ANALYSIS OF A FLOODED HEAT EXCHANGER

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Understanding heat-transfer phenomena is a vital part of all fields of engineering education. Exposure to the theory and practice of heat transfer is provided in all undergraduate chemical engineering departments. Most departments also have heat-transfer experiments to enhance understanding of basic concepts.

The Chemical Processing Laboratory at Lehigh University has several heat-transfer experiments, some operating at steady state conditions and some operating in dynamic batch mode. Experimental data are collected and analyzed to calculate heat-transfer rates, differential temperature driving forces, and overall heat-transfer coefficients under various conditions (Reynolds Numbers).

One of our most effective experiments has two pilot-plant-scale tube-in-shell heat exchangers operating in series as shown in Figure 1. This experiment is used for both steady-state and dynamic experiments, as discussed in detail in Luyben, Tuzla, and Bader[1] in the two senior chemical processing laboratory courses. The students have taken a course in heat transfer in the first semester of the junior year.

Process water is pumped from a tank into the tubes of the first heat exchanger, which is heated by steam on the shell side. Condensate leaves through a steam trap, discharging into a drain. The process water then flows into the tubes of a downstream heat exchanger, which is cooled by cooling water. The process water flows back into the feed tank. All three flows are controlled by control valves. The steam valve is "air-to-open" and is located in the steam line upstream of the heater. The other two valves are "air-to-close."

Stream temperatures are measured at the inlet and outlet conditions of all heat exchangers, using redundant dial and thermocouple sensors (to demonstrate the uncertainties in temperature measurements). Steam pressures are measured at the supply header and inside the shell of the heater. Flowrates of the process water, cooling water, and steam condensate are measured by the old reliable "bucket and stopwatch" method.

The experiment is started up following the sequence of establishing cooling water, then starting process water circulation, and finally slowly opening the steam valve. For a fixed process water flowrate, the steam valve is opened until the steam pressure in the shell side of the heater is about 15 psig. At steady-state conditions, the experimental data are used to calculate heat-transfer rates on both sides of both heat exchangers. In theory, these four duties should all be the same at steady state, but experimental measurement errors and small heat losses to the atmosphere produce results that do not match exactly. Some data reconciliation is required before overall heat-transfer coefficients can be calculated. The dependence of the coefficients on the flowrate of process water through the two heat exchangers is determined in a series of appropriate experiments.

FLOODED HEAT EXCHANGERS

Our experimental heater normally operates with the shell completely full of vapor. As the steam condenses, it flows out the bottom of the shell into the steam trap, which only permits liquid condensate to exit into the drain. So the heat-transfer area exposed to condensing vapor is the entire heat-transfer area, which is the total number of tubes times the surface area per tube (π D L).

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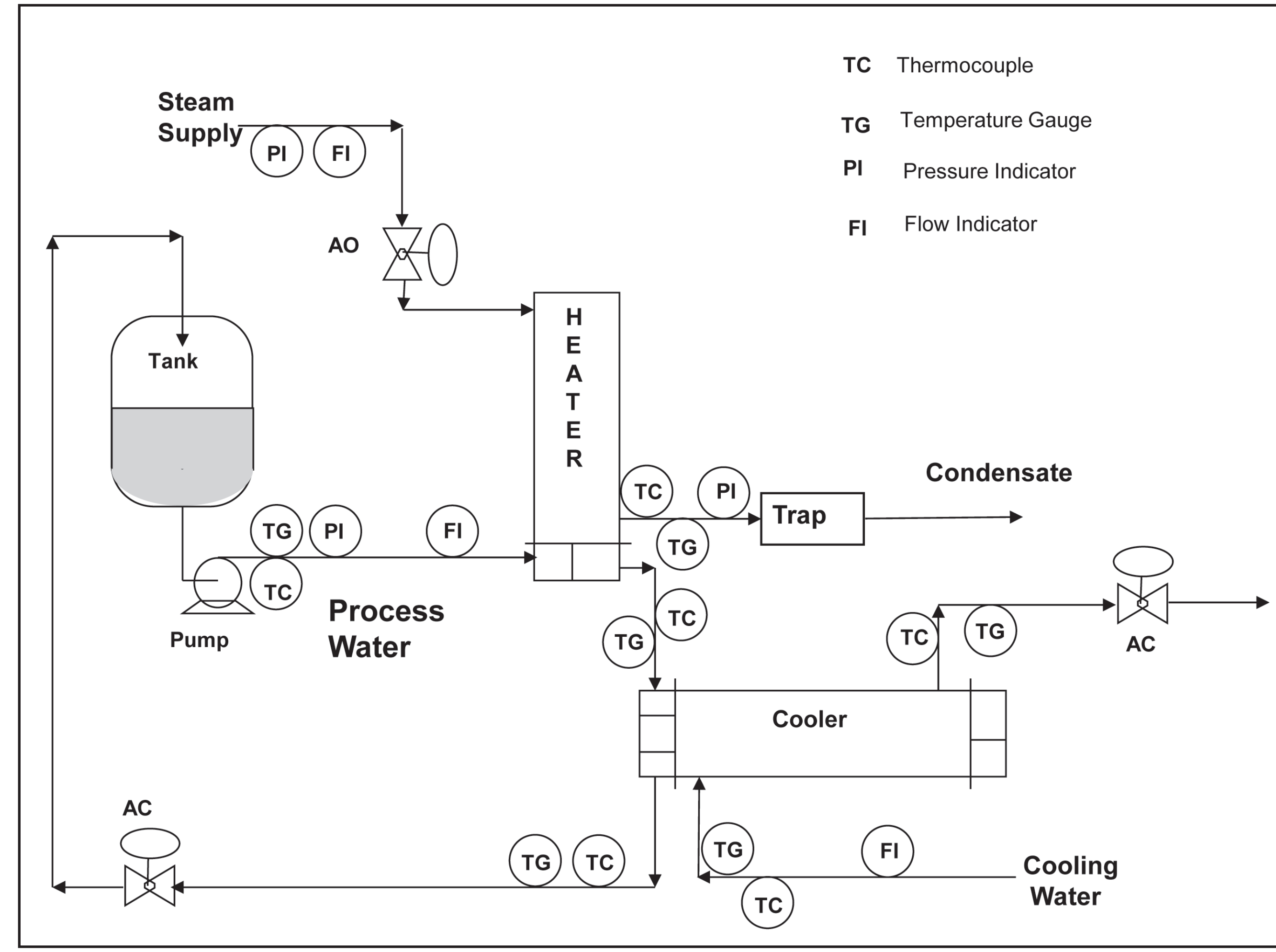

Figure 1. *Flowsheet.*

In this situation, the heat-transfer rate is changed by changing the position of the upstream steam valve, which changes the pressure (and temperature) on the steam side of the heat exchanger. Thus different heat-transfer rates are achieved by altering the differential temperature driving force (ΔT_{LM}) between the hot condensing steam in the shell and the colder process water in the tubes. Heat-transfer area is fixed. The differential temperature driving force between the steam and the cooler process water changes.

An alternative configuration is sometimes used in industry. Instead of having a large steam valve upstream of the heat exchanger with gas flowing through it, a smaller control valve is located on the exit liquid condensate line with liquid flowing through it. There is no steam trap (or condensate drum with its liquid level held by a control valve in the liquid exit line). Liquid condensate collects in the bottom of the shell.

The steam pressure in the shell is at the full supply pressure. The liquid level is adjusted to alter the heat-transfer rate by changing liquid level and hence the heat-transfer area. Thus in this configuration, differential temperature (ΔT) is fixed. The heat-transfer area changes.

The flooded design has the advantage that a smaller heat-transfer area is required because of the larger ΔT driving force. The steam temperature in the shell is the temperature of saturated steam at the full steam supply pressure. It is not reduced because of pressure drop over a control valve. Another advantage is the smaller valve required in the liquid line compared to a larger valve on the vapor line.

The flooded design has some dynamic control disadvantages. Because it takes some time to change the liquid holdup in the shell, there can be an undesirable dynamic lag in a control loop that is using heat input to control some variable.

The discussion above has considered a steam-heated heat exchanger. The same situation occurs when a process vapor is being condensed by a cooling medium. Flooded condensers are sometimes used in distillation columns and reactors.

References 2 and 3 give information about flooded heat exchangers.

LEHIGH FLOODED HEAT EXCHANGER

The heater is mounted vertically with 54 tubes (14.1 inches in length and 0.25 inches in outside diameter). The process water inlet and outlet are located at the bottom of the vessel. There are two tube passes, which means that the process water first flows up through 27 tubes and then down through 27 tubes. Figure 2 gives a sketch of the system during flooded operation.

To achieve flooded conditions, we start from normal steady-state conditions with the process water valve opened wide and the steam valve positioned (about 50% open) to achieve 15 psig in the shell. Then the steam valve opening is gradually reduced. This results in the steam pressure in the heater gradually dropping. At some point the pressure gauge shows zero psig. Further reduction of the steam valve opening produces a liquid level in the sight glass installed on the side of the heater.

The liquid condensate backs up into the heater because the pressure at the trap is 14.7 psia and the pressure of the steam inside the heater is lower than atmospheric. Enough hydraulic head (ρh) must exist for the liquid to leave the system.

In the numerical example presented below, the elevation of the bottom of the heater above the trap is 20 inches and the liquid level in the sight glass is 8 inches above the base of the heater. The total liquid height between the trap exit and the top of the liquid in the shell of the heater is 28 inches. Therefore the absolute pressure inside the heater shell is

$$P_S = 14.7 \text{ psia} - \left(\frac{28}{12}\text{ft}\right)\left(62.3 \text{ lb/ft}^3\right)\left(\text{ft}^2/144 \text{ in}^2\right) = 13.69 \text{ psia} \quad (1)$$

At this pressure, the saturation temperature of steam is 208.4 ˚F (T_S = 98.00 ˚C). Therefore, steam is condensing at 98.00 ˚C and the temperature of the liquid at the top of the condensate inside the heater is 98.00 ˚C. This pressure cannot be measured because the pressure gauge does not display vacuum pressures.

The temperature of the condensate leaving the trap is measured (38.9 ˚C) and the flowrate of the condensate is also measured (0.012 kg/s). So the sensible heat removed from the liquid in the bottom of the heater can be calculated.

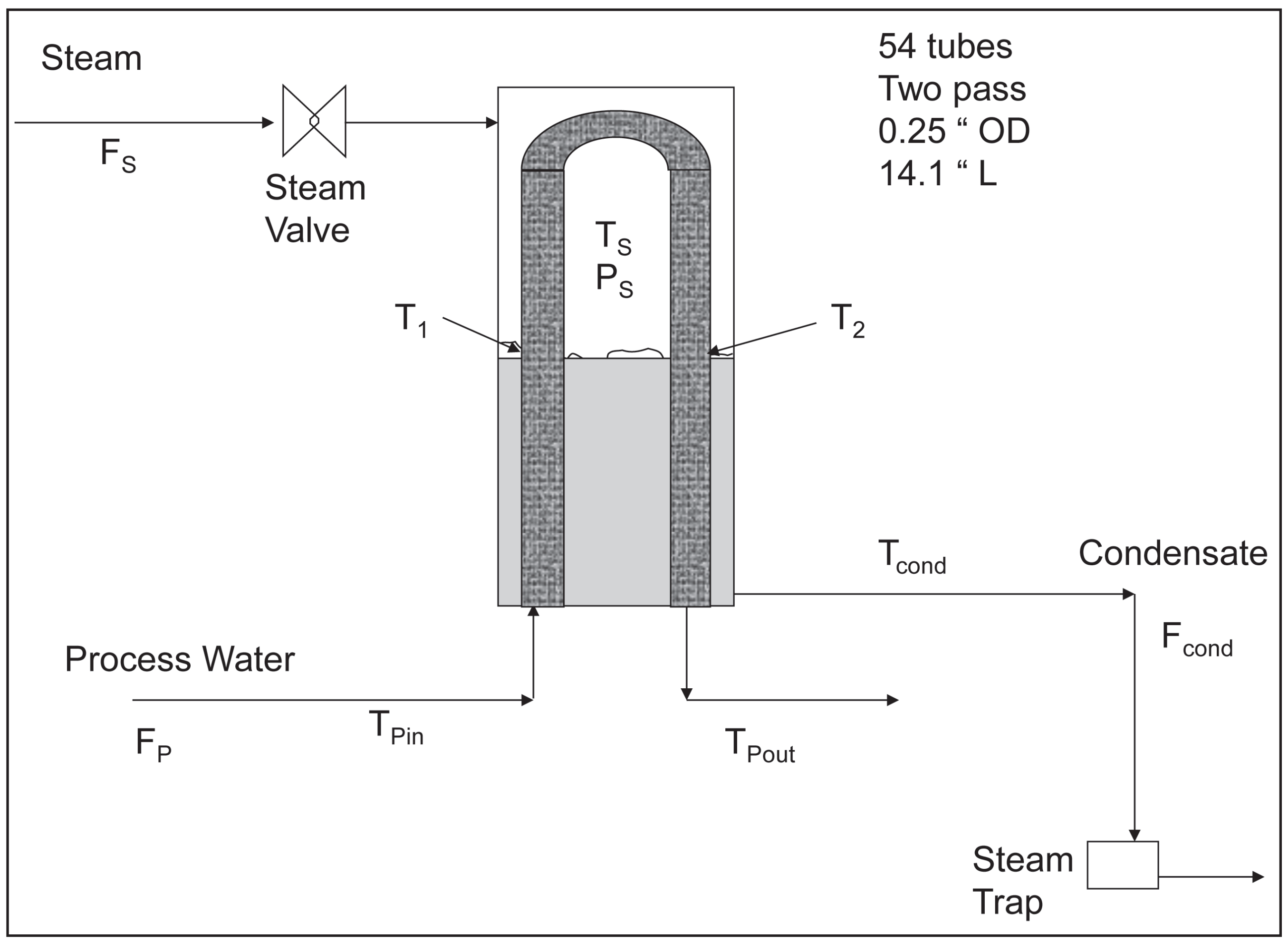

Figure 2. *Flooded heater.*

$$\begin{aligned} Q_L &= F_{cond} C_p \left(T_S - T_{cond}\right) \quad (2)\\ &= (0.012 \text{ kg/s})(4.18 \text{ kJ/kg} - \text{C}^\circ)(98.00 - 38.9 \text{ ˚C})\\ &= 2.964 \text{ kW} \end{aligned}$$

Some of this heat is removed as the process water flows up through half of the tubes (Q_{L1}) and some as it flows down through the other half (Q_{L2}). We do not know either of these heat-transfer rates, but we know they total Q_L.

The *total* heat transferred is calculated from the measured process water flowrate and its measured inlet and outlet temperatures.

$$\begin{aligned} Q_{Total} &= F_P C_p \left(T_{Pout} - T_{Pin}\right) \quad (3)\\ &= (1.31 \text{ kg/s})(4.18 \text{ kJ/kg} - \text{C}^\circ)(36.9 - 30.0 \text{ ˚C})\\ &= 37.78 \text{ kW} \end{aligned}$$

The heat transferred from the condensing steam vapor in the upper part of the heater above the pool of liquid condensate must be the difference between the total and that transferred to the liquid.

$$\begin{aligned} Q_{Total} &= Q_V + Q_L \quad (4)\\ Q_V &= 37.78 - 2.964 = 34.82 \text{ kW} \end{aligned}$$

The total heat-transfer area is calculated from the physical dimensions of the 54 tubes.

$$\begin{aligned} A_{Total} &= (54 \text{ tubes})\pi(0.25 \text{ inch})(0.0254 \text{ m/inch})^2(14.1 \text{ in})\\ &= 0.386 \text{ m}^2 \quad (5) \end{aligned}$$

If the liquid level in the heater shell is 8 inches and the total tube length is 14.1 inches, the heat-transfer area covered by liquid is

$$A_L = \left(\frac{8}{14.1}\right)(0.386) = 0.219 \text{ m}^2 \quad (6)$$

Half of A_L is where the process water is flowing upwards, starting from the inlet temperature T_{Pin} = 30.0 ˚C and entering the vapor space at some unknown temperature we call T_1. The other half of A_L is where the process water is flowing downwards, starting from some unknown temperature we call T_2 and leaving the heater at the process water exit temperature T_{Pout} = 36.9 ˚C.

The heat-transfer area exposed to condensing steam A_V is

$$A_V = A_{Total} - A_L = 0.386 - 0.219 = 0.167 \text{ m}^2 \quad (7)$$

All of the experimentally measured data and several calculated variables have now been presented. We would like to calculate the overall heat-transfer coefficients in the vapor and liquid phases (U_V and U_L).

SIMULTANEOUS NONLINEAR EQUATIONS

The *known* variables presented in the previous section are Q_V, Q_L, F_P, F_{cond}, C_P, A_V, A_L, T_S, T_{cond}, T_{Pin}, and T_{Pout}. The five *unknown* variables are T_1, T_2, U_V, U_L, and Q_{L1}. The overall heat-transfer coefficients in the vapor and liquid spaces (U_V and U_L) are unknown, but we would expect U_V to be much larger because condensing steam should have a large film coefficient compared to stagnant liquid.

The five equations describing the system are listed below.

$$Q_V = F_P C_P \left(T_2 - T_1\right) \quad (8)$$

$$Q_V = U_V A_V \left[\frac{(T_S - T_2) - (T_S - T_1)}{\ln\left(\frac{(T_S - T_2)}{(T_S - T_1)}\right)}\right] \quad (9)$$

$$Q_{L1} = U_L (0.5) A_L \left[\frac{(T_S - T_1) - (T_{cond} - T_{Pin})}{\ln\left(\frac{(T_S - T_1)}{(T_{cond} - T_{Pin})}\right)}\right] \quad (10)$$

$$Q_L - Q_{L1} = U_L (0.5) A_L \left[\frac{(T_S - T_2) - (T_{cond} - T_{Pout})}{\ln\left(\frac{(T_S - T_2)}{(T_{cond} - T_{Pout})}\right)}\right] \quad (11)$$

$$Q_{L1} = F_P C_P \left(T_1 - T_{Pin}\right) \quad (12)$$

Table 1 shows the Matlab program used to solve these five simultaneous nonlinear equations using the Matlab function "fsolve."

The results are:

T_1 = 30.34 ˚C

T_2 = 36.70 ˚C

U_V = 3.236 kW K^{-1} m^{-2}

U_L = 0.5845 kW K^{-1} m^{-2}

Q_{L1} = 1.855 kW

Table 2 gives the Mathcad program.

By way of an approximate confirmation of these results, the overall heat-transfer coefficient found for the heater in normal un-flooded operation at the same process water flowrate is 3.07 kW K^{-1} m^{-2}.

CONCLUSION

The flooded heater provides an excellent experiment to demonstrate basic principles of heat transfer in a non-conventional unit. It reinforces fundamental understanding

TABLE 1
Matlab Program

```
% Program "floodmain.m"
%
% Uses fsolve to find steady-state flooded heater variables
%
% Unknowns are 5:  t1. t2, uvap. uliq, qliq1
%
clear
% fixed variables
fp=1.31;fs=0.012;cp=4.18;tpin=30;tpout=36.9;ts=98.06;tcond=38.9;
atot=0.386;aliq=0.219;avap=0.167;
% calc variables
qtot=fp*cp*(tpout - tpin);qliq=fs*cp*(ts-tcond);qvap=qtot-qliq;
% Initial guesses of 5 unknowns:
xo=[30.6 36.9 1.5 .4 1]';
options=optimset('MaxFunEvals',1000);
% Use fsolve to solve 5 nonlinear algebraic equations
[x,fval,exitflag,parameters]=fsolve(@floodfunc,xo,options);
 t1=x(1,1),t2=x(2,1),uvap=x(3,1),uliq=x(4,1),qliq1=x(5,1),
  qtot,qvap,qliq


% function "floodfunc.m"
% Use fsolve to find solution of 5 nonlinear algebraic equations
function f=floodfunc(x)
% Unknowns are  t1, t2, uvap, uliq, qliq1
t1=x(1,1);t2=x(2,1);uvap=x(3,1);uliq=x(4,1);qliq1=x(5,1);
% fixed variables
fp=1.31;fs=0.012;cp=4.18;tpin=30;tpout=36.9;ts=98.06;tcond=38.9;
atot=0.386;aliq=0.219;avap=0.167;
% calc variables
qtot=fp*cp*(tpout - tpin);qliq=fs*cp*(ts-tcond);qvap=qtot-qliq;
% 5 equations
f(1,1)=qvap-fp*cp*(t2-t1);
f(2,1)=qvap-uvap*avap*((ts-t2)-(ts-t1))/log((ts-t2)/(ts-t1));
f(3,1)=qliq1-uliq*aliq*0.5*((ts-t1)-(tcond-tpin))/log((ts-t1)/(tcond-tpin));
f(4,1)=qliq-qliq1-uliq*aliq*0.5*((ts-t2  )-(tcond-tpout))/log((ts-t2)/(tcond-
tpout));
f(5,1)=qliq1-fp*cp*(t1-tpin);
```


TABLE 2
MATHCAD PROGRAM

Fixed Variables

$t_{pin} := 303.15K$ $\quad t_{pout} := 310.05K$ $\quad t_{cond} := 312.05K$ $\quad t_s := 371.21K$

$f_p := 1.31\frac{kg}{s}$ $\quad f_s := 0.012\frac{kg}{s}$ $\quad c_p := 4.18\frac{kJ}{kg \cdot K}$

$a_{tot} := 0.386m^2$ $\quad a_{liq} := 0.219\ m^2$ $\quad a_{vap} := 0.167\ m^2$

Calculated Variables

$q_{tot} := f_p \cdot c_p \cdot (t_{pout} - t_{pin})$ $\quad q_{liq} := f_s \cdot c_p \cdot (t_s - t_{cond})$ $\quad q_{vap} := q_{tot} - q_{liq}$

Initial Guess

$t_1 := 305K$ $\quad t_2 := 307K$ $\quad q_{liql} := 1kW$ $\quad u_{liq} := 1.5\frac{kW}{m^2K}$ $\quad u_{vap} := 0.4\frac{kW}{m^2K}$

Given

$$q_{liql} = f_p \cdot c_p \cdot (t_1 - t_{pin}) \qquad q_{liql} = u_{liq} \cdot \frac{a_{liq}}{2} \cdot \left[\frac{(t_s - t_1) - (t_{cond} - t_{pin})}{\ln\left[\frac{(t_s - t_1)}{(t_{cond} - t_{pin})}\right]}\right]$$

$$q_{vap} = f_p \cdot c_p \cdot (t_2 - t_1) \qquad q_{vap} = u_{vap} \cdot a_{vap} \cdot \left[\frac{(t_s - t_2) - (t_s - t_1)}{\ln\left[\frac{(t_s - t_2)}{(t_s - t_1)}\right]}\right]$$

$$q_{liq} - q_{liql} = u_{liq} \cdot \frac{a_{liq}}{2} \cdot \left[\frac{(t_s - t_2) - (t_{cond} - t_{pout})}{\ln\left[\frac{(t_s - t_2)}{(t_{cond} - t_{pout})}\right]}\right]$$

Find Block

$$\begin{pmatrix} t_1 \\ t_2 \\ u_{vap} \\ u_{liq} \\ q_{liql} \end{pmatrix} := \text{find}(t_1, t_2, u_{vap}, u_{liq}, q_{liql})$$

Solutions

$t_1 = 303.489K$ $\quad u_{vap} = 3.233\frac{kW}{m^2 \cdot K}$ $\quad q_{liql} = 1.857kW$

$t_2 = 309.847K$ $\quad u_{liq} = 0.585\frac{kW}{m^2 \cdot K}$

of convective heat transfer in a realistically complex process unit in which phase changes are occurring.

NOMENCLATURE

A_L – total heat-transfer area covered by liquid (m^2)
A_{Total} – total heat-transfer area of 54 tubes (m^2)
A_V – total heat-transfer area exposed to steam (m^2)
C_p – heat capacity of water (kJ kg^{-1} K^{-1})
F_{cond} – flowrate of condensate and steam (kg/s)
F_P – flowrate of process stream (kg/s)
P_S – steam pressure in shell (psia)
Q_L – total energy transferred from condensate in lower section (kW)
Q_{L1} – energy transferred from condensate into process stream as it flows up from the process inlet to the vapor/liquid interface in the lower section (kW)
Q_{L2} – energy transferred from condensate into process stream as it flows down from the vapor/liquid interface to the process exit (kW)
Q_{Total} – total energy transferred into process stream (kW)
Q_V – energy transferred from condensing steam in upper section (kW)
T_1 – temperature of process stream as it leaves the top of the liquid flooded section and enters the vapor-heated section (˚C)
T_2 – temperature of process stream as it leaves the top vapor-heated section and enters the liquid flooded section (˚C)
T_{cond} – condensate temperature leaving shell side of heater (˚C)
T_{Pin} – temperature of process stream into heater (˚C)
T_{Pout} – temperature of process stream out of heater (˚C)
T_S – saturation temperature of steam at P_S (˚C)
U_L – overall heat-transfer coefficient in lower region where liquid covers tubes (kW K^{-1} m^{-2})
U_V – overall heat-transfer coefficient in upper vapor region where steam is condensing (kW K^{-1} m^{-2})

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