DYNAMIC MODELLING OF A SYSTEM FOR PRODUCTION OF DISTRICT HEATING

Karin Axelsson

Department of Chemical Engineering, Lund University, Sweden

Abstract: The purpose of this master thesis has been to build a dynamic model of a plant for production of district heating, using the software Extend. The physical plant is located in Mörrum, Sweden, and has had problems with oscillations in the outgoing temperature to the consumers. The motivation for building the model was to identify the cause of these oscillations and suggest measures to improve the performance of the system. Computer files containing logged variables from the plant were used to validate the model. It was concluded that the oscillations in all probability were caused by friction in a control valve. A new control strategy that gives very pleasing results in the computer model has been developed.

1 INTRODUCTION

This article is part of a master thesis that has been carried out in association with Solvina AB and Södra Cell Mörrum, SCM. SCM provides district heating to the local energy company Karlshamn Energi AB, KEAB. The basic idea is that SCM heats the volume of water returned by KEAB to a specified temperature, set by KEAB. However, SCM has had problems with oscillations in the process leading to unacceptably large variations in outgoing temperature to KEAB. The purpose of this master thesis has been to identify the source of these oscillations and suggest measures on how to eliminate them. This was to be done by building a dynamic model of the system using the software Extend and then use the created model to come up with suitable ideas to improve the performance of the system.

2 PROCESS DESCRIPTION

All symbols used are listed in the list of symbols, Section 9.

In the district heating system at SCM, the energy required to heat the incoming water is supplied as steam, condensing in a condenser. A pair of control valves on the condensate pipe, denoted *TV501*, regulates the condensate flow from the condenser. Changing the positions of these valves, changes the condensate level and hence the area available for condensation is changed. The area available for condensation determines incoming steam flow to the condenser. A controller working with the steam flow as its set point regulates the condensate valves. The steam flow set point is specified outside the system boundaries and is hence an uncontrolled variable. To compensate for unpredictable changes in steam supply some kind of heat buffer is necessary in order to keep the outgoing temperature at its set point. This buffering capacity is provided by an accumulator which basically is a hot water reserve. When the amount of available steam is insufficient to heat the incoming water, hot water can be drawn from the accumulator and mixed with the outgoing stream from the condenser to obtain the specified temperature. On the other hand, when steam supply is large, the accumulator can be charged with excess hot water. In this case, the outgoing stream from the condenser is mixed with a bypassed fraction of the incoming cold water stream to obtain the desired temperature. Flow can go in both directions through the accumulator, depending on whether it is being charged or discharged with hot water. See Figure 1.

The sizes of the water streams in the systems are regulated by two pairs of control valves, denoted TV513 and TV515. Both pairs receive control signals from temperature controllers.

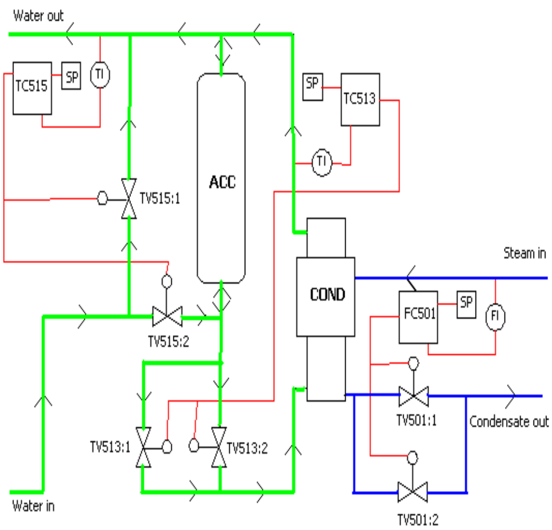

Figure 1: Sketch of the system, control loops are marked in red

The first valve pair, TV515, regulates the fraction of the incoming cold stream that is being bypassed the entire system. Their controller operates with the by KEAB specified temperature as its set point. The other pair, *TV513*, regulates the amount of water that flows through the condenser in relation to the amount of water that flows through the accumulator. Their controller set point can either be specified by the operator or be set to 10°C higher than the outgoing temperature specified by KEAB.

3 THE MODEL

The simulation of the system was performed using Extend. In Extend models of systems are built by connecting blocks and passing variables between them. Blocks receive input variables, perform computations and return output variables. Process operators and units are usually modelled as separate blocks. To model this system, blocks available at SCM were used along with new blocks created to model the key components of the system: the condenser, the accumulator and a flow computing block. To capture the behaviour of the real system, the model needed to be dynamic and allow for time varying inputs. For simulation purposes, the time step could be no smaller than one second, something that had to be considered when choosing numerical methods.

3.1 Condenser

The condenser is a shell and tube condenser with one pass on the tube side and one pass on the shell side. This simplifies calculations by allowing the unit to be modelled in two parts. The part of the condenser covered with condensate was modelled as a pure heat exchanger and the part where condensation occurs was modelled as a pure condenser. In each time step, flows and temperatures were assumed to be constant. Therefore the conventional steady state heat transfer equations could be used to solve for the unknowns. As long as the time step was kept reasonably small the variables will not change much between two consecutive time steps and the assumption is justified. The equations used for the condensate phase were:

$$\dot{Q}_1 = c_p \cdot \dot{m}_{water} \cdot (T_y - T_{in}) \qquad (1)$$

$$\dot{Q}_2 = c_p \cdot \dot{m}_{cond} \cdot (T_{sat} - T_{cond}) \qquad (2)$$

$$\dot{Q}_3 = A_{cond} \cdot k_{cond} \cdot \Delta T_{\ln,cond} \qquad (3)$$

$$\Delta T_{\ln,cond} = \frac{(T_{sat} - T_{cond}) - (T_y - T_{in})}{\ln\left(\dfrac{T_{sat} - T_{cond}}{T_y - T_{in}}\right)} \qquad (4)$$

$$\dot{Q}_1 = \dot{Q}_2 \qquad (5)$$

$$\dot{Q}_1 = \dot{Q}_3 \qquad (6)$$

With T_y and T_{cond} being the unknowns

And similarly for the steam phase:

$$\dot{Q}_1 = c_p \cdot \dot{m}_{water} \cdot (T_{out} - T_y) \qquad (7)$$

$$\dot{Q}_2 = \dot{m}_{steam} \cdot \Delta H_{vap} \qquad (8)$$

$$\dot{Q}_3 = A_{steam} \cdot k_{steam} \cdot \Delta T_{\ln,steam} \qquad (9)$$

$$\Delta T_{\ln,steam} = \frac{(T_{sat} - T_{out}) - (T_{sat} - T_y)}{\ln\left(\dfrac{T_{sat} - T_{out}}{T_{sat} - T_y}\right)} \qquad (10)$$

$$\dot{Q}_1 = \dot{Q}_2 \qquad (11)$$

$$\dot{Q}_1 = \dot{Q}_3 \qquad (12)$$

With T_{out} and m_{steam} being the unknowns

Since the systems of equations (5)-(6) and (11)-(12) containing the unknowns are nonlinear, solving them called for iterative solving techniques. In this case the Newton Raphson method was used. This method requires reasonable starting guesses in order to converge. For all but the first time step this was no problem since values from the previous time steps provided good starting guesses. However, starting the simulation could sometimes generate problems due to bad initial guesses for the unknowns. When the unknowns had been solved for, the condensate flow and condensate level were updated. The condensate flow was computed from the positions of condensate valves. The change in condensate level was computed from the difference between outgoing condensate flow and incoming steam flow.

$$\Delta y = \frac{\dot{m}_{steam} - \dot{m}_{cond}}{A_{cross,steam} \cdot \rho_{cond}} \qquad (13)$$

$$y(t + \Delta t) = y(t) + \Delta y \qquad (14)$$

3.2 Accumulator

The accumulator was modelled using a one dimensional model that takes convective and diffusive heat transfer and heat leakage to the surroundings into account. When using a one dimensional model it is assumed that there is no variation in temperature or velocity in the radial direction. In this case this was a valid assumption since flow velocities are low. In addition the accumulator is equipped with flow dispersers at the inlet and outlet to smooth out velocity and temperature variations across the radius. The governing equation and boundary conditions used to describe the temperature distribution in the accumulator are:

$$\frac{\partial T}{\partial t} = -v\frac{\partial T}{\partial z} + D\frac{\partial^2 T}{\partial z^2} - k_s(T - T_\infty) \qquad (15)$$

$$\left.\frac{\partial T}{\partial z}\right|_{z=0} + \frac{k_t L}{D}\left[T(t, z=0) - T_\infty\right] - \frac{v}{D}\left[T(t, z=0) - T_{in}\right] = 0 \qquad (16)$$

$$\left.\frac{\partial T}{\partial z}\right|_{z=L} - \frac{k_b L}{D}\left[T(t, z=L) - T_\infty\right] = 0 \qquad (17)$$

Equation (15) was solved using the finite difference method. In the axial direction, a second order symmetric method was used to discretize the second derivative and an asymmetric first order method was used to discretize the first derivative. The motivation for using a first order method instead of a symmetric second order method for the first derivate was that this put a far less severe restriction on the time step required for stability. The existence of a time step restriction was due to the fact that an explicit method, the explicit Euler method, was used to update the solution in time.

3.3 Flow computing block

The flow computing block was needed to determine the sizes of the water flows in the system, given the total incoming flow and the valve positions of *TV513* and *TV515*. The flows to be determined were the flows through the valve pair *TV513*, the flow through the two valves *TV515:1* and *TV515:2* and the flow through the accumulator. See Figure 2.The basis for the computations was that the pressure drops over two parallel pipes with common start and end points are identical. Using this, one equation could be written for the pressure drop between points A and B, and one equation could be written for the pressure drop between C and D, see Figure 2. In addition to these two equations, conventional mass balances were required to solve the system. Substituting the mass balances into the pressure drop equations resulted in a non-linear system of equations in two unknowns. The system was solved using the Newton-Raphson method. To write the pressure drop equations, expressions for pressure drops over the individual components - the valves, the accumulator and the condenser - were needed. Standard equations for pressure drops over valves were used. For an arbitrary valve *x*, the pressure drop is:

$$\Delta p_x = \frac{\rho_{fluidum}}{\rho_{H_2O,25°C}} \left(\frac{F_x}{Kv_x(u)} \right)^2 \approx \left(\frac{F_x}{Kv_x(u)} \right) \quad (18)$$

Note that *Kv* is a function of *u*. Valve characteristics describing this relation for all valves were available from SCM. The pressure drop over the accumulator and the condenser were written completely analogous, using fictive *Kv* values that were assumed to be constants.

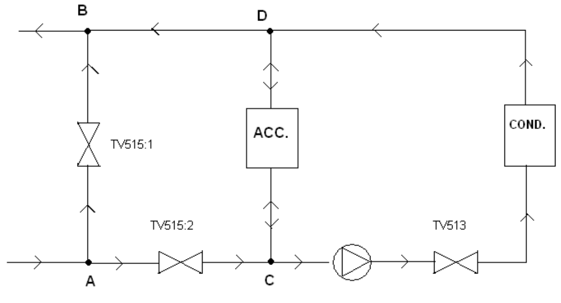

Figure 2: Simplified sketch of the system

4 MODEL VALIDATION

The model was validated using files containing logged variables from SCM. The computed temperatures could easily be validated, but since the only measured flows in the system are the steam flow and incoming water flow, validating the flow computing block proved difficult. However, estimation of the flow through the accumulator could be made by studying the temperature profiles in the unit. The model proved able to capture the large scale dynamics of the system. At first it did not, however, capture the oscillations. This was expected since the purpose of this thesis was to identify the source of these.

5 USING THE EXTEND MODEL TO IDENTIFY WEAKNESSES IN THE SYSTEM

The oscillations observed in the computer files seemed to arise in the condenser and were especially prominent in the following measured variables: incoming steam flow, outgoing water temperature from the condenser, condensate level and output from the steam flow controller. Closer analysis of the graphs showing steam flow and output from the steam flow controller revealed that they followed patterns typical for systems with valve friction. Hence it was concluded that the oscillations in all probability were caused by friction in the large condensate valve.

6 VALVE FRICTION

Since control valves are the only moving parts in control loops they are likely to be the cause of many control problems in the process industry, especially since they often operate under high pressure and/or high temperatures. Oscillatory control loops are often caused by valve friction. To understand this phenomenon, it is crucial to realise the difference between controller output and the valve position. A control valve has two main components; the body and the actuator. The body is the hollow part where the fluid flows. By adjusting the position of the valve stem, the resistance to flow and hence the actual flow through the valve is changed. The actuator is a device that receives signals from the controller and translates this into the position of the valve stem. Ideally, the movement of the valve stem perfectly follows the controller output, but practically this is not always the case. What happens when valve friction is present is that the valve stem gets stuck and does not move at all for small changes in the controller output. When it finally moves, it jumps too far, resulting in the control error changing sign. Then the same thing is repeated in the other direction. With valve friction, the valve does not move in a continuous fashion, but stepwise. This movement pattern is referred to as stick-slip motion and results in oscillations of the controlled variable around its set point.

7 SUGGESTION TO IMRPOVE CONTOL AT SCM

As mentioned earlier, the incoming steam flow to the condenser is regulated by two condensate valves. The output from the flow controller is converted into two control signals, one for each valve. Today split range control is implemented on the valve pair. This means that the small valve operates in the lower span for flow control outputs. When it saturates, the large valve takes over. This strategy relies on both valves being able to move continuously and hence fails when friction is present in the large valve.

A new strategy is proposed where instead of being active one at a time; both control valves work simultaneously in the entire range of controller outputs. The control signal to the large valve as a function of controller output is now a discrete stepwise function. The small valve is used for fine tuning at each discrete level. This strategy has two apparent advantages; it does not rely on the large valve being able to move continuously and it allows the small valve to operate in its optimal working span, i.e. near 50 % opened.

The control signal to the small valve is computed as

$$u_1 = \begin{cases} 125 + 15u_{501} - 30u_2 & u_{501} > 3 \\ 20 \cdot u_{501} & u_{501} \leq 3 \end{cases} \qquad (19)$$

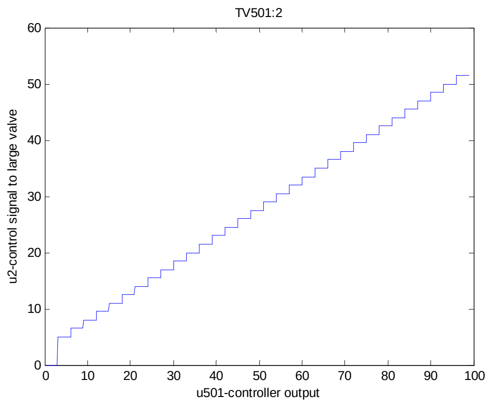

Figure 3: Control signal to large valve as a function of controller output

RESULTS AND DISCUSSION

The control strategy described in Section 7 was sent to SCM and implement in the control system. As far as observations were made, the strategy gave pleasing results. The oscillations in outgoing temperature were reduced and the system showed significant improvement in its ability to follow the steam flow set point.

8 LIST OF SYMBOLS

c_p	Heat capacity of water (J/kg·°C)
$\dot{m}_{water}$	Flow rate of water (kg/s)
y	Condensate level (m)
T_y	Temperature of the water at height y (°C)
T_{in}	Temp. of incoming water (°C)
$\dot{m}_{cond}$	Flow rate of condensate (kg/s)
T_{sat}	Saturation temperature of incoming steam (°C)
T_{cond}	Temperature of outgoing condensate (°C)
A_{cond}	Heat exchanger area to cond phase (m^2)
k_{cond}	Heat transfer coeff cond phase ($W/(m2^0C)$)
T_{out}	Temp. of outgoing water (°C)
$\dot{m}_{steam}$	Flow rate of steam (kg/s)
ΔH_{vap}	Heat of condensation (J/kg)
A_{steam}	Heat exchanger area exposed to steam (m^2)
k_{cond}	Heat transfer coeff steam phase ($W/(m2^0C)$)
$A_{cross,steam}$	Cross-sectional area of shell part (m^2)
T	Temperature (°C)
t	Time (s)
v	Velocity (m/s)
z	Axial coordinate (m)
D	Diffusion coefficient (m^2/s)
k_s	Heat transfer rate through walls(s^{-1})
T_∞	Surrounding temperature (°C)
k_t	Heat transfer rate through top (s^{-1})
k_b	Heat transfer rate through bottom (s^{-1})
T_{in}	Temp. of incoming water (°C)
Δp_x	Pressure drop (bar)
$\rho_{fluidum}$	Density of fluid (kg/m^3)
$\rho_{H_2O,25°C}$	Density of water at 25 °C (kg/m^3)
F	Flow rate (m^3/h)
Kv	Flow factor ($m3/(h \cdot bar^{1/2})$)
u_1	control signal to small condensate valve
u_2	control signal to large condensate valve
u_{501}	output from flow controller FC501

9 REFERENCES

Sell,Nancy J, Process Control Fundamentals for The Pulp & Paper Industry, Tappi Press, 1995

Forsman, Krister, *Reglerteknik för processindustrin*, Studentlitteratur, 2005