中國機械工程學刊第十六卷第六期第 529~542 頁（民國八十三年）
Journal of the Chinese Society of Mechanical Engineers, Vol.15, No.6, pp.529~542 (1994)

A Direct Digital Control of Outlet Temperature in a Long Pipeline

Bin-Juine Huang* and Fong-Chin Tsai*

Keywords: digital control, heat exchanger control, distributed system.

ABSTRACT

A direct digital control system is developed to control precisely the outlet temperature of a long pipeline in a solar collector testing system. A pipeline dynamics model is derived that takes into account the effect of thermal capacitance of the wall and heat loss to the surroundings. We show that the pipeline dynamics can be approximated by a second-order lumped model with a time delay. This result indicates that a lumped model can be used to approximate the heat exchanger dynamics. A two-mode dynamics model (heating and cooling modes) is also derived and identified experimentally for the constant-level bath. A direct digital control system that includes a switching device to eliminate overshoot of outlet temperature, a Smith predictor to compensate time delay in the pipeline and a PI controller to obtain better feedback performance is developed to control precisely the outlet temperature. The switching of bath operation is divided into three regions. The proportional gain K_p and the switching temperature T_{sw} are tuned by a system simulation. It is shown that compensation of pipeline delay by the Smith predictor and switching from the saturation region into the linear heating region when $T_0 < T_{set}$ during heating mode eliminate overshoot and oscillation of T_0. The performance of the direct digital control system is satisfactory.

INTRODUCTION

The control of a local temperature in a pipeline is required in many processes, for instance, the temperature control at the outlet of a pipeline connected to a reactor or to oil heaters. Shown in Fig.1 is a system to test a solar collector that was developed by Huang and Hsieh (1990). The system supplies water flowing from a constant-level bath to the collector inlet at a fixed temperature T_0 (within $0.5°C$). To assure the system durability and minimize the maintenance problem, the whole hot water supply system including the constant-level bath should be installed indoor.

Paper Received December, 1993. Revised March, 1994. Accepted July, 1994. Author for Correspondence: B. J. Huang.

* *Department of Mechanical Engineering, National Taiwan University, Taipei, Taiwan, ROC.*

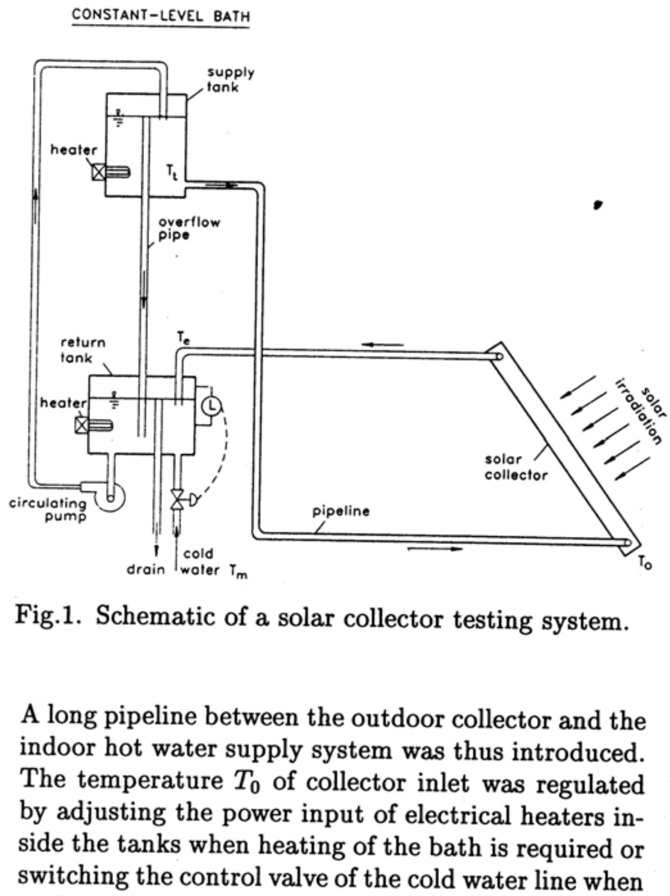

Fig.1. Schematic of a solar collector testing system.

A long pipeline between the outdoor collector and the indoor hot water supply system was thus introduced. The temperature T_0 of collector inlet was regulated by adjusting the power input of electrical heaters inside the tanks when heating of the bath is required or switching the control valve of the cold water line when cooling of the bath is required, depending upon the difference between the outlet temperature T_0 and the bath temperature T_t. The controller design was thus complicated due to the dynamics of a long pipeline and a constant-level bath and to the disturbances induced by temperature variations of the return water caused by fluctuations of incident solar radiation.

The temperature T_0 of the collector inlet (the desired control setpoint) was not directly controlled in the original design of Huang and Hsieh (1990). Instead, a PI controller was used to adjust the heating of the bath to regulate the tank temperature T_t in the heating mode of operation. The collector inlet temperature T_0 was left floating. The controller setpoint

T_t required to meet the desired T_0 was determined according to a correction function that experimentally correlates T_t with T_0 and the flow rate in the steady state. The quality of indirect control of T_0 was thus poor in the original design. A direct digital control system has been developed in order to solve this problem. The present work first focused on the modeling and identification of system dynamics of pipeline and constant-level bath. The dynamics of constant-level bath is divided into heating and cooling modes, depending upon the operating conditions. A computer system to control the outlet temperature which includes a Smith predictor, a PI controller and a switching device was then developed. The implementation of this control system has been shown to be successful.

MODELING AND IDENTIFICATION OF PIPELINE DYNAMICS

In general, control of the outlet temperature is difficult for a long pipeline due to heat loss to the surroundings and mixing or dispersion inside the pipe during the process of fluid transport. The heat loss makes the pipeline behave like a single fluid heat exchanger and thus belongs to a distributed parameter system. The heat-exchanger dynamics and control were reviewed by Williams and Morris (1961). Modeling for dynamics of double pipe heat exchanger of the types of liquid to condensing vapor and parallel flow was studied by Cohen and Johnson (1961) and frequency response tests were performed to identify the dynamics model. Hsu and Gilbert (1962) summarized transfer functions of various heat exchangers. Kawata et al.(1989) studied the dynamic response and PI control of a counterflow heat exchanger. Harrell et al.(1987) used a linear lumped dynamic model to design a model reference adaptive control system for a fluid heat-exchanging process and implemented it to control the water temperature of an evacuated tube solar collector.

All this work neglects the effect of the heat-exchanging surface in heat exchanger dynamics and does not apply model reduction to the distributed model. The outlet temperature response of a heat exchanger is affected by the heating surface material due to the effect of thermal capacitance. Thus the prediction of system dynamics behavior is misleading if the capacitance effect of wall surface is neglected (Huang and Hu, 1992). Furthermore, the controller design for a heat exchanger becomes complicated if model reduction is not applied (Watts and Schoenhals, 1966).

The present problem is described by the schematic diagram shown in Fig.2 as the two tanks in the constant-level bath are well mixed due to the internal circulation of water by a circulating pump (refer to Fig.1.). A large delay exists for outlet temperature responses and a distributed dynamic model results first. A lumped model with a delay is then derived and identified by the method of model reduction and system identification.

Modeling

The effect of thermal capacitance of wall material including insulation is taken into account in the system dynamics of the pipeline. The following assumptions were made:

(1) The variation of fluid and pipe wall temperature in the radial direction is negligible. The pipe wall, insulation and construction materials are lumped together as the solid phase in the radial direction, i.e. only temperature variations on the axial direction are considered (Fig.3). Hence the solid phase temperature varies as $T_w(x,t)$;
(2) The flow in the pipe is one-dimensional and incompressible;
(3) All physical properties are constant;
(4) The mass flowrate and ambient temperature are constant;
(5) Radiative heating on the pipe outside the surface is ignored;
(6) The axial heat conduction along the pipe wall is neglected since the heat transfer process inside the pipe is dominated by forced convection.

Therefore, energy balance taken to the fluid phase (CV1 in Fig.3) and solid phase (CV2) leads to

$$\tau_f \frac{\partial \widetilde{T}_f(x,t)}{\partial t} + K_f \frac{\partial \widetilde{T}_f(x,t)}{\partial x} + \widetilde{T}_f(x,t) - \widetilde{T}_w(x,t) = 0,$$

$$\tau_w \frac{d\widetilde{T}_w(x,t)}{dt} + (K_w+1)\widetilde{T}_w(x,t) - \widetilde{T}_f(x,t) = 0, \quad (2.1)$$

where

$$\widetilde{T}_f(x,t) = T_f(x,t) - T_a, \ \widetilde{T}_w(x,t) = T_w(x,t) - T_a, \quad (2.2)$$

$$\tau_f = \frac{\rho_f (D_i/4) C_{pf}}{h_f}, \ K_f = \frac{\rho_f v (D_i/4) C_{pf}}{h_f}, \quad (2.3)$$

$$\tau_w = \frac{(\rho_w/4)(D_o^2 - D_i^2) C_{pw}}{h_f D_i}, \ K_w = \frac{h_o D_o}{h_f D_i}. \quad (2.4)$$

Equation (2.1) is the governing equation of the pipeline. Sovling the Laplace transform of Eq.(2.1)

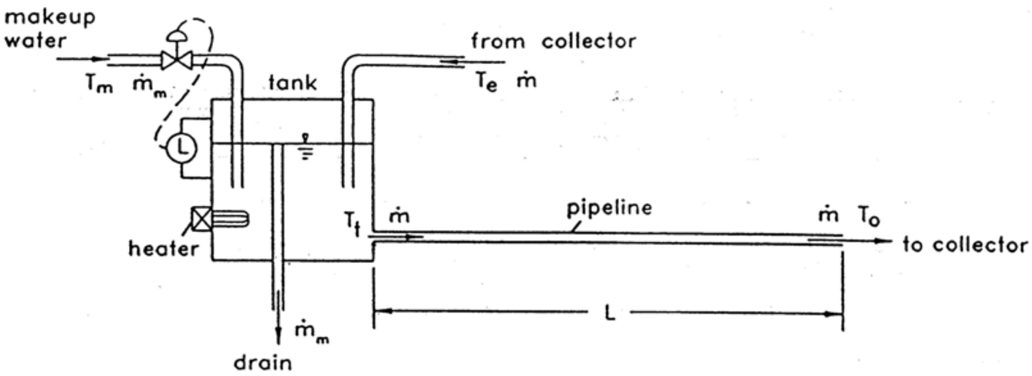

Fig.2. Equivalent representation of bath-pipeline system.

with zero initial conditions, we obtain a distributed model as

$$\frac{\widetilde{T}_f(s,x)}{\widetilde{T}_t(s)} = \exp\left[-\frac{\tau_f\tau_w s^2 + (\tau_f+\tau_w+\tau_f K_w)s}{K_f(\tau_w s + K_w + 1)}\cdot x\right]. \tag{2.5}$$

The system dynamics model of the pipeline $G(s)$ in terms of the outlet condition ($x = L$) is

$$G(s) \equiv \frac{\widetilde{T}_0(s)}{\widetilde{T}_t(s)} = \exp\left[-\frac{\tau_f\tau_w s^2 + (\tau_f+\tau_w+\tau_f K_w)s}{K_f(\tau_w s + K_w + 1)}\cdot L\right], \tag{2.6}$$

where $\tilde{T}_0(s) \equiv \tilde{T}_f(s,L); \tilde{T}_t(s) = \tilde{T}_f(s,0)$.

Equation (2.6) is a distributed model similar to that of a single-fluid heat exchanger (Watt and Schonehals, 1966). However, the effect of thermal capacitance of the pipeline wall is included in the present model through the time constant τ_w. τ_w is determined by system identification method as theoretical calculation of τ_w is inaccurate because of the difficulties to estimate specific heats and densities of pipe wall, insulation and construction materials etc.

Model Reduction

The dynamics model, Eq.(2.6), can be rewritten in the form

$$\frac{\widetilde{T}_0(s)}{\widetilde{T}_t(s)} = e^{-\tau ds}\exp\left[\frac{-\tau_w Ls}{K_f(\tau_w s + K_w + 1)}\right], \tag{2.7}$$

where

$$\tau_d \equiv \frac{\tau_f L}{K_f} = \frac{L}{v}, \tag{2.8}$$

is the average velocity of fluid.

The dynamics model of Eq.(2.7) is a distributed type with infinite order. For simplification, a model reduction is desired. Equation (2.7) is written, by series expansion, as

$$G(s) = \frac{\widetilde{T}_0(s)}{\widetilde{T}_t(s)} = e^{-\tau_d s}\cdot\left[1 - \frac{\tau_w Ls}{K_f(\tau_w s + K_w + 1)} + \frac{1}{2!}\cdot\frac{\tau_w^2 L^2 s^2}{K_f^2(\tau_w s + K_w + 1)^2} - \cdots\right]. \tag{2.9}$$

It is expected that the model can be reduced to second order with time delay τ_d by taking two terms in Eq.(2.9),

$$G(s) = \frac{\widetilde{T}_0(s)}{\widetilde{T}_t(s)} = G_0(s)e^{-\tau_d s}, \tag{2.10}$$

where

$$G_0 = \frac{\dfrac{(\tau_w-\tau_d)^2+\gamma_d\tau_w}{(K_w+1)^2}s^2 + \dfrac{2\tau_w-\gamma_d}{K_w+1}s+1}{\dfrac{\tau_w^2}{(K_w+1)^2}s^2 + \dfrac{2\tau_w}{K_w+1}s+1}, \tag{2.11}$$

where $\gamma_d = \tau_w L/K_f$. For application in the design of a computer-controlled system, Eq.(2.10) is converted into the z-transfer function model:

$$G(z^{-1}) = \frac{\widetilde{T}_0(z^{-1})}{\widetilde{T}_t(z^{-1})} = G_0(z^{-1})z^{-n_d}, \tag{2.12}$$

where $n_d = \tau_f L/(K_f T_s) = \tau_d/T_s$ and

$$G_0(z^{-1}) = \frac{b_0 + b_1 z^{-1} + b_2 z^{-2}}{1 + a_1 z^{-1} + a_2 z^{-2}}. \tag{2.13}$$

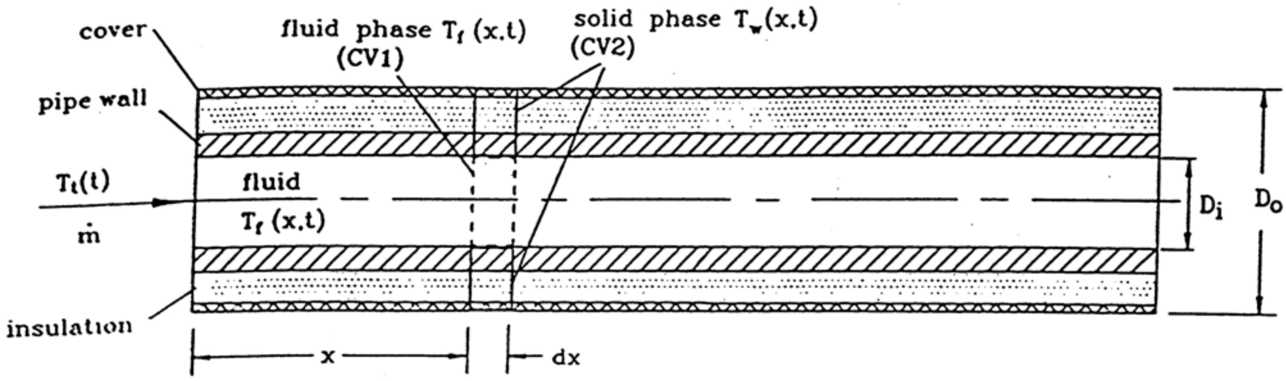

Fig.3. Modeling of a pipeline.

The pipeline dynamics, Eq.(2.10) or (2.12), needs to be identified experimentally.

Identification of Pipeline Dynamics

We use the solar collector testing system designed by Huang and Hsieh (1990) for experiment. The measuring instruments are basically the same. However, to improve accuracy and eliminate noise, thermocouples that were originally used to measure temperatures in the tanks, the inlet and outlet of collector, and the surroundings were all replaced by platinum resistance temperature (PRT) probes. The uncertainty of PRT probes is within $\pm 0.07°C$ from a precise calibration using a quartz thermometer (HP2804). The flowrate in the pipeline is measured by a turbine flowmeter (MK508) with uncertainty of $\pm 3\%$.

The connecting pipeline is made from stainless steel pipe of inside diameter $40mm$, wall thickness $3mm$, and length $10m$. A PU insulation layer ($3cm$) is installed at the outside surface of the pipeline.

The total capacity of the electric heaters in the tanks is $8.3kW$. The electric heater is regulated by a SCR triggered by a signal $0-10V$ from a D/A interface card. The tank is cooled by turning on the control valve of the cold water line while the heater is switched off. The flowrate of cold water is properly adjusted to provide a cooling capacity of $8.3kW$ in order to keep the whole system balanced in the steady state. The heating and cooling of the bath is controlled by a microcomputer.

A PRBS testing signal for heating power of the electrical heater in the bath is generated by a software in a microcomputer with sequence length 25 and clock period $10s$. As shown in Fig.4(a) is one heating power signal generated in a test run. Figure 4(b) is the signal of the tank temperature or pipe inlet temperature. Figure 4(c) is the outlet temperature response of the pipeline.

The pipeline dynamics model, Eq.(2.12), is thus identified using the measured input and output signals (i.e. T_t, T_0) and least-squares estimation. The identified pipeline model is

$$G(z^{-1}) = \frac{\widetilde{T}_0(z^{-1})}{\widetilde{T}_t(z^{-1})} =$$

$$\frac{0.1971 - 0.2597z^{-1} + 0.0674z^{-2}}{1 - 1.6877z^{-1} + 0.6928z^{-2}} \cdot z^{-41}. \qquad (2.14)$$

Figure 5 shows that the measured response of T_0 is near the predicted values according to the identified model. This result validates the pipeline dynamics model. The dynamics model of Eq.(2.14) represented in the frequency domain is shown in Fig.6.

MODELING AND IDENTIFICATION OF CONSTANT-LEVEL BATH DYNAMICS

Modeling

The modeling of the constant-level bath is illustrated according to Fig.2. The operation of the bath is divided into heating and cooling modes. Heating of the bath is required when the tank temperature T_t is less than that of pipeline outlet T_0; cooling is required when T_t exceeds T_0, i.e., the water in the bath is overheated. Cooling of the bath is provided by flushing the cold water into the tank by opening the flow control valve (on/off) and switching off the heater. For simplification, a on/off valve is used for cold water flow control. The dynamics of the bath is thus divided into two modes and derived by applying energy balance to the bath and assuming constant temperature T_m and flowrate $\dot{m}_m$ of cold water to obtain

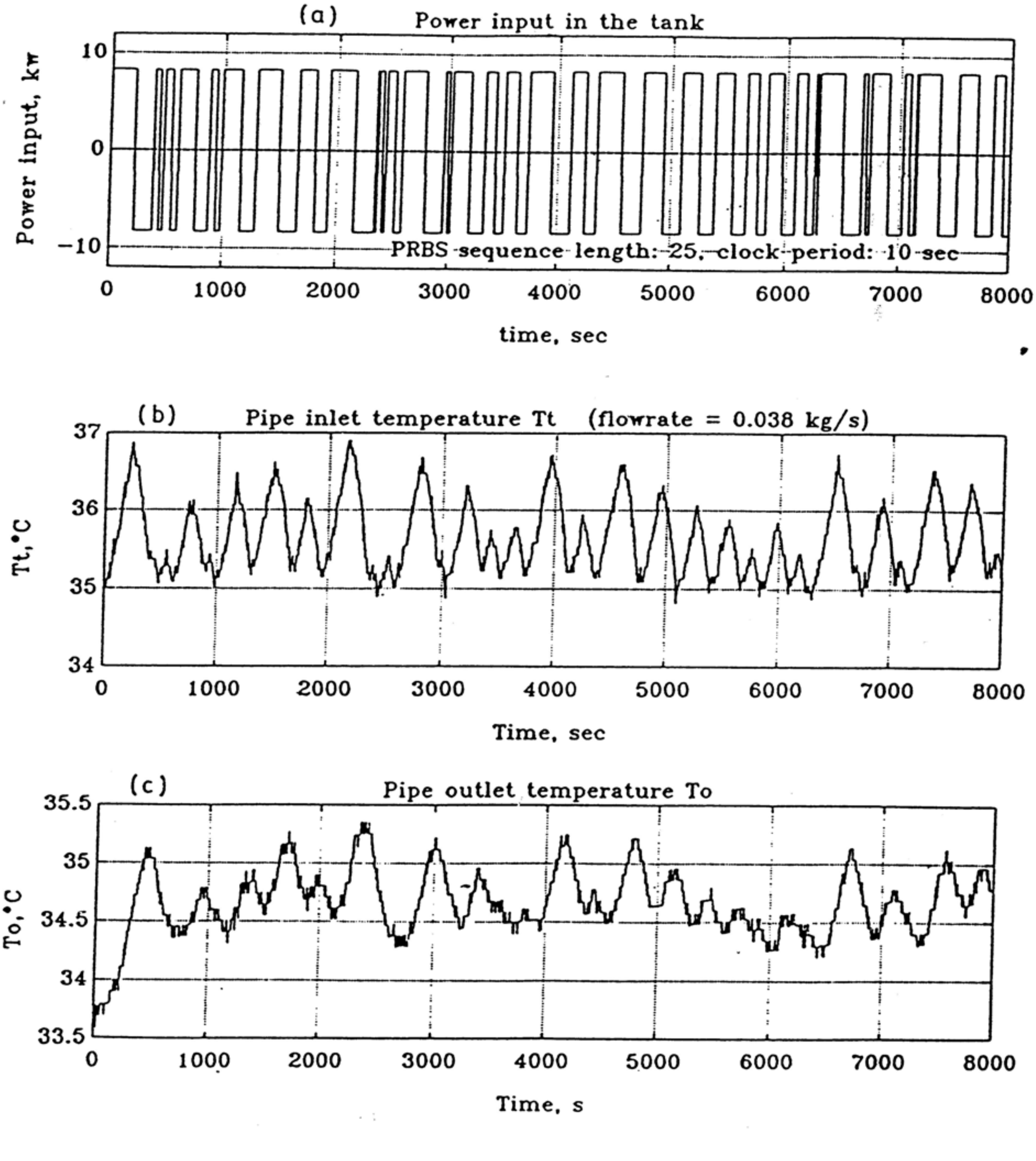

Fig.4. Test signals of pipeline dynamics. (a) PRBS signal of power input; (b) temperature response of tank; (c) temperature response of pipeline outlet.

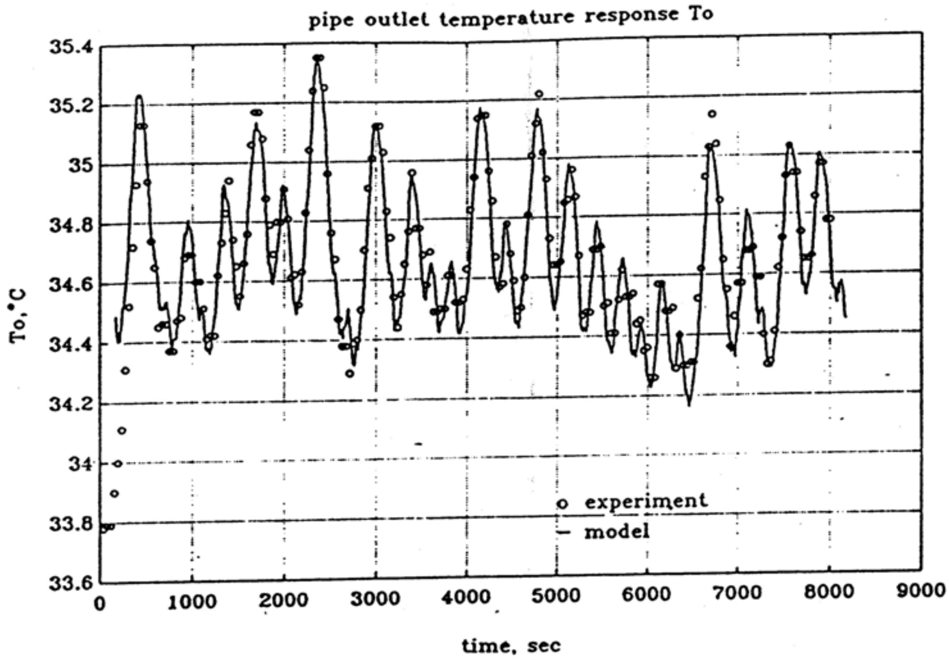

Fig.5. Measured and predicted response of outlet temperature.

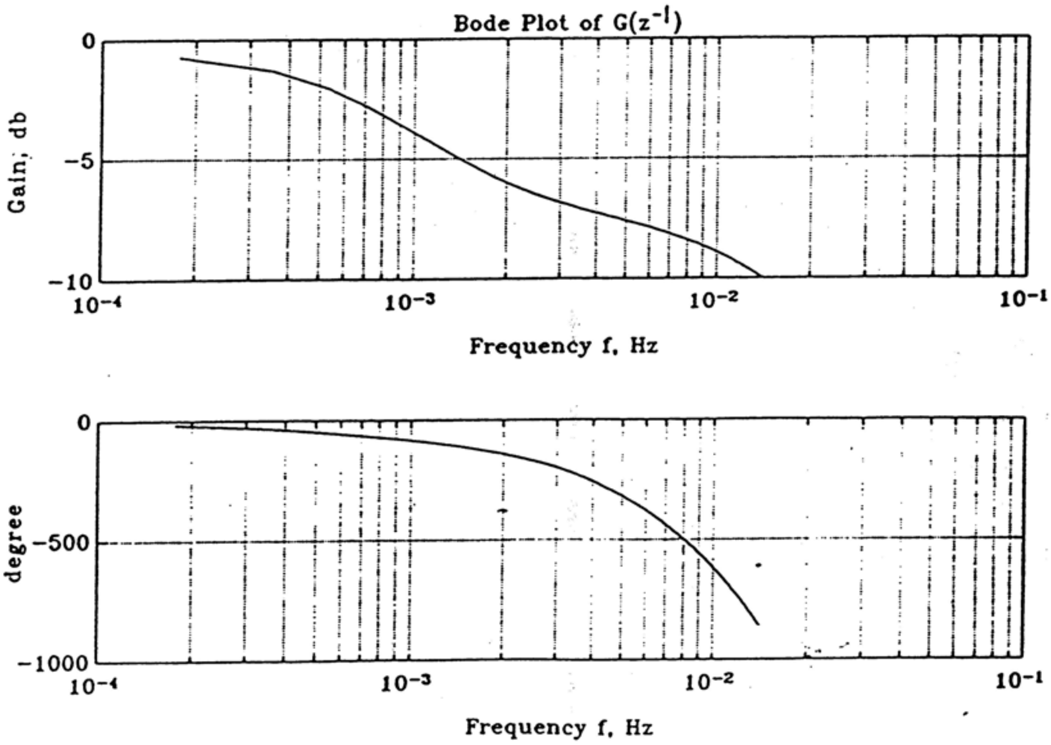

Fig.6. Frequency response of pipeline dynamics model.

$$\tau_t \frac{dT_t(t)}{dt} = \begin{cases} K_Q Q(t) - [T_t(t) - T_a] + K_e[T_e(t) - T_t(t)] \ ; & \text{for heating mode} \\ -[T_t(t) - T_a] + K_e[T_e(t) - T_t(t)] + K_m[T_m(t - t_d) - T_t(t)] \ ; & \text{for cooling mode} \end{cases} \tag{3.1}$$

where

$$\tau_t = \frac{M_t C_{pf}}{U_t A_t}, \quad K_Q = \frac{1}{U_t A_t},$$

$$K_e = \frac{\dot{m}_m C_{pf}}{U_t A_t} = \frac{1}{NTU_e},$$

$$K_m = \frac{\dot{m}_m C_{pf}}{U_t A_t} = \frac{1}{NTU_m}. \tag{3.2}$$

Here, the constants NTU_e and NTU_m are, respectively, the number of heat transfer units for the return flow $\dot{m}$ and the flowrate $\dot{m}_m$ of cold water. A delay t_d in response of tank temperature with respect to cold water inflow is introduced here in cooling mode as there are two tanks in the bath with flow recirculation between them.

Equation (3.1) is rewritten in terms of perturbation variables,

$$\tau_t \frac{d\widetilde{T}_t(t)}{dt} = \begin{cases} K_Q \tilde{Q}(t) - \widetilde{T}_t(t) + K_e[\widetilde{T}_e(t) - \widetilde{T}_t(t)] \ ; & \text{for heating mode} \\ -\widetilde{T}_t(t) + K_e[\widetilde{T}_e(t) - \widetilde{T}_t(t)] + K_m[\widetilde{T}_m(t) - \widetilde{T}_t(t)] \ ; & \text{for cooling mode} \end{cases} \tag{3.3}$$

where

$$\widetilde{T}_t(t) = T_t(t) - T_a, \quad \tilde{Q}(t) = Q(t), \quad \widetilde{T}_e(t) = T_e(t) - T_a,$$

$$\widetilde{T}_m(t) = (T_m - T_a)u(t), \tag{3.4}$$

where $u(t)$ is the unit step function.

The dynamics of the constant-level bath is thus obtained from the Laplace transform of Eq.(3.3):

$$\widetilde{T}_t(s) = \begin{cases} H_Q(s)\tilde{Q}(s) + H_e(s)\widetilde{T}_e(s) \ ; & \text{for heating mode} \\ C_m(s)\widetilde{T}_m(s) + C_e(s)\widetilde{T}_e(s); & \text{for cooling mode} \end{cases}, \tag{3.5}$$

where

$$H_Q(s) = \frac{K_Q}{\tau_t s + K_e + 1}, \quad H_e(s) = \frac{K_e}{\tau_t s + K_e + 1},$$

$$C_m(s) = \frac{K_m e^{-t_d s}}{\tau_t s + K_e + K_m + 1},$$

$$C_e(s) = \frac{K_e}{\tau_t s + K_e + K_m + 1}. \tag{3.6}$$

Parameter Identification

The parameters in Eq.(3.5) can be identified experimentally. τ_t and K_Q are identified in the heating mode by closing return flow from the collector, i.e. $\dot{m}_e = 0 (K_e = 0)$ or $H_e(s) = 0$, and applying a step input of $\tilde{Q}(t)$. A step of $882W$ was applied in the experiments and the step response curve was used to determine the parameters. The results are

$\tau_t = 16500s$, and $K_Q = 0.062°C/W$.

K_e is also identified in the heating mode by closing the cold water flow, i.e. $\dot{m}_m = 0 (K_m = 0)$ or $C_m(s) = 0$, and switching off heaters in the bath, i.e. $\tilde{Q} = 0$. A step input of $\widetilde{T}_e$ was then applied by sudden opening of a radiation shield of the solar collector to produce a step increase of collector exit temperature T_c. The result is

$$K_e = \frac{1}{NTU_e} = 7.5.$$

K_m and t_d are identified in the cooling mode by letting $Q = 0$, i.e. no heat to the bath, and closing the return flow from the collector, i.e. $\dot{m}_e = 0 (K_e = 0)$ or $C_e(s) = 0$, and suddenly applying a cold water flow, i.e., corresponding to a step input of $\widetilde{T}_m$. The response of $T_t(t)$ curve is then used to determine K_m and t_d. The results are

$$K_m = \frac{1}{NTU_m} = 46 \ , \quad t_d = 36s.$$

The dynamics of the constant-level bath then becomes, according to the definition of Eq.(3.6),

$$H_Q(s) = \frac{0.062}{16500s + 8.5}; \quad H_e(s) = \frac{7.5}{16500s + 8.5};$$

$$C_m(s) = \frac{46e^{-36s}}{16500s + 54.5}; \quad C_e(s) = \frac{7.5}{16500s + 54.5}. \tag{3.7}$$

The above dynamic models are converted into pulse transfer functions, and a computer-controlled system is designed based on the combination of the sampled systems to get

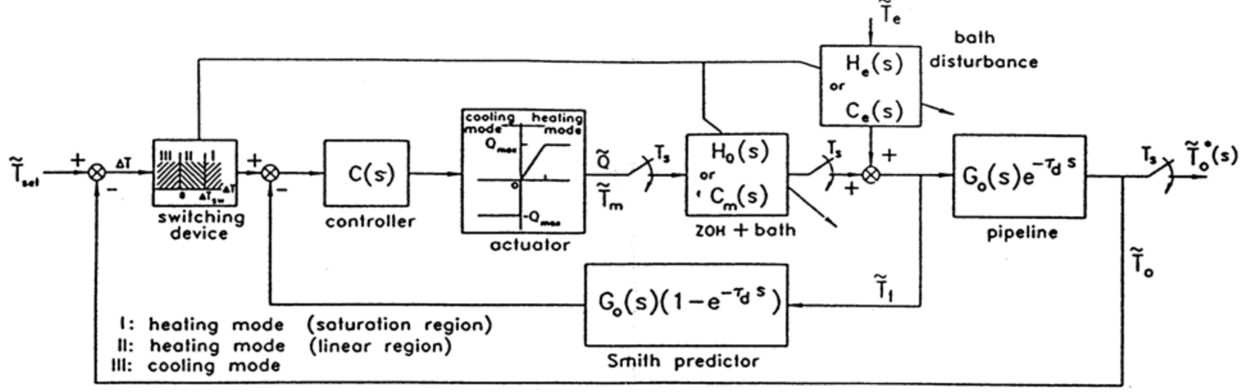

Fig.7. Design of direct digital control system.

$$\widetilde{T}_t(z^{-1}) = \begin{cases} H_Q(z^{-1})\tilde{Q}(z^{-1}) + H_e(z^{-1})\widetilde{T}_e(z^{-1}); \\ \quad \text{for heating mode} \\ C_m(z^{-1})\widetilde{T}_m(z^{-1}) + C_e(z^{-1})\widetilde{T}_e(z^{-1}); \\ \quad \text{for cooling mode} \end{cases}, \tag{3.8}$$

where

$$H_Q(z^{-1}) = \frac{1.5015 \times 10^{-5} z^{-1}}{1 - 0.99794 z^{-1}},$$

$$H_e(z^{-1}) = \frac{1.8163 \times 10^{-3} z^{-1}}{1 - 0.99794 z^{-1}},$$

$$C_m(z^{-1}) = \frac{1.1078 \times 10^{-2} z^{-10}}{1 - 0.98687 z^{-1}},$$

$$C_e(z^{-1}) = \frac{1.8163 \times 10^{-3} z^{-1}}{1 - 0.98687 z^{-1}}. \tag{3.9}$$

DIRECT DIGITAL CONTROL OF OUTLET TEMPERATURE

Watts and Schoenhals (1966) studied the control of outlet temperature control of a single-fluid heat exchanger by finding an optimum location of the temperature probe along the heat exchanger. A proportional controller was designed and the mean of the squared sum of the outlet temperature fluctuations was minimized to determine the optimum location of the probe and gain of the controller. As that work was only analytical, the results are inapplicable to the present problem as a temperature probe can be installed only at the pipeline outlet. We designed a direct digital system to control the outlet temperature of a long pipeline.

Control System Structure

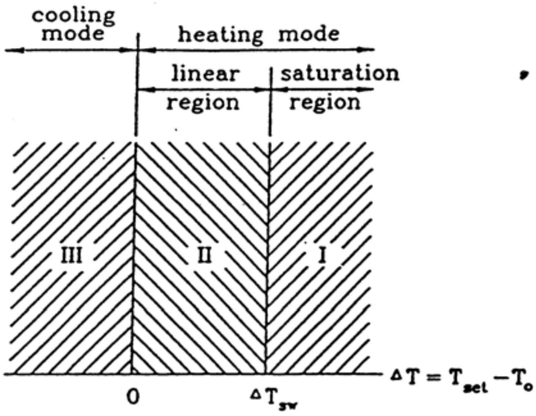

Fig.8. Switching regions.

According to Sections 2 and 3 the pipeline dynamics is described by a lumped model with a delay; the bath dynamics is described by a lumped model that is divided into heating and cooling modes. A direct digital control system based on a microcomputer was developed (Fig.7) in which a Smith predictor is used to compensate the delay in the long pipeline. A software switching device was also used to switch the bath dynamics into the heating or cooling mode. The saturation of heating power in the bath is considered in the heating-mode operation. A controller $C(s)$ is used in order to improve feedback performance.

The switching is divided into three regions (Fig.8), according to the error of the outlet temperature

$$\Delta T \equiv \widetilde{T}_{set} - \widetilde{T}_0 = T_{set} - T_0. \tag{4.1}$$

Region I is in the heating mode but with saturated heating power when $\Delta T > \Delta T_{sw}$ (i.e. $T_0 < T_{sw}$). Region II is also in the heating mode but with linear heating when $0 < \Delta T < \Delta T_{sw}$ (i.e.

$T_{sw} < T_0 < T_{set}$). Region III is in the cooling mode with a fixed cooling rate when $\Delta T < 0$ (i.e. $T_{set} < T_0$) with ΔT_{sw} defined as

$$\Delta T_{sw} \equiv T_{set} - T_{sw}, \tag{4.2}$$

where T_{sw} is the switching temperature.

The switching point T_{sw} defines the outlet temperature at which the controller switches the operation of bath between saturated (Region I) and linear (Region II) heating regions in the heating mode. A proper switching from saturation into linear heating when $T_0 < T_{set}$ is desired; otherwise, overheating or overshoot of the outlet temperature T_0 results and the controller is switched immediately into the cooling mode with a fixed cooling rate in the bath in order to decrease the bath temperature. This effect may cause oscillation of T_0 and poor quality of control. Compensation of the pipeline delay by a Smith predictor and switching from saturation into linear heating region in heating mode can eliminate overshoot and oscillation of T_0

Controller Design

The PI controller is designed such that its zero cancels the pole of the bath dynamics in the heating mode $H_Q(z)$ and the proportional gain is left to be optimized. The controller thus becomes

$$C(z^{-1}) = \frac{K_p(1 - 0.99794z^{-1})}{1 - z^{-1}}, \tag{4.3}$$

where K_p is the proportional gain.

The controller design described above can be optimized with respect to the proportional gain K_p and to the switching temperature T_{sw}. The controller thus belongs to a two-term regulator. The controller is designed based on the performance index IAE (integral of absolute error) that is defined as

$$IAE = \int_0^\infty |e(t)|dt, \tag{4.4}$$

where $e(t) = T_{set} - T_0(t)$.

A simulation was made to assess the performance of the control system for the design. The IAE is seen to vary with K_p and T_{sw} (Fig.9). An optimum value of K_p is $4800°C/W$. Shown in Fig.10 is the relation between IEA and T_{sw} at $K_p = 4800°C/W$ and $T_a = 20°C$ for various setpoint temperatures T_{set}. There exists an optimum T_{sw} for each T_{set}. The empirical relation is obtained as

$$\Delta T_{sw} = 3.30 - 0.03(T_{set} - T_a). \tag{4.5}$$

Implementation and Test Results

The experimental setup used to verify the design of the control system is the same as that described above. Flowrate in the pipeline is measured by a turbine flowmeter MK-508 that is calibrated with an uncertainty ±3%. Solar radiation incident upon the solar collector connected to the pipeline is measured by a PSP pyranometer with an uncertainty ±3%. The temperatures are measured by PRT probes and read by a YEW 3880 multimeter. All signals are transmitted to a microcomputer by a GPIB interface. The heaters in the bath are controlled by a SCR and a D/A interface card. The maximum input power and maximum cooling rate in the bath is $6730W$. The cooling is provided by flushing cold cold water into the bath by a solenoidal valve and a relay which is controlled by a microcomputer.

K_p is chosen as $4800°C/W$ in the present controller and the switching equation, Eq.(4.5), is stored in the computer for on-line switching. According to Fig.11(a) the regulation of the temperature T_0 of the pipeline outlet is satisfactory, with errors $< 0.25°C$ which are tolerable. It is seen that the control of T_0 is accurate and has a small fluctuation that has already met the requirements of the solar collector testing system (Huang and Hsieh, 1990). The small offset error is resulted from the $(1 - z^{-1})$ term in the PI controller which is not canceled, though the pole of the bath dynamics has been canceled by the zero of the PI controller. According to Fig.11(a) the switching point is about $56.5°C$. No overshoot of the outlet temperature response is observed and the temperature oscillation is small.

Figure 11(b) shows the instantaneous heating power during the temperature regulation. The maximum or saturated heating power is applied when $T_0 < T_{sw}$ (Region I). The heating power is adjusted linearly when $T_0 > T_{sw}$ (Region II). However, the controller may also be switched into cooling mode (Region III) intermittently when the system is operating in the linear heating region. This eflfect is due to the external disturbance of T_e induced by fluctuations of solar radiation, Figs. 11 (b)(c) and (d). Operation of cooling mode is seen from Fig.11(b) at the point $Q = 0$ that represents the occurrence of a fixed cooling rate in the bath. The accurate control of T_0 with small fluctuation shows that the disturbance rejection of the controller is satisfactory.

A proper selection of switching point T_{sw} results in a outlet temperature T_0 having very small overshoot and oscillation. Figure 12 shows the regulation of pipeline outlet temperatures. The responses of T_0 are seen to be rapid and to have small offset errors

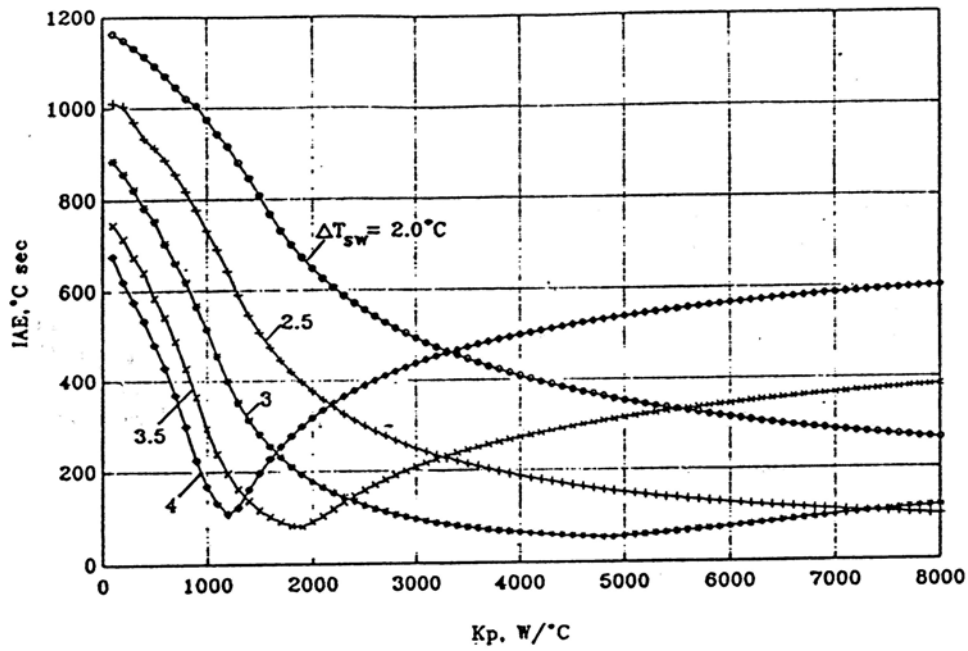

Fig.9. Simulation of the direct digital control system.

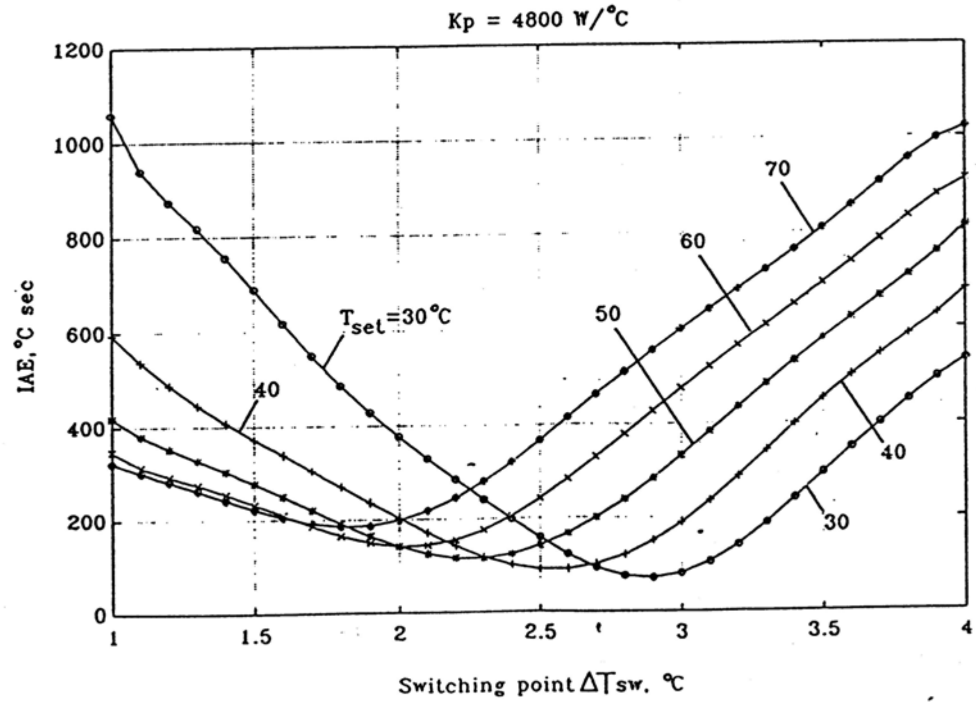

Fig.10. Simulation of the direct digital control system.

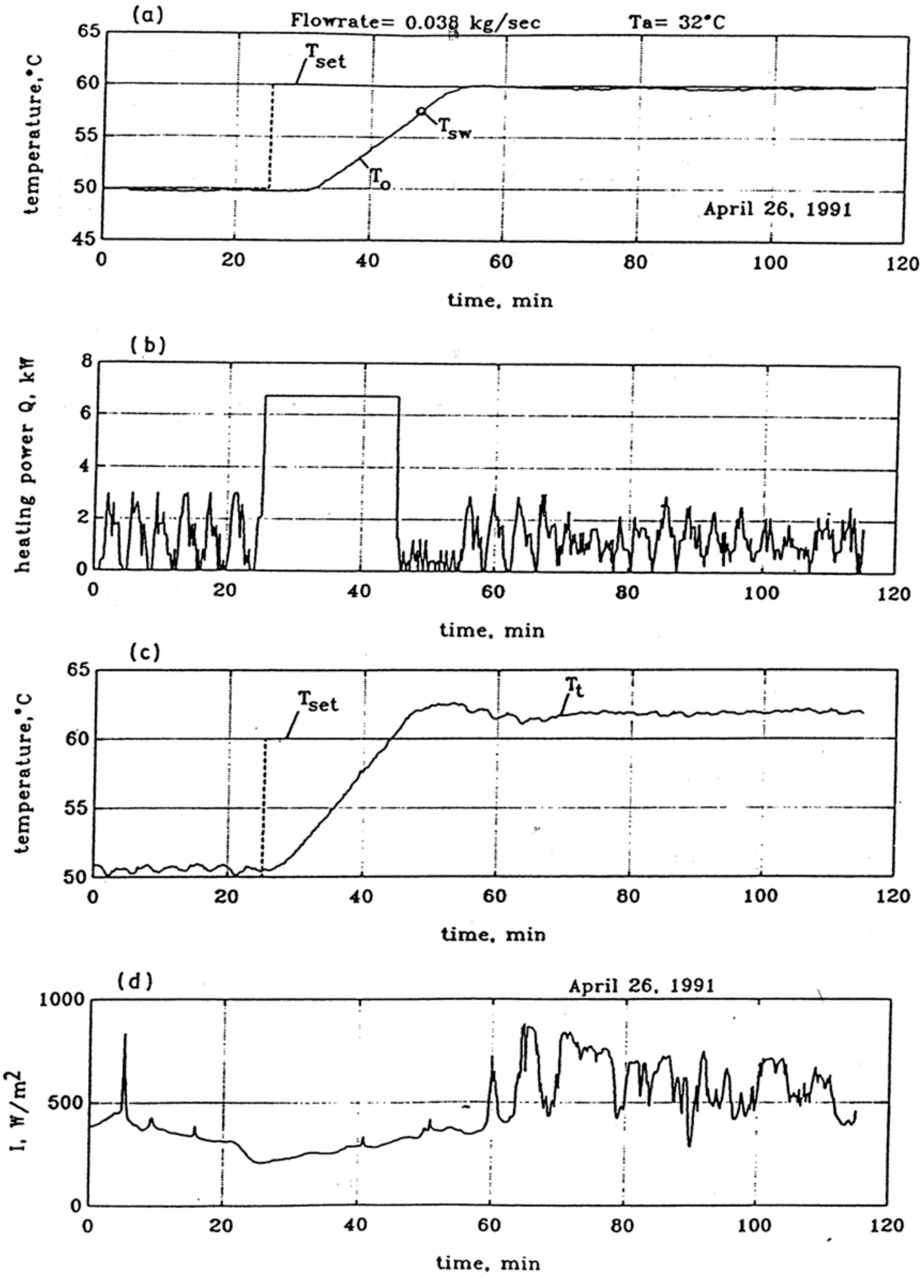

Fig.11. Test results of the control system. (a) Outlet temperature response and set temperature; (b) instantaneous heating power in the bath; (c) bath temperature response; (d) solar radiation.

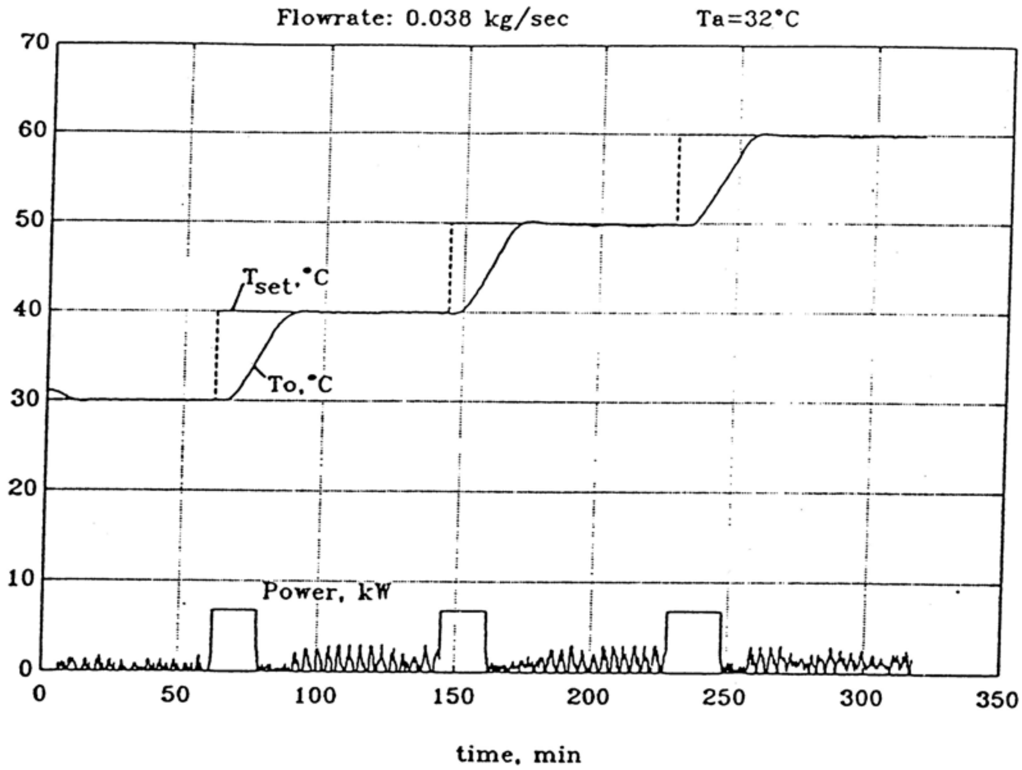

Fig.12. Outlet temperature control using the modified DDC control system.

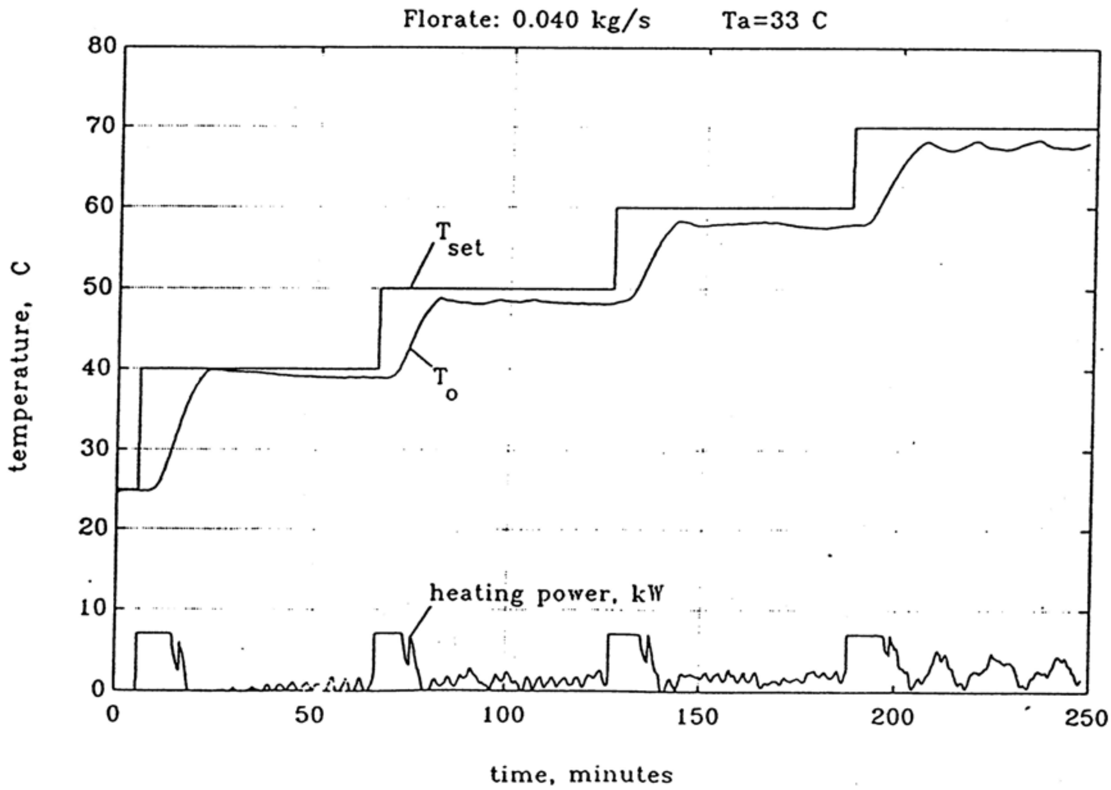

Fig.13. Outlet temperature control using the original controller of Huang and Hsieh (1990).

(all fluctuated within the range $0.25°C$). The rise periods of T_0 are shorter than $25min$ for a step change $10°C$ which is satisfactory in the collector testing system. A more brief rise period or more rapid response may be obtained if a greater heating power in the saturation heating region (Region I) is used.

The direct digital control system that we developed has been implemented in the collector testing system for more than three year. The controller performance is found to be satisfactory not only in accuracy and fluctuation, but also in disturbance rejection and response speed. The control quality has been improved dramatically as compared to the performance of the original controller which is shown in Fig.13.

CONCLUSION

A direct digital control system was developed to control precisely the outlet temperature of a long pipeline in a solar collector testing system. A pipeline dynamics model is first derived. The effect of thermal capacitance of the pipeline wall and heat loss to the surroundings were taken into account in the model. The pipeline dynamics are well represented by a second-order lumped model with a delay that is reduced from a distributed model. The thermal performance of a pipeline is similar to that of a simple heat exchanger that is usually described by a distributed model. The present result indicates that a lumped model can be used to approximate the dynamics of the heat exchanger in order to simplify the controller design of a heat exchanger.

A two-mode dynamics model (heating and cooling modes) was also derived and identified for the constant-level bath. A direct digital control system was then developed that includes a software switching device, PI controller and Smith predictor. The Smith predictor was used mainly to compensate the delay in the pipeline, whereas the PI controller was used to improve the feedback performance. A switching device was used to control the bath operation in three regions in order to reduce overshoot of T_0. Region I is in heating mode but with saturated heating power when $T_0 < T_{sw}$. Region II is also in heating mode but with linear heating when $T_{sw} < T_0 < T_{set}$. Region III is in cooling mode with maximum cooling rate when $T_{set} < T_0$. The cooling of the bath is provided by flushing a flow of cold water into the bath.

The PI controller is designed with the zero canceled by the pole of the bath dynamics. The controller thus can be tuned in terms of the best values of the proportional gain K_p and the switching temperature T_{sw} by simulation using IAE criterion. It was experimentally shown that compensation of the pipeline delay by the Smith predictor and the switching from saturation into a linear heating region when $T_0 < T_{set}$ during the heating mode can eliminate overshoot and oscillation of T_0. Good performance was obtained even if T_{set} was lowered. The controller also shows very good performance during the cooling mode. The accuracy within $\pm 0.25°C$ can be obtained for the new controller; while the old controller has a very large offset error (Fig.13).

The direct digital control system that was developed has been implemented in the collector testing system for more than three years. The control system performance has high accuracy, small fluctuation, good disturbance rejection and rapid response.

NOMENCLATURE

A_t	surface area for heat loss of the bath, m^2
C_{pf}	heat capacity of fluid (water), $kJ/kg\ °C$
C_{pw}	heat capacity of solid phase, $kJ/kg\ °C$
D_i	inside diameter of pipe, m
D_o	outside diameter of pipe, m
h_f	convective heat transfer coefficient inside the pipe, $W/m^2\ °C$
h_0	convective heat transfer coefficient of pipe outside surface, $W/m^2\ °C$
I	solar radiation incident upon the collector, W/m^2
K_e	parameter defined in Eq.(3.2)
K_f	parameter defined in Eq.(2.3)
K_m	parameter defined in Eq.(3.2)
K_Q	parameter defined in Eq.(3.2)
K_w	parameter defined in Eq.(2.4)
L	pipeline length, m
M_t	mass of water in the bath, kg
$\dot{m}$	mass flowrate in the pipeline, kg/s
$\dot{m}_m$	cold water flowrate to the bath, kg/s
Q	heating power in the bath, W
s	Laplace transform variable
T_a	ambient temperature, $°C$
T_e	temperature of return water, $°C$
T_f	fluid temperature, $°C$
T_m	temperature of cold water, $°C$
T_0	temperature of pipeline outlet, $°C$
T_s	computer sampling interval, s.
ΔT_{sw}	$= T_{set} - T_{sw}$
T_{set}	setpoint temperature for the control of T_0, $°C$
T_{sw}	switching temperature, $°C$
T_t	bath temperature or pipeline inlet temperature, $°C$
U_t	overall heat loss coefficient of the bath, $W/m^2\ °C$

v water velocity in pipeline, m/s
x pipeline position, m
τ_d time delay in the pipeline defined in Eq.(2.8), s.
ρ_f water density, kg/m^3
τ_f time constant of pipeline fluid phase defined in Eq.(2.3), s.
τ_t time constant of bath defined in Eq.(3.2), s.
τ_w time constant of pipeline solid phase defined in Eq.(2.4), s.

Greeks letters

superscript $\tilde{}$ perturbation from steady state.

ACKNOWLEDGMENT

We thank the National Science Council of the Republic of China, Taiwan, through Grant No. NSC78-0401-E002-18 and the Energy Commission, the Ministry of Economic Affairs, Taiwan, through Project No. 792J1 for support of this study.

REFERENCES

Cohen, W.C., and Johnson, E.F., "Distributed Parameter Process Dynamics," *Chem. Eng. Progress Symp. Series.*, pp.86-99 (1961).

Hsu, J.P., and Gilbert, N., "Transfer Functions of Heat Exchangers," *AIChE Journal*, Vol.8, No.5, pp.593-598 (1962).

Harrell, R.C., Kranzler, G.A., and Hsu, C.S., "Adaptive Control of the Fluid Heat Exchange," *ASME J. Dynamic Systems, Measurements and Control*, Vol.109, pp.49-52 (1987).

Huang, B.J., and Hsieh, S.W., "Automation of Collector Testing and Modification of ANSI/ASHRAE 93-1986 Standard," *ASME J. Solar Energy Eng.*, Vol.112, pp.257-267 (1990).

Huang, B.J., and Hu, S. P., "Hydrodynamic Instability of a Thermosyphon Loop with Wall Damping Effect," *J. of Chinese Soc. of Mech. Eng.*, Vol.13, pp.132-138 (1992).

Kawata, S. Kanoh, H., and Masubuchi, M., "A Correlation Between Steady-state and Dynamic Response of a Counterflow Heat Exchanger," *ASME J. Dynamic Systems, Measurement and Control*, Vol.111, pp.115-118 (1989).

Watts, R.G., and Schoenhals, R.J., "Feedback Control Optimization of a Single Fluid Heat Exchanger," *ASME Paper 66-Wa/HT-56*, ASME Winter Annual Meeting and Energy Systems Exposition, New York (1966).

Williams, T.J., and and Morris, H.J., "A Survey of the Literature on Heat Exchanger Dynamics and Control," *Chem. Eng. Progress Symp. Series.*, pp.20-32 (1961).

長管路出口的數位溫度控制

黃秉鈞　蔡豐欽

國立臺灣大學機械工程學系

摘　要

本研究主要發展一個數位控制系統，用來準確控制長管路出口的溫度，以應用於一個太陽能集熱器的自動化熱性能測試系統。本研究首先考慮管壁的熱容效應以及管內至大氣的熱損，推導出一個管線的分佈型動態系統模式，並進一步將之簡化成一個含時滯的兩階聚容模式。由此可知，一般熱交換器的動態系統行為可以用聚容模式來近似。本研究也依加熱或冷卻的操作區分，推導了一個兩段式的等位水槽動態系統模式。依據這些動態系統模式，本研究設計了一個包含轉換器、史密斯預測器、以及PI控制器的數位控制系統，用來準確控制管線出口的溫度。轉換器可用來減小溫度超越現象，史密斯預測器用來補償管線的時滯，PI控制器用來改良回饋性能。等位水槽操作的轉換分成三個區域來調整。PI控制器的比例常數 K_p 以及轉換溫度 T_{sw} 則經由系統模擬來調整。實驗結果顯示，史密斯預測器確可補償管線的時滯；當 $T_0 < T_{set}$ 時，將加熱控制由飽和區轉換為線性區的操作，可以減緩出口溫度的超越以及振盪現象。本套數位控制系統確實可準確控制長管線出口的溫度，並且已在現場連續使用了三年多，都能正常操作且效果優良。