The Operation of Direct-contact Condenser at Thermodynamic Equilibrium

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Abstract

Multiple-effect evaporators and crystallizers in the sugar industry operate under vacuum, which may be generated by using direct-contact condensers. The condenser uses cooling water to condense the vapor from the evaporator or the crystallizer, whereas a vacuum pump removes both uncondensed vapor and incondensable gases from the condenser, which results in sub-atmospheric pressure in the system. Factors influencing the amount of vacuum that can be achieved include the flow rate of cooling water, the water temperature, and the flow rate of mixture removed by the vacuum pump. There has been limited analytical study of the system of evaporator and condenser. Therefore, the understanding of the influences of these factors is mostly qualitative. This paper aims at quantifying factors influencing the performance of the system of evaporator and condenser using the assumption that system is at thermodynamic equilibrium. Results from this study should enable the control of vacuum in the system to be more efficient.

Keywords : vacuum, condenser, thermodynamics

1. Introduction

Many industrial processes require evaporation of water at sub-atmospheric pressures. An example is the evaporation process in the production of raw sugar in a crystallizer and the last effect of a multiple-effect evaporator. Benefits of evaporation under vacuum condition are twofold. First, it allows the input steam to be used as a heating medium many times before it becomes condensate. As a result, the amount of removed water exceeds the amount of input steam. In other words, the steam economy will be greater than unity. Second, evaporation at a low pressure occurs at also a low temperature, which is beneficial to the quality of produced raw sugar.

Because the pressure of saturated vapor is a function of temperature, the reduction of pressure can be achieved by decreasing the vapor temperature. With sufficient temperature reduction and the corresponding pressure reduction, a vacuum will be created. The equipment used for this task is a condenser. The type of condenser normally used in sugar factories is the direct-contact condenser, which has advantages of cheap construction and the ability to achieve a close approach of temperature. The direct-contact condenser operates by mixing vapor with cooling water from either spray ponds or cooling towers.

Normally, not all of the vapor is condensed by cooling water. The remaining vapor must be removed by a vacuum pump. In addition to removing uncondensed vapor, the vacuum pump also removes incondensable gases. Sources of incondensable gases are dissolved gases in cooling water, air leaking into the crystallizer or the evaporator, and gases dissolved or entrained in the juice or syrup. Unless removed, the accumulation these gases will keep on rising, interfering with the condensation process and raising condenser pressure.

Three important factors that affect the vacuum condition in the condenser are the cooling water temperature, the cooling water flow rate, and the volumetric flow rate of the vapor and gases removed by the vacuum pump. Since the first one is uncontrollable, the control of a vacuum in the condenser requires the control of the last two factors. It is a common practice to vary the cooling water flow rate in response to changing cooling water temperature. It is

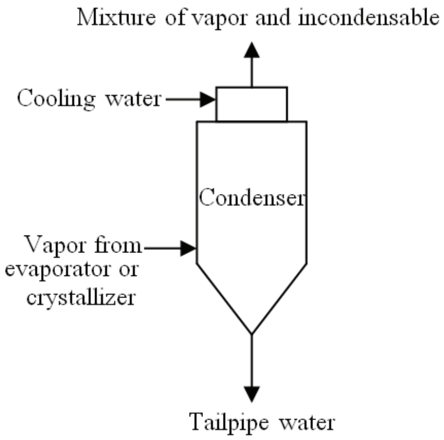

Figure 1: Inputs and outputs of direct-contact condenser

well known qualitatively that, with a fixed vacuum, required cooling water flow increases with cooling water temperature. The objective of this paper is to derive quantitative relationships between condenser pressure and the three factors that affect the pressure based on the analysis of the steady-state operation of direct-contact condenser.

2. Description of the System

Figure 1 illustrates the operation of the direct-contact condenser. Vapor from the evaporator and cooling water enter the condenser, whereas tailpipe water and the mixture of vapor and incondensable gases leave the condenser. The type of condenser shown in Fig. 1 is the counter-flow type because vapor flows upward, and cooling water flows downward. An alternative condenser type is the parallel-flow type, in which both vapor and cooling water flow downward. Condenser is designed so that all vapor is condensed with the minimum use of cooling water. Therefore, cooling water enters in forms of jets, sprays, or water curtain to promote mixing with vapor. Furthermore, trays or perforated plates are installed inside the condenser to increase the mixing time.

3. Analysis of the System

There has been limited publication related to the system of condenser and vacuum pump. Love [1] presents a simple model that can be used to study the performance of condenser under various conditions. The analysis in this paper is based on his work, but certain modifications are made in the proposed model of condenser. The following assumptions are made in the development of the model.

- The condenser is at thermodynamic equilibrium with the evaporator and the vacuum pump.
- Pressure drop across the condenser is negligible, since, according to Hugot [2], it is about 0.4 kPa.
- Vapor and air behave like ideal gases with constant specific heat capacities.
- All cooling water is mixed with vapor; no cooling water bypasses the mixing process.
- Mixing between cooling water and vapor is homogeneous. The mixture is saturated liquid water.
- There is no droplet carry-over in the mixture of vapor and incondensable gases at the exit of condenser.
- Thermodynamic properties of incondensable gases and air are identical.

The schematic representation of the operation of condenser is shown in Fig. 2. The vapor flow rate into the condenser (m_v) is the sum of vapor mixed with cooling water (m_{v1}) and unmixed vapor (m_{v2}). The latter is determined from mass balance at the exit from the condenser to the vacuum pump.

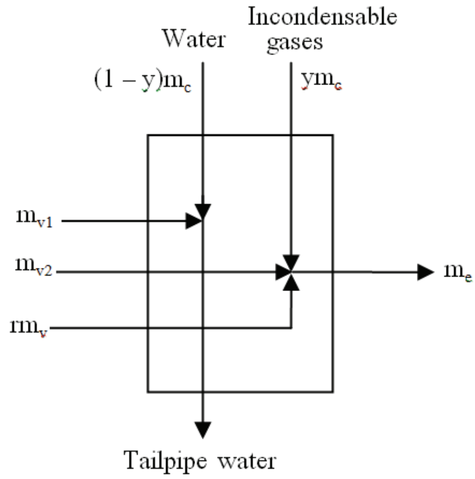

Figure 2: Mass flows in direct-contact condenser

$$m_{v2} = m_e - rm_v - ym_c \tag{1}$$

where r is the ratio of mass of incondensable gases to vapor mass, and y is the fraction of incondensable gases dissolved in cooling water. The thermodynamic equilibrium between the condenser and the vacuum pump implies that

$$p_e = \frac{p}{1+\left(M_w/M_a\right)\left(rm_v + ym_c\right)/m_{v2}} \tag{2}$$

The molecular weights of water and air are, respectively, $M_w = 18.00$ and $M_a = 28.97$.

Since the vapor at the exit of the condenser is saturated, the temperature of the mixture of vapor and incondensable gases leaving the condenser (T_e) can be determined from the corresponding pressure (p_e).

$$T_e = -227.03 + \frac{3816.44}{18.3036 - \ln\left(7.5p_e\right)} \tag{3}$$

The energy balance equation is used to find m_{v1}.

$$m_{v1} = \frac{1}{h_{vl}(T_i)}\Big\{(1-y)m_c c_{pw}(T_i - T_c) + m_{v2}\left[h_v(T_e) - h_v(T_i)\right] + rm_v c_{pa}(T_e - T_i) + ym_c c_{pa}(T_e - T_c)\Big\} \tag{4}$$

where c_{pw} is the specific heat capacity of water (4.18 kJ/kg.K) and c_{pa} is the specific heat capacity of air (1.00 kJ/kg.K). Specific heat of evaporation (h_{vl}) and enthalpy of saturated steam (h_v) are given by Rein [3].

$$h_{vl}(T) = 2492.9 - 2.0523T - 3.0752 \times 10^{-3}\, T^2 \tag{5}$$

$$h_v(T) = 2502.04 + 1.8125T + 2.585 \times 10^{-4}\, T^2 - 9.8 \times 10^{-6}\, T^3 \tag{6}$$

The thermodynamic equilibrium at the inlet to the condenser requires that the total pressure of the mixture is equal to the condenser pressure (p). Therefore, the vapor pressure (p_v) may be expressed as.

$$p_v = \frac{p}{1 + r\left(M_w/M_a\right)} \tag{7}$$

The corresponding vapor temperature (T_v) can be determined from pv using Eq. (3) because the vapor is assumed to be saturated. Since $m_v = m_{v1} + m_{v2}$, Eqs. (1) and (4) may be combined into

$$\{(1+r)h_{vl}(T_v) + r[h_v(T_e) - h_v(T_v) - c_{pa}(T_e - T_v)]\}m_v = [h_{vl}(T_v) + h_v(T_e) - h_v(T_v)]m_e + \{yc_{pa}(T_e - T_c) + (1-y)c_{pw}(T_v - T_c) - y[h_{vl}(T_v) + h_v(T_e) - h_v(T_v)]\}m_c \tag{8}$$

where m_c and T_c are the mass flow rate and the temperature of the cooling water, respectively.

The above analysis is based on the assumption that m_c is not too large because Eq. (1) indicates that m_{v2} may be negative. In reality, there must be a minimum amount vapor ($m_{v2,min}$) left at the exit of the condenser because the mixture exchanges heat with the cooling water, and the temperature of the mixture leaving the condenser cannot be lower than the cooling water temperature [4]. It occurs when mc is larger than the critical value ($m_{c,crit}$). Under this situation, the amount of cooling water involved in the mixing process is mc,crit, and the remaining cooling water bypasses the mixing process. The expression for $m_{v2,min}$ is as follows.

$$m_{v2,min} = \left(\frac{M_w/M_a}{p/p_c - 1}\right)\left(rm_v + ym_{c,crit}\right) \tag{9}$$

where p_c is the vapor pressure corresponding to T_c.

$$p_c = \frac{1}{7.5}\exp\left[18.3036 - \frac{3816.44}{\left(227.03 + T_c\right)}\right] \tag{10}$$

Equations (1) and (9) can now be solved for $m_{v2,min}$ and $m_{c,crit}$ in terms of m_v and m_e.

$$m_{v2,min} = \left[\frac{M_w p_c}{M_a\left(p - p_c\right) + M_w p_c}\right] m_e \tag{11}$$

$$m_{c,crit} = \frac{1}{y}\left[\frac{m_e}{\left(M_w/M_a\right)/\left(p/p_c - 1\right) + 1} - rm_v\right] \tag{12}$$

Substituting $m_{v2,min}$ and $m_{c,crit}$ from Eqs. (11) and (12), and $T_e = T_c$ yields

$$\left[h_{vl}(T_v)+yc_{pa}(T_v-T_c)+\frac{r}{y}(1-y)c_{pw}(T_v-T_c)\right]m_v =$$
$$\left\{\left[\frac{M_w p_c}{M_a(p-p_c)+M_w p_c}\right]\left[h_{vl}(T_v)+h_v(T_c)-h_v(T_v)\right]\right.$$
$$\left.\frac{1}{y}\left[\frac{(1-y)c_{pw}(T_v-T_c)}{M_w p_c/M_a(p-p_c)+1}\right]\right\}m_e \qquad (13)$$

Assume that m_v and p_v are fixed. Equation (8) provides a relationship among mc, T_c and m_e when $m_c < m_{c,crit}$, whereas Eq. (13) provides a relationship between T_c and me when $m_c > m_{c,crit}$. It should be noted that both equations are nonlinear. Solutions must, therefore, be obtained by iteration.

4. Results and Discussion

Values of $y = 3.5 \times 10^{-5}$ and $r = 0.003$ provided by Love [1] are used in the simulation of the operation of direct-contact condenser according to the model described in the previous section. Figure 3 shows the result of simulation when $m_v = 5$ kg/s and $T_c = 35$ °C. It can be seen that p and T_e decrease monotonically with m_c until T_e reaches 35 °C. After that, p and T_e remain constant because m_c reaches the critical value. A control system that uses the cooling water flow rate to control condenser pressure will, therefore, be effective only when the flow rate is less than the critical value. An attempt to control condenser when m_c exceeds $m_{c,crit}$ will result in a phenomenon known as "frozen condenser" as reported by Love [1].

In order to decrease condenser pressure after m_c reaches the critical value, m_e must be increased. It can be seen from Fig. 4 that p decreases as m_e increases. However, the effect of m_e on p is not noticeable before m_c reaches the critical value.

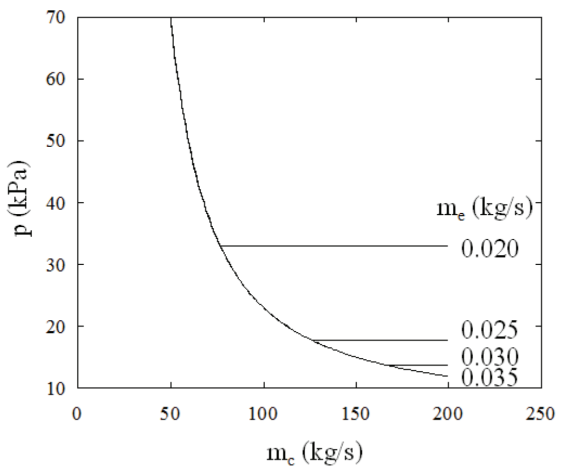

Figure 4: Effect of the mass flow rate exiting the condenser (m_e) on the variation of condenser pressure (p) with the mass flow rate cooling water (m_c)

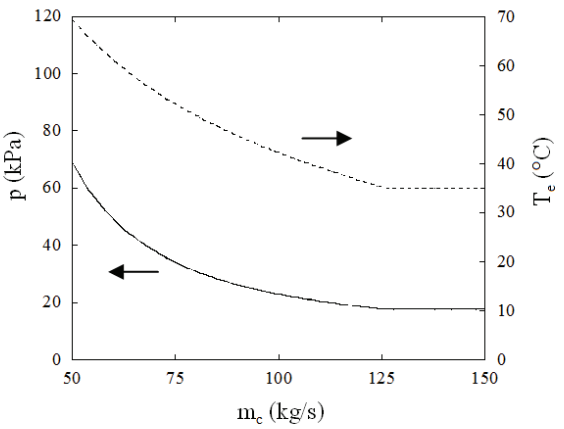

Figure 3: Variations of the condenser pressure (p) and the temperature of mixture leaving the condenser (T_e) with the mass flow rate cooling water (m_c)

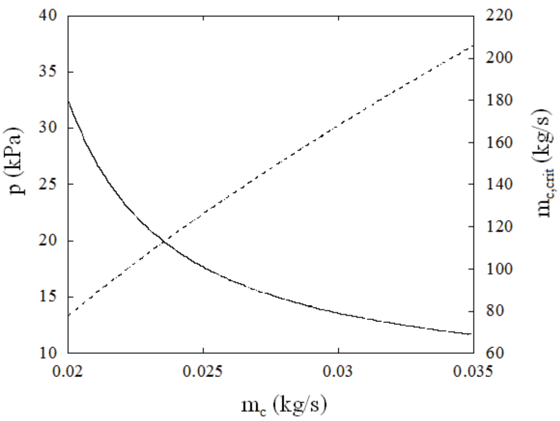

Figure 5: Variations of critical cooling water flow rate ($m_{c,crit}$) and the corresponding condenser pressure (p) with the mass flow rate exiting the condenser (m_e)

amount of cooling water involved in the mixing process with vapor is $m_{c,crit}$. The rest of cooling water becomes tailpipe water. When this occurs, only the increase of m_e can decrease p, as shown in Fig. 5. Also shown is the increase in $m_{c,crit}$ as m_e increases.

5. Conclusions

The understanding of the operation of direct-contact condensers has been mostly qualitative. The present study attempts to provide quantitative explanation of how condenser pressure are affected by factors such as cooling water flow rate, water temperature, and the flow rate of saturated gases removed by vacuum pump. The condenser is assumed to be at thermodynamic equilibrium, which means that gases at the inlet and outlet of the condenser are saturated with water vapor. Numerical results show that increasing water flow rate reduces condenser pressure and increases the temperature of saturated gases removed by vacuum pump until the temperature is equal to the cooling water temperature. After that, the water flow rate exceeds the critical value, and condenser pressure is longer affected by the water flow rate. In order to decrease condenser pressure when water flow rate is larger than the critical value, the vacuum pump must remove more saturated gases.

6. References

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