An Experimental Study of the Control of Condensate Subcooling In A Vertical Condenser

Temperature control of subcooled reflux in a distillation column was studied experimentally using coolant flow rate as the manipulative variable. Methanol vapor was condensed and subcooled at one atmosphere in a vertical heat exchanger with 210 copper tubes (0.77 cm I.D., 91.5 cm long). Cooling water flowed countercurrently through the shell side (20.6 cm I.D.).

A proportional-integral feedback controller provided good temperature control at normal design throughputs. But dynamic control performance deteriorated at low throughputs (or large areas). Thus the normal conservative design practice of providing excess area in this type of condenser leads to poorer temperature control.

JEAN-PAUL ARCHAMBAULT
JEAN-PAUL JAUFFRET
and
WILLIAM L. LUYBEN

Department of Chemical Engineering
Lehigh University
Bethlehem, Pennsylvania 18017

SCOPE

Subcooling condensers are commonly used on many distillation columns, particularly in atmospheric and vacuum columns. The control of the temperature of the subcooled reflux has been found to be quite difficult in many installations and has caused off-specification products and/or excessive utility consumption. The reason for these difficulties was thought to be the large gain, that is, a small change in vapor load or in coolant flow produces a large change in reflux temperature. This occurs because the sensible heat required for subcooling is usually a small fraction of the heat transferred in the condenser.

Little has appeared in the literature on the dynamics of subcooling condensers. Williams (1961, 1967) has presented literature reviews of heat exchanger dynamics. Stainthorp and Axon (1965) and Heideman et al. (1971) have studied the condensation of steam, but without subcooling. Steady state studies of the effect of inerts were described by Colburn and Hougen (1934). Bell (1972) discusses steady state temperature profiles in condensers with desuperheating and subcooling.

The purpose of this research was to study experimentally the dynamic performance of feedback control systems for a vertical tube-in-shell subcooling condenser. A steady state mathematical model of the condenser was developed to help understand the behavior of the condenser.

CONCLUSIONS AND SIGNIFICANCE

Experimental studies showed that a proportional-integral feedback controller could provide good temperature control when the condenser was operated at normal design throughputs. Dynamic control performance was poor at low throughputs (large heat transfer areas). These experiments demonstrated that the common practice of overdesigning subcooling condensers can result in poorer control.

Derivative action could not be used because the reflux temperature signal was found to be quite noisy, indicating vapor/liquid flow fluctuations through the tubes.

A cascade control system (cooling water exit temperature controller setpoint coming from the reflux temperature controller) was found to provide little improvement.

DESCRIPTION OF THE SYSTEM

A sketch of the system is shown in Figure 1. The condenser used in this study was a modified American standard heat exchanger. It had a one-pass vapor side in 210 tubes (BWG 20, .77 cm I.D., 91.5 length) and a one-pass countercurrent cooling water flow in the shell side (five baffles, 15 cm apart, 35% cut). The condenser was mounted vertically and was vented at its bottom. The vapor bonnet was redesigned to improve the vapor distribution in the tubes.

Condenser

Make: American Standard type 803 HCF
Modifications: One tube pass. Redesigned vapor bonnet
Working conditions: Vertical. Process fluid in the tubes

Correspondence concerning this paper should be addressed to W. L. Luyben.

Tube side

Number of tubes: 210
Metal: Copper
Arrangement: Triangular pitch on 1.25 cm centers
I.D., O.D.: 0.77 cm, 0.95 cm
BWG: 20
Heat exchange area: 5.75 m^2 outside, 4.66 m^2 inside

Shell side

Metal: Seamless brass
I.D.: 20.6 cm
Number of baffles: 5
Baffle spacing: 15 cm
Baffle cut: 35%

Vapor bonnet

Metal: Brass
I.D.: 20.6 cm
Height: 30 cm
Vapor coming in horizontally on the side by a 3.8 cm diameter pipe

Condensate bonnet

Metal: Cast iron
Half sphere shape, 12 cm radius
Condensate and vent lines connected at its bottom. 1.9 cm diameter pipes
A cap protects the vent line from receiving the condensate

A U-seal was installed in the reflux line to maintain a liquid pool in the bottom of the condenser.

The vapor came from a 24-tray, 20-cm diam. distillation column working under total reflux at 1 atm. The vapor was almost pure methanol (98% methanol, 2% water). The steam input to the reboiler of the column was maintained constant. The reflux overflowed from the condenser to a reflux drum and then back into the column. The cooling water was drawn from a constant-temperature, constant-head tank.

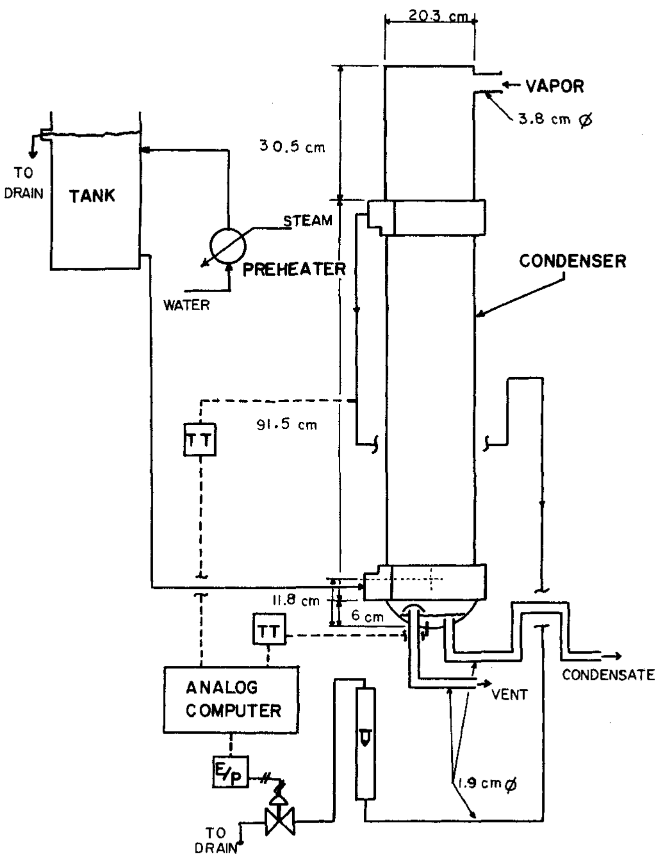

Fig. 1. Condenser and related equipment.

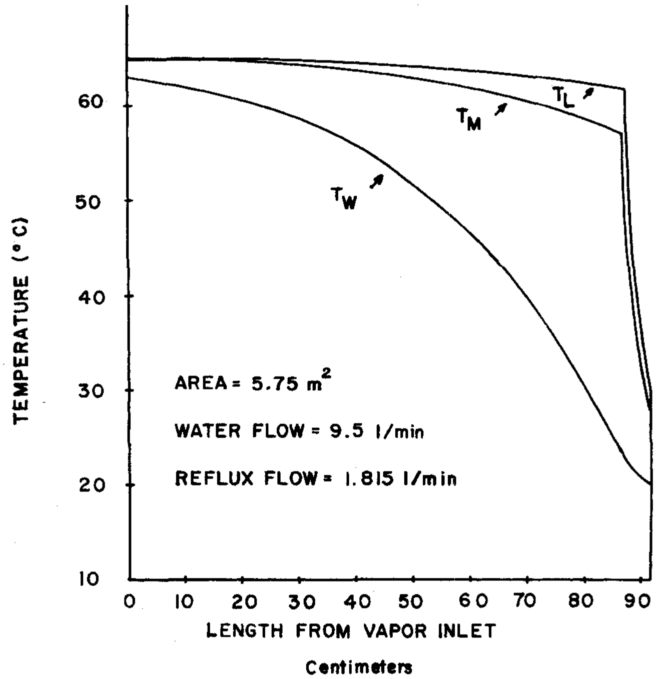

Fig. 2. Temperature profiles—large area.

The temperatures of the liquid pool in the bottom of the condenser and of the exit water were measured by resistance bulbs and converted into 4-20 mA current signals in Taylor transmitters.

An analog computer simulated a *P*, *PI*, or cascade control system and produced a voltage output which was converted into a pressure signal through a voltage-to-pneumatic transducer. Ultimately a pneumatic air-to-close control valve regulated the water flow rate. The C_v of this valve was 0.642. The maximum flow obtained was 20.3 liter/min with a pressure drop of one atmosphere.

STEADY STATE MATHEMATICAL MODEL

A digital computer program was developed to predict the steady state performance of a condenser with subcooling. The necessary input data were the water flow rate, the reflux rate, and the water exit temperature. The body of the program was a step by step Euler integration down the length of the condenser. The Nusselt analysis (Kern, 1950) as modified by Rosenhow (1956) was assumed locally valid in the condensing section. The heat transfer coefficient in the subcooling section was given by a relationship from Hewitt and Hall-Taylor (1970) for a falling liquid film. Whenever possible the effects of temperature on the physical data were included. Vapor was assumed to be pure methanol.

Predicted temperature profiles in the condenser for two base cases are shown in Figures 2 and 3. Table 1 gives parameter values and steady state variables for two cases. Figure 2 shows the predicted temperature profile when the condenser had a large area (210 tubes) and the reflux was at 30°C, the inlet cooling water being 20°C. The exit water temperature T_W was 63°C, and in the top of the condenser very little condensation occurred due to this small temperature difference. Changes in vapor load did not affect this temperature. In Figure 3 the condenser area was reduced to 35% (73 tubes) of its original value. The water temperature profile was more linear. More condensation occurred near the top because the temperature

TABLE 1. BASE CASE CONDITIONS

	Large area	Low area	Units
Number of tubes	210	73	
Heat exchange area, inside	4.66	1.63	m^2
outside	5.75	2.01	m^2
Vapor velocity in inlet pipe	18.1	17.25	m/sec
in bonnet	0.614	0.594	m/sec
in tube	2.15	5.84	m/sec
Vapor Reynolds number in tube	2,590	7,030	
Reflux flow rate R	1.815	1.75	1/min
Reflux exit temp., T_R	30	30	°C
Cooling water flow rate, F	9.5	11.3	1/min
Water Reynolds number	280-120	290-140	
Cooling water temp., inlet	20	20	°C
exit	63	55	°C
Energy load for condensation	374	361	Kcal/min
subcooling	31	30	Kcal/min
Total energy load	405	391	Kcal/min
Water heat transfer coefficient	375-307	845-705	Kcal/hr · m^2 · °C
Condensate coeff.: condensation	12,800-1,800	5,750-1,200	Kcal/hr · m^2 · °C
subcooling	2,700-2,500	2,000-1,800	Kcal/hr · m^2 · °C

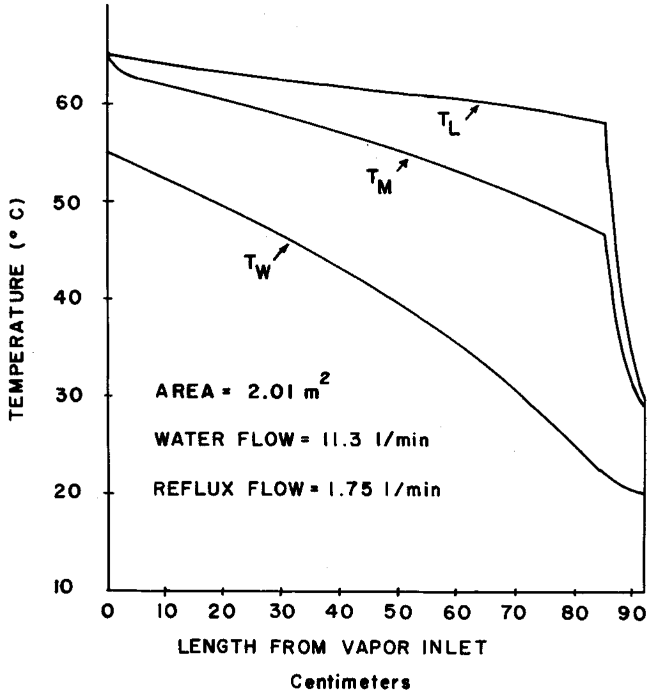

Fig. 3. Temperature profiles—small area.

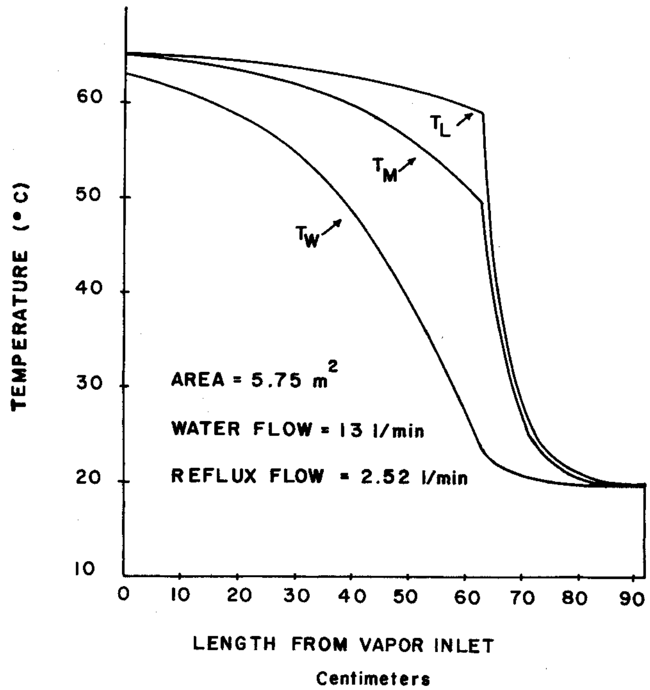

Fig. 4. Temperature profiles—high water flow.

difference across the tubes was increased. Vapor load changes did affect the exit water temperature.

The model predicted a very steep slope in the temperature profile of the condensate at the beginning of the subcooling section. When the reflux temperature T_R was not close to the inlet water temperature, the condensing section occupied most of the condenser. If the vapor was not equally distributed in the tubes, some could easily break through at low subcooling regimes.

EXPERIMENTAL WORK

Since methanol's boiling point was 64.7°C at 1 atm., only 30° to 40°C of subcooling could be obtained using normal cooling water temperatures. Furthermore, the heat of condensation of methanol was relatively high. Thus the subcooling that took place was only a small fraction of the total heat exchange.

The only practical way to run the condenser manually was to use an excess of water and subcool the condensate down to almost inlet cooling water temperature. Then the subcooling section was long enough to absorb perturbations of the vapor rate or composition and the reflux was at almost the inlet water temperature (see Figure 4).

It was not possible to run the condenser manually at a fixed amount of subcooling. If the temperature of the reflux was too high, a very slight increase in cooling water flow rate would decrease the reflux temperature considerably. This colder reflux exchanged heat with the vapor coming up in the column and condensed part of it. Subsequently there was less vapor going to the condenser. The temperature of the reflux further decreased, making another change in the water flow rate necessary and so creating a low frequency thermal cycle.

To compare the results, a standard load disturbance was used. At steady state, the steam flow rate to the reboiler was increased stepwise 10%. When the new steady state was achieved, a step change in the steam flow rate back to its original position was made.

During the experiments, the reflux pool temperature T_R and the exit water temperature T_W were recorded. It was also possible to record the voltage output of the analog computer and so measure the control effort.

A proportional controller was used to stabilize the process and give information on its dynamic behavior. The simulation on the analog computer is reproduced in Figure 5. A very low gain had to be used to prevent large oscillations in temperature with no visible damping. Because of the nonlinearity of the system, increasing gain merely increased the amplitude of these oscillations. The ultimate gain *Ku* was defined as that gain for which the system stabilized after four oscillations. The ultimate period *Pu* of the oscillations, when gain was *Ku*, was used to find the *PI* settings by the Ziegler Nichols method.

The ultimate gains and ultimate periods decreased slightly with increasing reflux temperature.

For a 20°C cooling water:

T_R (°C)	30	40	50
Ku	0.4	0.3	0.3
Pu (min)	4.92	4.82	4.0

The ultimate gain is the value of the setting on the analog computer (volts/volts). The gain of the transducing equipment and control valve was 0.545 (liters/min of water per °C). Notice that the proportional band settings must be very wide (greater than 250 to 330) for stable operation.

Actually the recorded temperature was constantly moving and was a very noisy signal indicating vapor/liquid fluctuations in the condenser. Because of this noisy signal, derivative action could not be used.

SINGLE LOOP PI CONTROL

A proportional-integral controller was first set with the Ziegler Nichols settings $Kc_R = Ku/2.2$ and $\tau_{IR} = Pu/1.2$ (see Figure 5). Experimental tuning was also attempted. Responses to the load disturbance were compared. When we tightened the control, T_R got oscillatory with a low frequency of oscillations (slow period of about 20 min.). When we further increased Kc_R or decreased τ_{IR}, this phenomenon became more noticeable and the frequency of oscillations increased slightly. This slow dynamic effect came from the interacting of the distillation column with the condenser. The control was rather poor (see Figure 6).

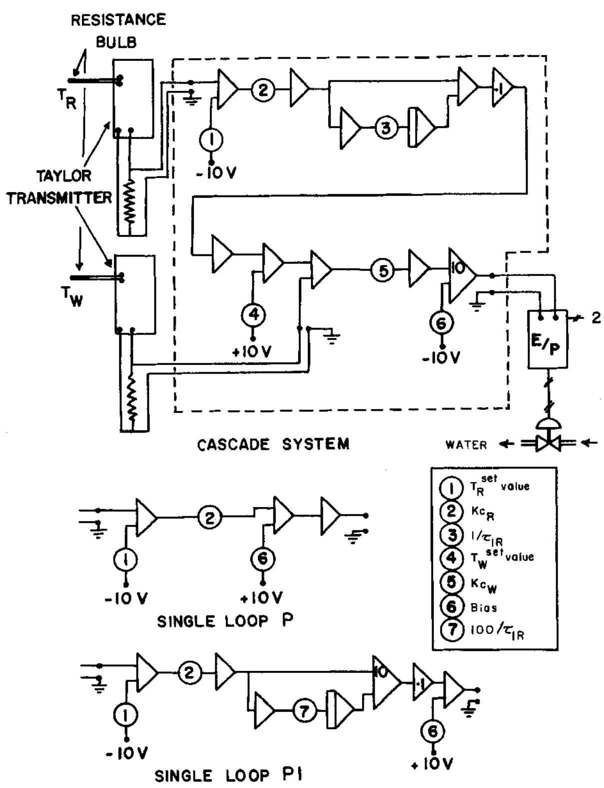

Fig. 5. Analog computer set-ups.

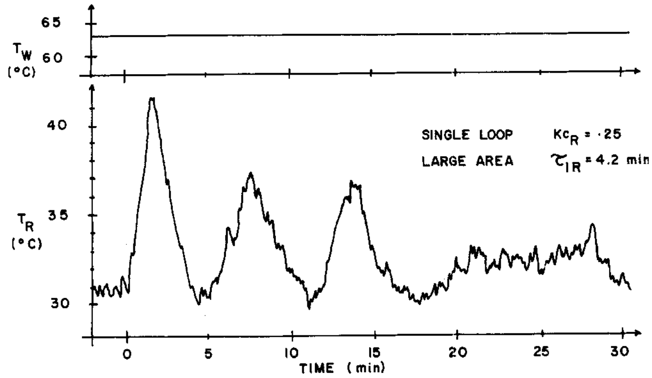

Fig. 6. Single loop load response—large area.

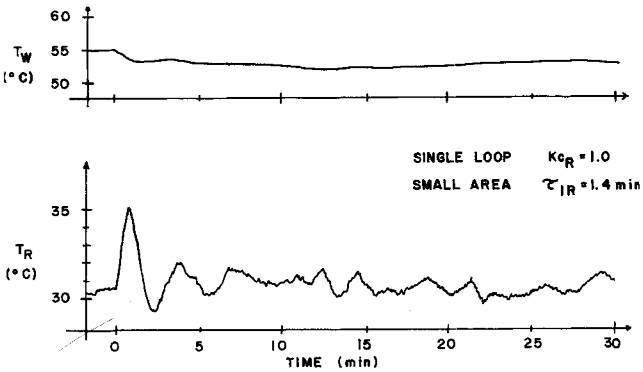

Fig. 7. Single loop load response—small area.

Conditions: Area: 5.75 m² (210 tubes)
Water flow: 9.5 l/min
Reflux flow: 1.815 l/min
Best settings: $Kc_R = .25$; $\tau_{IR} = 4.2$ min

The exit cooling water temperature T_W was, at 63°C, quite close to the vapor temperature and did not vary with either vapor rate changes or water flow rate changes. Obviously a cascade control system using this temperature as the slave loop signal would not be a cascade system at all. Exit cooling water temperature cascade control systems have been successfully used on some industrial columns. In order to get a system in which exit water temperature did vary and in which a cascade control system could be evaluated, the load on the condenser must be increased or the area of the condenser decreased. Since vapor rates were limited by column reboiler capacity, the condenser area was reduced by plugging tubes.

We plugged 137 tubes, leaving only 73 tubes (or 35% of the original area) open to the vapor flow. As was expected, the temperature T_W dropped to 55°C and a higher cooling water flow rate was required. Under these new conditions experiments with a *PI* feedback loop were repeated. Control was much better (see Figure 7). The control settings could be tighter.

Conditions: Area: 2.01 m² (73 tubes)
Water flow: 11.3 l/min
Reflux flow: 1.75 l/min
Best settings: $Kc_R = 1.0$; $\tau_{IR} = 1.4$ min

The controller gain was higher due to lower process gain. Also, by having less area, the residence time of condensate in the tubes was reduced and, having more water flow, the residence time of water in the shell was decreased. The dynamics were faster and consequently the load response was better. The upset was reduced to a few degrees C and lasted less than five minutes. No low frequency cycles, caused by interaction with the column, were observed. Thus control of subcooled reflux temperature was quite acceptable when the condenser was run under more fully loaded conditions which would be more typical of design conditions.

This demonstrated that control dynamics get worse when a condenser is overdesigned or run at low throughputs.

CASCADE CONTROL SYSTEM

Using the low area case, the cooling water exit temperature was no longer constant and a cascade control system could be tested. This cascade system was not a conventional cascade set up in which slave and master process transfer functions are in series. The process transfer functions in this system are in parallel. A representation of these two systems is given in Figure 8. The equivalent slave loop transfer function $\left(\frac{F}{T_W{}^{set}}\right)$ is $\left(\frac{B_W}{1+B_W G_W}\right)$. In the usual series cascade it is $\left(\frac{B_W G_W}{1+B_W G_W}\right)$.

A gain of 10 was used in the slave controller. This value gave a stable response to a change in set point.

The cascade system was found to be unstable when the master *PI* controller settings of $Kc_R = 1.0$ and $\tau_{IR} = 1.4$ were used. Experimental tuning led to some very different settings than the Ziegler Nichols settings in the single loop system.

The best control was achieved with

$$Kc_R = 0.1; \quad \tau_{IR} = 0.083\,\text{min}; \quad Kc_W = 10.$$

Control (see Figure 9) was somewhat better than obtained with a single-loop feedback system. The proportional action was divided by a factor of 10 and the integral action was increased by even more than 10. When the above settings were tried on a single *PI* feedback loop the system was unstable.

A floating master controller was also tested. Its performance was still good but slightly more oscillatory.

The cascade system presented some slight advantage, as far as performance goes, under these conditions. In the low flow case (or large area case), the cascade system is of no theoretical use since the exit water temperature was a constant. The slave controller behaved then as a gain in series with the master controller. This could be helpful in commercial instrumentation equipment with limited gain settings. It permits one to attain a lower loop gain.

THEORETICAL ANALYSIS OF PARALLEL CASCADE

Open loop step changes were made to get a rough idea of the system transfer functions (see Figure 8). The two transfer functions G_R and G_W relating the water flow rate to the temperatures were described by first-order lags with deadtime.

$$G_R = \frac{70\, e^{-.23s}}{(2.9s+1)} \qquad G_W = \frac{4\, e^{-.25s}}{(1.7s+1)}$$

$$B_R = Kc_R \frac{\left(s + \frac{1}{\tau_{IR}}\right)}{s} \qquad B_W = Kc_W = 10$$

Figure 10 is a polar plot of $B_R G_R$ with the single loop and cascade settings for B_R. One can see that if the cascade settings are used in a single loop, the system is unstable. Figure 10 shows also a plot of $\left(B_R G_R \frac{B_W}{1+B_W G_W}\right)$ for different values of B_W.

This small theoretical insight, although the values of the parameters are inaccurate, confirms qualitatively our experimental results.

POSSIBLE METHOD OF IMPROVING CONTROL AT LOW THROUGHPUTS

A *PI* single-loop feedback controller gave a satisfactory performance when used on a condenser that was operating at typical design flow rates and areas. It yielded poor control when the area was much larger than necessary, which is the case, for example, at low throughputs. Since tight enough settings to give good control at design rates produce instability at low throughputs, it is necessary to run the controller with very conservative settings in order to remain stable at all times. Consequently, at design conditions the control will not be as good as in Figure 9.

Since the heat exchange area is a major parameter in the control of the condenser, it would certainly be useful to be able to vary the condenser area with throughput. A piping configuration such as the one in Figure 11 could be used. Control valves on the draw-offs of the cooling water from the shell allow the effective area of the condenser to change. Vapor flow rate, reflux flow rate, water

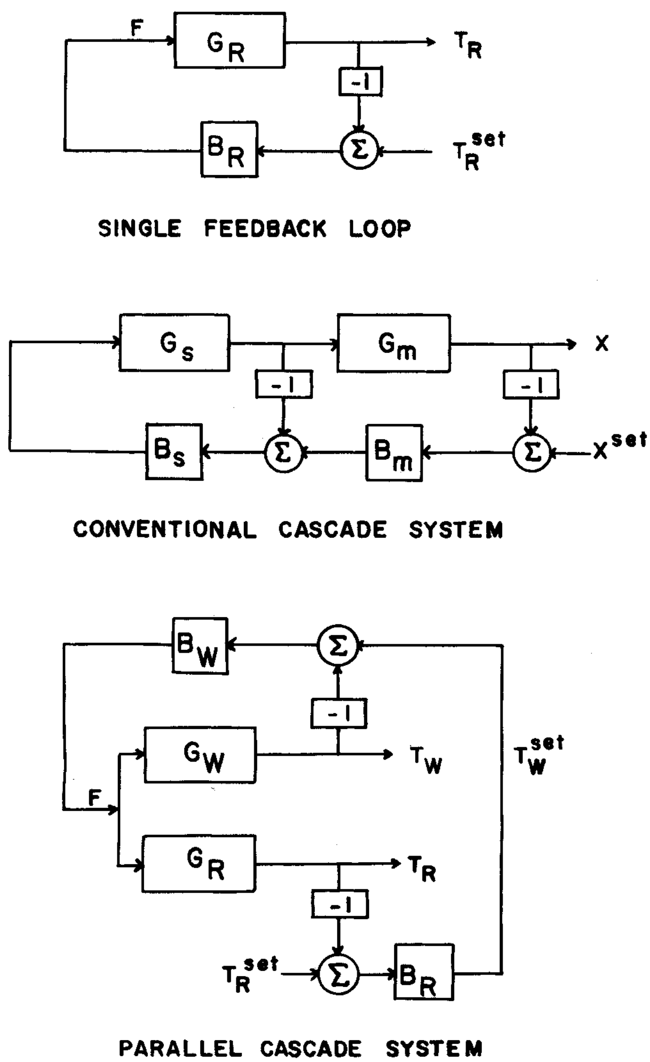

Fig. 8. Control schemes block diagrams.

flow rate or exit water temperature could be used to adjust the water exit position. This adjustment could be done automatically or manually when operating at different throughputs.

The use of feedforward control could be used to improve the load response. A nonlinear feedback controller might also improve loop performance.

NOTATION

B_R = reflux temperature controller transfer function
B_W = water temperature controller transfer function
F = water flow rate
G_R = process transfer function, T_R/F
G_W = water transfer function, T_W/F
Kc_R = reflux temperature controller gain = Kc_W
Kc_W = water temperature controller gain = K_R
K_R = process gain = Kc_R
K_W = water transfer function gain
Ku = ultimate gain
Lc = latent heat of condensation
P = proportional controller
PI = proportional-integral controller
Pu = ultimate period

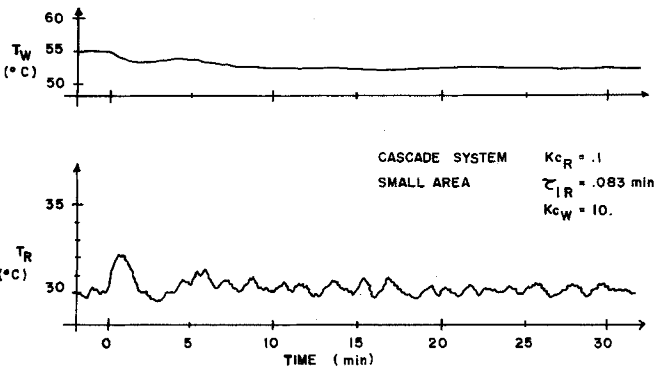

Fig. 9. Cascade system load response—small area.

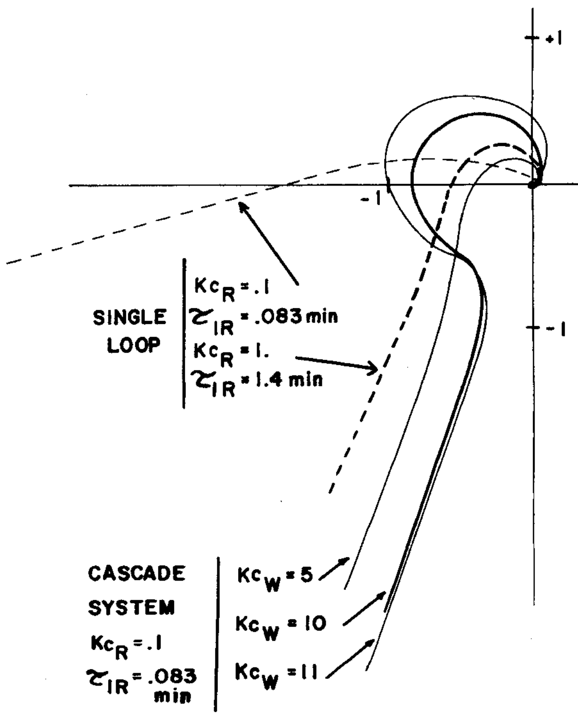

Fig. 10. Polar plot—single loop and cascade systems.

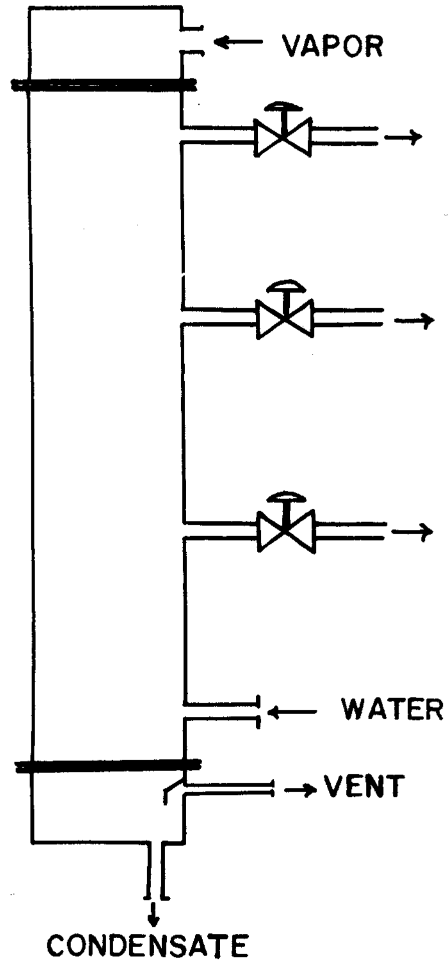

Fig. 11. Proposed system modification.

R = reflux flow rate
T_R = reflux temperature
T_W = water exit temperature
τ_{IR} = reflux temperature controller reset time

LITERATURE CITED

Bell, K. J., "Temperature Profile in Condensers," *Chem. Eng. Prog.*, **68**, 81 (1972).

Bergelin, O. P., G. A. Brown, and S. C. Doberstein, "Heat Transfer and Fluid Friction during Flow across Banks of Tubes," *Trans. ASME*, **74**, 953 (1952).

Colburn, A. P., and O. A. Hougen, "Design of Cooler Condensers for Mixtures of Vapor with Noncondensing Gases," *Ind. Eng. Chem.*, **26**, 1178 (1934).

Heidemann, R. A., et al., "Dynamics of Convective Heat Exchangers," *Can. J. Chem. Eng.*, **49**, 147 (1971).

Hewitt, G. F., and N. S. Hall-Taylor, "Annular Two-Phase Flow," pp. 188-191, Pergamon Press, Oxford (1970).

Kern, D. Q., *Process Heat Transfer*, pp. 256-263, McGraw-Hill, New York (1950).

Rohsenow, W. M., "Heat Transfer and Temperature Distribution in Laminar-Film Condensation," *Trans. ASME*, **78**, 1645 (1956).

Stainthorp, F. P., and A. G. Axon, "The Dynamic Behavior of a Multipass Steam-Heated Exchanger," *Chem. Eng. Sci.*, **20**, 107 (1965).

Williams, T. J., "Computers and Process Control," *Ind. Eng. Chem.*, **59**, 12, 53 (1967).

———., and H. J. Morris, "A Survey of the Literature on Heat Exchanger Dynamics and Control," *Chem. Eng. Prog. Symp. Ser. No. 37*, **57**, 20 (1961).

Manuscript received January 22, 1973; revision received March 8 and accepted April 18, 1973.