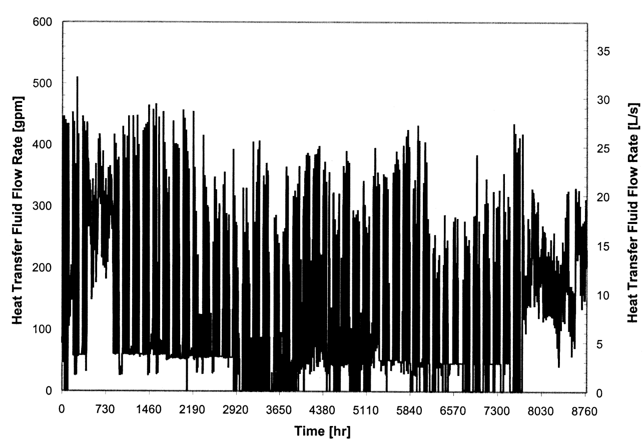

Figure 3 *Heat transfer fluid flow rate from 12 a.m. January 1, 1996, to 11:55 p.m. December 31, 1996, plotted hourly.*

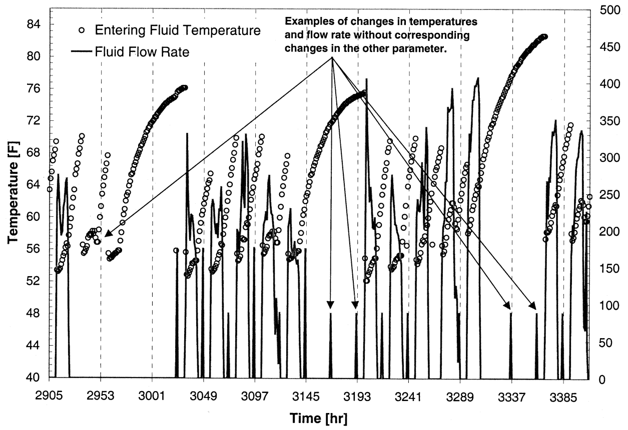

Figure 4 *Measured heat pump entering fluid temperatures and fluid flow rate plotted.*

inconsistencies is provided in Figure 4, where the measured heat pump entering fluid temperature and fluid flow rates are plotted for three weeks in May 1996. Specifically, there are a number of instances where the flow rate goes from zero to approximately 75 gpm for several hours, without any noticeable change in the entering fluid temperature. A number of significant changes in the entering fluid temperature are recorded between hours 2930 and 2970, although the flow rate is zero for this time interval. Arrows on Figure 4 show a few of these incidents. This behavior has not been satisfactorily explained, but it can be seen throughout the data set. It appears to happen at regular intervals and, on a purely speculative basis, it may be inferred that it is due to an error in the data acquisition, which is performed by the building energy management system.

In the analyses presented here, no measured temperatures are plotted for comparison with the predicted temperatures when the fluid flow rate is below 50 gpm (3.15 L/s). Although this "cut-off" point is admittedly somewhat arbitrary, examination of the data indicates that a flow rate of about 50 gpm (3.15 L/s) is the intended minimum flow through the ground loop during winter months. Therefore, there is some "legitimate" data down to about 50 gpm (3.15 L/s). Below that, the frequency of anomalous data increases significantly. For very low flow rates, less than 10 gpm (0.63 L/s), for which no significant temperature responses are observed, the flow rate into the ground-loop heat exchanger is set to zero. The removal of spurious and insignificant data allows for a better (cleaner) comparison between measured and predicted temperature responses. However, since no continuous set of experimental data was available, discontinuities in temperature predictions of the model were unavoidable.

Furthermore, the location of instrumentation for data acquisition at the site is not ideal. The mechanical room of the Maxey building is about 120 yd (110 m) away from the borehole field. The model predictions for the heat pump entering fluid temperature are temperature responses of the borehole field (exiting fluid temperatures from the borefield) and neglect heat transfer between the borefield and the mechanical room of the building. It is also highly likely that there would be heat gains/losses through horizontal piping buried at varying depths from the building to the field.

Refinement of the Short Time Step Component Model

The short time step component model, as originally developed, expected a constant fluid flow rate in the ground loop and a constant borehole thermal resistance. Since, in reality, the fluid flow rate is not constant during the operation of the ground-source heat pump system at Maxey, the short time step component model was modified to account for the variable borehole resistance.

The total thermal resistance of a borehole is estimated as the sum of thermal resistances due to the borehole backfill material, the pipe material, and the fluid flow.

$$R_{Total} = R_{Backfill} + R_{Convection} + R_{PipeConduction} \tag{2}$$

The backfill material thermal resistance is estimated using the approach of Paul (1996), which uses resistance shape factor coefficients. The backfill material thermal resistance is dependent on the position of the U-tube and its shank spacing in the borehole. Although it is very unlikely that all 120 boreholes at the Maxey school site have identical borehole/U-tube geometry, a constant backfill material resistance is nevertheless assumed for all boreholes for practical reasons. Similarly, a well-mixed backfill material with a constant thermal conductivity is stipulated. The thermal resistance due to the backfill material can then be calculated using Equation 3.

$$R_{Backfill} = \left[k_{Backfill}\beta_0\left(\frac{D_{Borehole}}{D_{Pipe}}\right)^{\beta_1}\right]^{-1} \tag{3}$$

β_0 and β_1 are resistance shape factor coefficients based on U-tube shank spacing. Paul's (1996) shape factor coefficients are based on experimental and finite element analysis of typical borehole and pipe geometry. Shape factor coefficients of β_0 = 17.44268 and β_1 = –0.605154 are suggested for U-tube pipe that is evenly spaced (centered) in the borehole. Shape factor coefficients of β_0 = 20.100377 and β_1 = –0.94467 are suggested for U-tube pipes that are 0.125 in. (3.2 mm) apart from each other (distance between pipe outer walls).

The thermal resistance due to high-density polyethylene pipe is determined using Equation 4.

$$R_{PipeConduction} = \frac{\ln\left(\frac{D_{out}}{D_{in}}\right)}{4\pi k_{Pipe}} \tag{4}$$

In order to account for the variations in the total borehole thermal resistance, the convective thermal resistance due to the fluid flow is adjusted for each time step considering the change in the flow rate of the heat transfer fluid. Since the variable fluid flow rate only impacts the inside convective heat transfer coefficient h_i, this adjustment is implemented by recomputing h_i for each time step in determining the borehole thermal resistance due to convection.

$$R_{Convection}(t) = [2\pi D_{in} h_{in}(t)]^{-1} \quad (5)$$

where

$$h_i(t) = \frac{\mathrm{Nu}(t) \cdot k_{Fluid}}{D_i} \quad (6)$$

The time-dependent Nusselt number in Equation 6 is determined based on the flow characteristics of the heat transfer fluid through the calculation of the Reynolds number.

$$\mathrm{Re}(t) = \frac{v(t)D_i}{\nu} \quad (7)$$

This equation for the Reynolds number may be recast in the following form to include the volumetric flow rate of the heat transfer fluid:

$$\mathrm{Re}(t) = \frac{\dot{V}(t)D_i}{A_i \nu} = \frac{4\dot{V}(t)}{\pi D_i \nu} \quad (8)$$

For laminar flow conditions (Re < 2300), it is sufficient to use a constant Nusselt number of 4.36, when a uniform surface heat flux and fully developed flow are assumed in the ground loop (Incropera and Dewitt 1980). For fully turbulent flow (Re > 10,000), the Dittus-Boelter correlation is used, as given by Incropera and Dewitt (1980):

$$\mathrm{Nu}(t) = 0.023\mathrm{Re}(t)^{0.8}\mathrm{Pr}^{0.35} \quad (9)$$

The exponent of the Prandtl number ($\mathrm{Pr} = \upsilon/\alpha$) is approximated with 0.35 for both heating and cooling modes.

For the transition region between fully turbulent and laminar flow (2300 < Re < 10,000), a correlation proposed by Gnielinski (Incropera and Dewitt 1980) is used.

$$\mathrm{Nu}(t) = \frac{(f/8)\mathrm{RePr}}{1.07 + 12.7(f/8)^{0.5}(\mathrm{Pr}^{0.67} - 1)} \quad (10)$$

where f is the friction coefficient and is determined using Petukhov's relationship as given by Incropera and Dewitt (1980):

$$f = [0.79\ln(\mathrm{Re}) - 1.64]^{-2} \quad (11)$$

RESULTS OF MODEL VALIDATION

Short Time Step Model Predictions

A comparison between the predicted and experimental hourly heat pump entering fluid temperatures is provided in Figures 5 through 10 for the months of January, March, August, October, November, and December 1996. These months illustrate a range of different conditions. The hourly system simulations are performed using TRNSYS (Klein et al. 1996) with the short time step temperature response factors for a 10 × 12 borehole field with a spacing of 20 ft (6.1 m) and depth of 240 ft (73.2 m). The following additional parameters are specified for the short time step component model:

a. Borehole radius r_b = 0.1875 ft (0.0572 m)
b. Thermal conductivity of the ground formation, k_{soil} = 1.36 Btu/h·ft·°F (2.353 W/m·K)
c. Ground far field temperature T_{ff} = 54.34°F (12.41°C)
d. Volumetric heat capacity of the ground $(\rho c_p)_{Soil}$ = 32.2 Btu/ft^3·°F (2160.2 kJ/m^3·K)
e. Number of boreholes NB = 120
f. Shank spacing between U-tube pipes (distance between the outer pipe walls of the two pipes) x = 1 in. (25.4 mm)
g. Thermal conductivity of the HDPE pipe k_{pipe} = 0.226 Btu/h·ft·°F (0.391 W/m·K)
h. Nominal diameter of the HDPE pipe $D_{Nominal}$ = 1 in. (25.4 mm) and outside diameter D_o = 1.315 in. (33.4 mm)
i. Load aggregation block = 730 hours

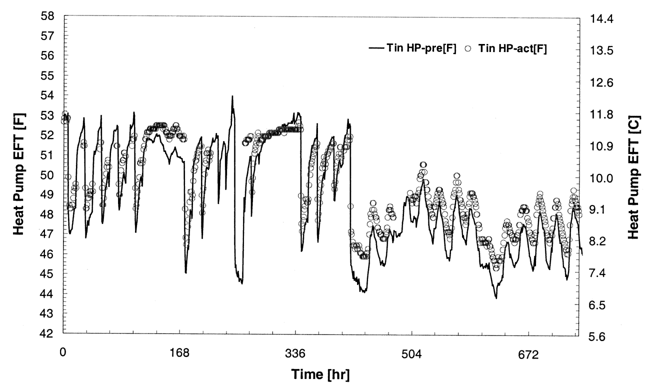

Figure 5 *Comparison of hourly heat pump entering fluid temperatures. Predicted vs. experimental for the month of January 1996.*

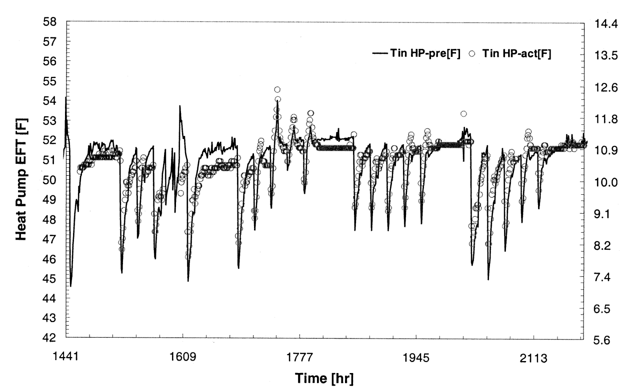

Figure 6 *Comparison of hourly heat pump entering fluid temperatures. Predicted vs. experimental for the month of March 1996.*

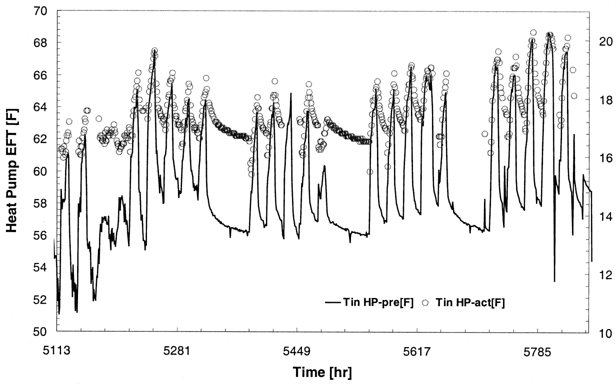

Figure 7 *Comparison of hourly heat pump entering fluid temperatures. Predicted vs. experimental for the month of August 1996.*

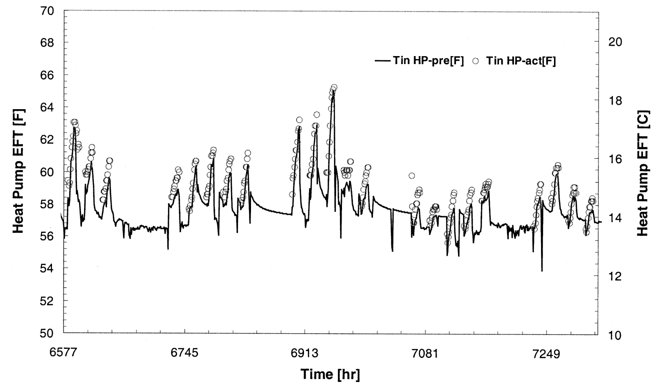

Figure 8 *Comparison of hourly heat pump entering fluid temperatures. Predicted vs. experimental for the month of October 1996.*

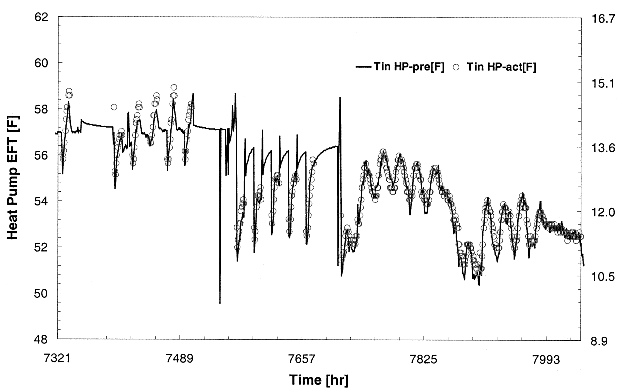

Figure 9 *Comparison of hourly heat pump entering fluid temperatures. Predicted vs. experimental for the month of November 1996.*

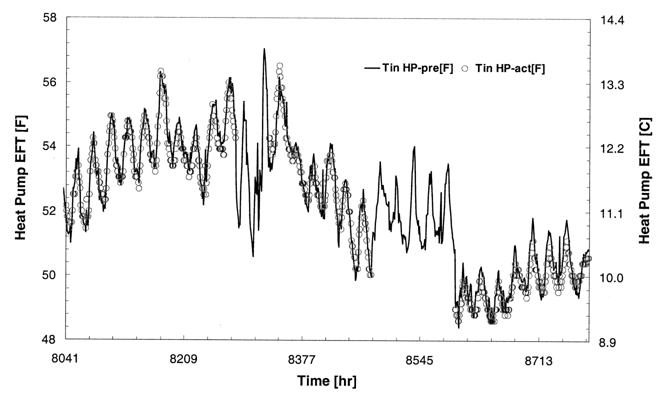

Figure 10 *Comparison of hourly heat pump entering fluid temperatures. Predicted vs. experimental for the month of December 1996.*

A comparison of the model predictions and measured entering fluid temperatures in Figures 5 through 10 shows reasonable agreement. The maximum deviation of the temperature predictions is observed when the heat transfer fluid flow rate shows significant discontinuities in the data set. This is especially noticeable for the month of August (Figure 7) when the school building is essentially shut down for the summer break. Presumably, this is due to the problematic data at low flow rates and the approximations made to deal with low flow fluid rates. The best agreement between the predicted and measured temperatures is observed when the fluid flow rate in the loop is relatively continuous, as in the second month of November and the month of December.

The "spikes" in the temperature predictions are due to peculiarities in the data set, where fluid flow changes are measured for some time intervals without corresponding temperature responses and/or changes. Figures 7, 8, and 9 best illustrate such occurrences.

ENERGY CONSUMPTION SENSITIVITY ANALYSIS

A series of sensitivity analyses are performed to assess the influence of errors associated with the predicted heat pump entering fluid temperature. The sensitivity analyses focus on the impact of the entering fluid temperatures on the system energy consumption, since the uncertainty in the predicted temperature has a corresponding uncertainty in the system energy consumption. Two different effects are considered for the worst and best case months, August and December 1996, respectively.

1. The hourly energy consumption is computed using the predicted and the actual heat pump entering fluid temperatures. For the months of August and December 1996, the relative error at the hour when the maximum system energy consumption occurs can then be calculated. It is interesting to assess the effect of the model on the predicted peak electricity consumption.

2. The total energy consumption is computed using the predicted and the actual heat pump entering fluid temperatures. The monthly energy consumption for August and December 1996 is compared. The cumulative energy consumption for a month is of interest for determining the electrical cost. The comparison for the months of August and December will show the impact of the model for the months with the best and worst predictions.

Sensitivity Based on Maximum Energy Consumption

A direct comparison between the energy consumption computed using the actual and predicted heat pump entering fluid temperatures can be made when the system energy consumption is computed using the predicted and the measured heat pump entering fluid temperatures. Figures 11 and 12 show the results of this analysis for the worst and best case months. As can be seen from these figures, the predicted energy consumption is minimally affected by the errors in the entering fluid temperature prediction.

For the month of August, the maximum energy consumption is recorded for hour 5799 of the year (Figure 11). At this time step, the predicted energy consumption of the heat pumps is only about 0.5% higher than the energy consumption determined using the measured heat pump entering fluid temperature.

For the month of December, the maximum energy consumption is recorded for hour 8282 of the year (Figure 12). For this time step, the predicted energy consumption of the heat pump is about 1.4% higher than the energy consumption determined using the measured heat pump entering fluid temperature. However, it should be noted that data for hour 8282 of the year are "filled" data. Therefore, a second maximum is also investigated, which occurs at hour 8501 of the year. The predicted energy consumption at this hour is about 0.9% lower than the actual energy consumption.

The analyses thus indicate that the errors, for both the best and worst case months, are within acceptable limits. This error envelope of 0.5% to 1.0% was consistently observed also for other months of the year for all time steps containing actual measured data (not "filled" data).

Sensitivity Based on Total Error in Energy Consumption

In order to estimate the effect of uncertainties in the ground-loop heat exchanger model on the heat pump energy consumption estimates, the total energy consumptions computed using the actual and predicted heat pump EFTs are compared for the months of August and December 1996. Again, the comparison for the months of August and December will demonstrate the effects of the model for the best and worst months. The results of the analyses are provided in Table 1.

The results show an excellent agreement in the predicted and actual energy consumption for the heat pump for the month of December. Because of the relatively continuous operating data for this period, the error is only about 0.4% and can safely be considered very small. As expected, the error for

TABLE 1
Comparison of Total Error on Maxey Elementary School Energy Consumption for the Months of August and December 1996

	Months	
	August 1996	**December 1996**
Total Energy Consumption (predicted) [kWh]	8338	25,457
Total Energy Consumption (actual) [kWh]	8665	25,555
Deviation [%]	3.8	0.4

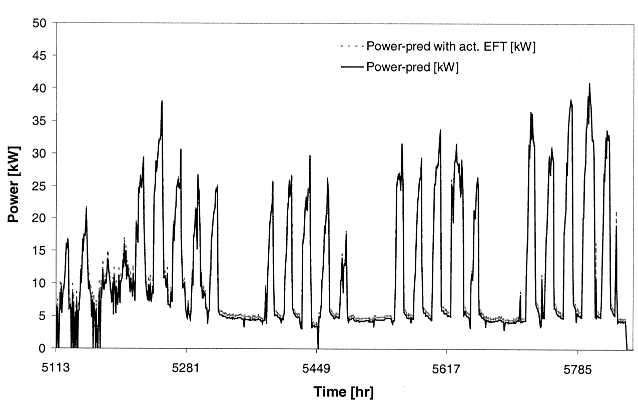

Figure 11 *Comparison of hourly heat pump energy consumption for the month of August 1996, considering predicted and actual heat pump EFT.*

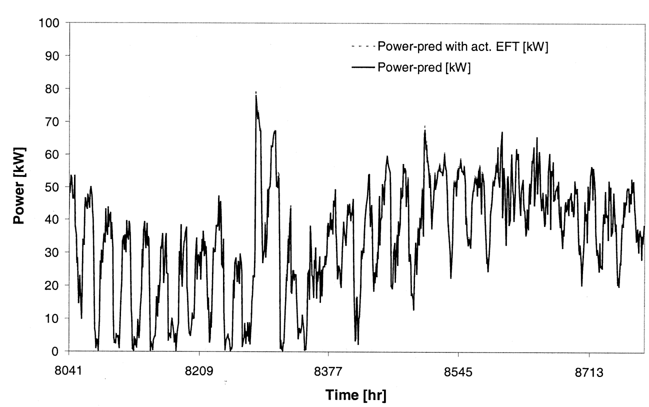

Figure 12 *Comparison of hourly heat pump energy consumption for the month of December 1996, considering predicted and actual heat pump EFT.*

the month of August is significantly higher (about 3.8%) than for the month of December. This relatively high percentage of deviation is most likely due to discontinuities in system operating data for this month.

DISCUSSION AND CONCLUSIONS

The comparisons presented here show that the goodness of the agreement between the predicted and the actual heat pump entering fluid temperature predictions depends on the continuity of the experimental data. The maximum deviations in the entering fluid temperatures are observed during periods where the heat transfer fluid flow was either frequently interrupted (see Figures 7, 8, and 9, corresponding to the months of August, October, and early November 1996) or a series of apparently spurious data were reported, coinciding with low flow rates. The best agreements can be found during the winter months (second half of November and December 1996), where there is a relatively continuous set of data, all at moderate to high flow rates.

In addition, there will be some heat loss through the horizontal piping runs between the building and the borefield. The errors that may be associated with horizontal piping are, however, difficult to estimate since the depth of the horizontally buried piping from/to the borefield is not constant and the surrounding temperature changes throughout the year. Accordingly, the magnitude of the impact of ambient environmental conditions will be different for different portions of the horizontal piping at different times of the year. The heat losses through the horizontal piping are most significant when the system is switched on and off, as is done during the summer months of system operation. This may also help to explain the relatively high deviation between the total energy consumption in August 1996 as compared to the consumption in December 1996.

In conclusion, it should be noted that additional field validation of the short time step model is highly desirable. An ideal field test would include

- an independent in-situ measurement of the ground thermal properties,
- carefully calibrated and monitored data acquisition, which would include, at the least, measurement of the entering and exiting fluid temperatures near to the borehole field and measurement of the fluid flow rate,
- continuous data collection from the beginning of the building operation, and
- well characterized borehole geometry, backfill material properties, and heat transfer fluid properties.

Regardless of the shortcomings in the experimental data set, the comparisons show reasonable agreement between the short time step model and the measured data. The sensitivity analyses indicate that the system energy consumption prediction is only slightly affected by errors in the predicted heat pump entering fluid temperature of the magnitude of those encountered here.

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NOMENCLATURE

α	=	thermal diffusivity, ft^2/h (m^2/s)
A	=	area, ft^2 (m^2)
B	=	spacing between ground heat exchanger boreholes, ft (m)
D	=	diameter, ft (m)
f	=	friction coefficient
g	=	g-function
H	=	borehole depth, ft (m)
h	=	convective heat transfer coefficient, Btu/h·ft^2·°F (W/m^2°K)
k	=	conductivity, Btu/h·ft^2·°F (W/m^2·°C).
Nu	=	Nusselt number
υ	=	kinematic viscosity, ft^2/h (m^2/h)
Pr	=	Prandtl number
Q	=	heat transfer rate, Btu/h·ft (W/m)
r	=	radius, ft (m)
R	=	thermal resistance, °F per Btu/h·ft (°C per W/m)
Re	=	Reynolds number
T	=	temperature, °F (°C)
t	=	time, h or sec
v	=	velocity, ft/s (m/s)
$\dot{V}$	=	volumetric flow rate, ft^3/h (m^3/h)

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DISCUSSION

Dennis Knight, CEO, Engineering Technology, Inc., Mt. Pleasant, S.C.: In your actual kWh vs. estimated kWh, is the actual kWh a measured kWh or a calculated kWh using your model with actual measured temperatures?

Cenk Yavuzturk: It is a calculated kWh. The comparison between the actual and predicted heat pump energy consumption is based on calculated energy consumption values using actually measured and model-predicted heat pump entering fluid temperatures.