

Boiling and Condensation

- Convection with phase change
 - Solid-liquid interface (boiling)
 - Solid-vapor interface (condensation)
- Due to fluid motion, mechanism is convection
- However, temperature need not change as a result of heat absorbed/released during phase change (latent heat)

Important Parameters:

- Latent heat
- Surface tension
- Density difference

Chemical Process Equipments:

- Boiler, Evaporator, Condenser

Note: Boiling and Evaporation are different

- Evaporation: due to vapor pressure being lower than saturation pressure of the liquid
eg: Water at room temperature evaporates when relative humidity is less than 100%
- No bubbles are formed during evaporation
- At the liquid-vapor interface

- Boiling: Liquid contacted with a surface at $T_s > T_{sat}$,
 T_{sat} = saturation temperature of the liquid
 eg: Water in contact with a surface at 110°C at 1 atm pressure boils
- Rapid formation of bubbles is observed
- At the solid-liquid interface

Boiling

"Evaporation" at the solid-liquid interface.

Boiling heat transfer flux, using Newton's law of cooling, is

$$q = h(T_s - T_{sat}) = h \Delta T_{excess}$$

Fluid mechanics is complicated

- Vapor bubble growth and dynamics depend on excess temperature
- Vapor bubble dynamics affect liquid motion near the surface, and hence heat transfer
 \Rightarrow coupled heat transfer and fluid mechanics
- Analytical solutions are nearly absent

Types:

Pool Boiling: No bulk fluid flow by external means (natural convection and motion due to bubbles only)

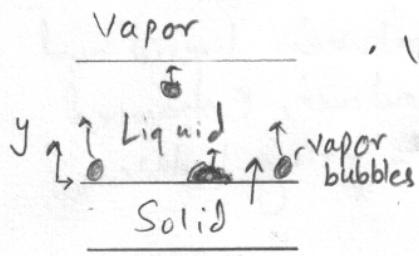
Forced Convection Boiling/Flow Boiling: Flow of fluid due to external means and natural convection and bubbles.

Subcooled boiling: $T_{bulk, liquid} < T_{sat, liquid}$
 \Rightarrow bubbles formed at the surface condense in the liquid

Saturated boiling: $T_{bulk, liquid} \geq T_{sat, liquid}$
 \Rightarrow bubbles formed at the surface rise to the top

Pool Boiling

- Boiling of "stationary" fluids



• Various regimes are identified within pool boiling, characterized by the boiling curve, available for many fluids, due to Nukiyama (1934)

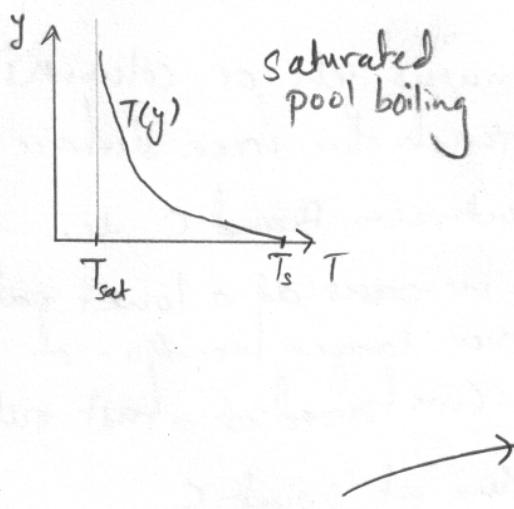
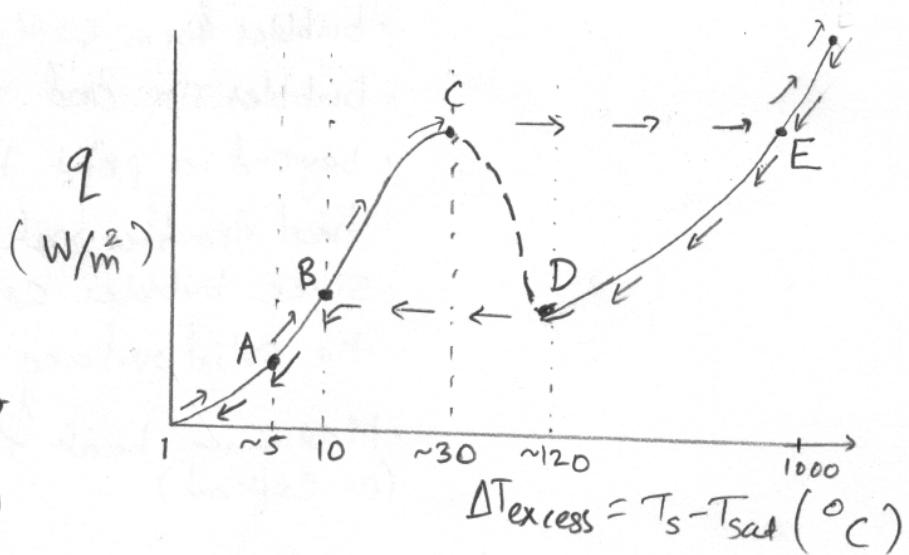


Figure 10.3 (Incopera)
or

Figure 10-6 (Engel)
(also Fig 10-8)



Schematic of boiling curve for water at 1 atm pressure

Four regimes

1. Natural convection

- upto point A
- no bubbles visible
- Slight Superheating
- Natural convection currents \Rightarrow fluid flow

2. Nucleate boiling

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- between A and C
- high heat transfer coefficient (and rate)
- bubbles form at surface
- Onset of Nucleate boiling at point A

Further divided into ^{'desirable' operating regime due to large heat transfer at moderate ΔT_{excess} .}

region between A and B

- isolated bubbles formed
- bubbles dissipate in liquid after leaving the solid surface
- heat transfer primarily between liquid and solid through direct contact, Enhanced by agitation due to the rising bubbles

region between B and C

- bubbles form continuous jets or columns
- bubbles rise and reach the free surface
- beyond a point between B and C, the heat transfer rate increases at a lower rate. Since bubbles cover larger fraction of the solid surface (are formed at a fast rate)
- Maximum heat flux at point C (or Critical)

3. Transition boiling

- between points C and D
- larger fraction of solid surface is covered by a "vapor film".
- partial nucleate and film boiling
- avoided in practice.

A. Film boiling

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- Surface of the solid is continuously covered by a film of vapor
- vapor film is continuous and stable
- beyond point D ($D \equiv$ Leidenfrost point)
- vapor film \Rightarrow increased resistance to heat transfer \Rightarrow lower heat flux
- Flux increases with ΔT_{excess} due to radiation heat transfer becoming significant

- Notes:
- In practice, q_s is the control variable (not T_s).
 - Thus, as q_s increases upto point C, T_s also increases. But beyond the critical heat flux (at point C), to increase T_s , one would increase q_s . However, liquid is not able to transfer heat from the surface at the same rate (or flux). \Rightarrow A sudden increase (jump) in T_s instead. If T_s exceeds beyond melting point of the solid surface before reaching point E, system failure will occur.
 - \Rightarrow point C is also called the "burnout point" or "boiling crisis".

Practically, operating as close to the critical heat flux as possible is desirable. However, the point C should not be exceeded.

Correlations in Pool Boiling

Natural Convection Boiling

• results of natural convection can be used

i.e.

$$Nu = f(Re, Pr)$$

Nucleate Boiling (Rohsenow, 1952)

$$q = \frac{M_l h_{fg}}{\sigma} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_{pl} (T_s - T_{sat})}{C_{sf} h_{fg} P_{\rho_l}^n} \right]^3$$

where M_l, ρ_l are viscosity and density of the liquid, respectively

ρ_v is the vapor's density, σ is the liquid-vapor interfacial tension

h_{fg} is the enthalpy of vaporization

C_{pl} is the specific heat capacity of the liquid

and C_{sf}, n are constants (Table 10-1 (Incropera's))
Table 10-3 (Engel's)

Note: • Only for clean smooth surfaces

• Involves errors upto $\pm 100\%$

Peak heat flux / Critical heat flux

$$q_{max} = C_{cr} h_{fg} \left[\sigma g \rho_v^2 v (\rho_l - \rho_v) \right]^{1/4}$$

with C_{cr} depending on geometry of the heating surface

large flat heating surface $C_{cr} = 0.149$

(Table 10-4 (Engel's) or Section 10.4.2 (Incropera's))

list the constant C_{cr} for large cylinders, spheres and also for small flat surfaces, cylinders and spheres).

Minimum heat flux at Leidenfrost point

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$$q_{\min} = 0.09 \rho_v h_{fg} \left[\frac{\sigma g (\ell_s - \ell_v)}{(\ell_s + \ell_v)^2} \right]^{1/4}$$

— Zuber, 1958,

Bergersen, 1961

Note: Estimate prone to
upto 50% error

Film boiling

$$q = C_{\text{film}} \left[\frac{g k_v l_v (\ell_s - \ell_v) [h_{fg} + 0.4 C_p v (T_s - T_{sat})]}{M_v D (T_s - T_{sat})} \right] (T_s - T_{sat})$$

with

k_v = thermal conductivity of the vapor

$C_{\text{film}} = 0.62$ for horizontal cylinders
 0.67 for spheres

With large ΔT_{excess} , radiation effects become significant.
Noting that radiation heat transfer increases film thickness,
convection heat transfer is adversely affected.

For $h_{\text{rad}} < h_{\text{conv}}$, Bromley, 1950 gave

$$h_{\text{total}} = h_{\text{conv}} + \frac{3}{4} h_{\text{rad}}$$

Example: Water boiling in a pan

Water is to be boiled in a mechanically polished stainless steel pan placed on top of a heating unit. The inner surface of the bottom of the pan is maintained at 108°C . If the diameter of the bottom of the pan is 30 cm, determine the rate of heat transfer to the water and rate of evaporation of water.

Assume a steady state and ignoring heat losses from the pan.
Properties of water at the saturation temperature of 100°C

$$\sigma = 0.0589 \text{ N/m} \quad \rho_s = 957.9 \text{ kg/m}^3 \quad \rho_v = 0.6 \text{ kg/m}^3$$

$$h_{fg} = 2257 \times 10^3 \text{ J/kg} \quad \mu_s = 0.282 \times 10^3 \text{ kg/m.s} \quad Pr_s = 1.75$$

$$C_p = 4217 \text{ J/kg.K}$$

$\epsilon_{st} = 0.013$ and $n = 1.0$ for water-stainless steel mechanically polished

$$\Delta T_{excess} = T_s - T_{sat} = 8^{\circ}\text{C} \Rightarrow \text{nucleate boiling}$$

Rohsenow relation gives

$$q = \mu_s h_{fg} \left[\frac{g(\rho_s - \rho_v)}{\sigma} \right]^{1/2} \left[\frac{C_p(T_s - T_{sat})}{\epsilon_{st} h_{fg} P_{st}^n} \right]^3$$

$$= 7.21 \times 10^4 \text{ W/m}^2$$

Area at the bottom of the pan

$$A = \pi D^2 / 4 = 0.07069 \text{ m}^2$$

$$Q = Aq = 5097 \text{ W}$$

The rate of evaporation

$$m = \frac{Q}{h_{fg}} = \frac{5097}{2257 \times 10^3} \text{ kg/s} = 2.26 \times 10^{-3} \text{ kg/s}$$

Flow Boiling / forced Convection Boiling

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External flow boiling : over surfaces,

Internal flow boiling: inside tubes

Complicated

known as two phase flow
boiling.

flow increases the heat flux.

Condensation

- Temperature of vapor is below its saturation temperature.
- Vapor in contact with a solid surface at temperature $T_s < T_{sat}$
also in absence of a solid surface.
such as free surface of a liquid or
even in gas at a lower temperature than T_{sat} .

Types :

Film Condensation

- Condensate: wets the surface, forming a film that slides down due to gravity

Droopwise condensation

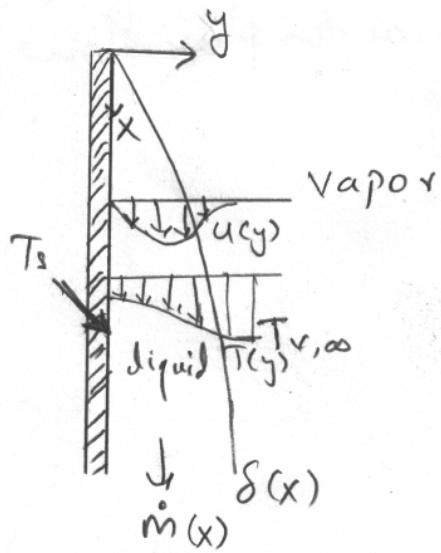
- Condensate is in form of droplets on the surface

Heat transfer rates are significantly high in droopwise condensation.

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However, it is difficult to sustain the drop wise condensation in its form (by modifying surfaces) and eventually a film is formed in most practical applications.

Film Condensation



Consider a vertical flat plate at surface temperature $T_s < T_{sat}$

- Liquid film forms, with thickness increasing with x .
- Velocity and temperature profiles are shown.
- $T = T_{sat}$ at $y = \delta(x)$

Depending on whether the flow is laminar or turbulent (analogy with forced convection), heat transfer is different.

$$Re = \frac{D_h \rho V_l}{\mu_l} = \left(\frac{A A_c}{P}\right) \frac{\rho_l V_l}{\mu_l}$$

Now $A_c = P \delta$, with δ = lowest part of the flow

$$\Rightarrow Re = A \delta \frac{\rho V_l}{\mu_l} = \frac{4 \dot{m}}{P \mu_l}$$

with \dot{m} = mass flow rate of the condensate at the lowest part of flow

Actual heat transfer = latent heat's contribution + extra cooling of the condensate (between T_s & T_{sat})

$$h_{fg}^* = h_{fg} + 0.68 C_p (T_{sat} - T_s) \quad \text{Rohsenow, 1956}$$

Where \bar{h}_{fg}^* = modified latent heat of vaporization

For superheated vapor entering the condenser, similarly, vapor has to be cooled to T_{sat} before condensing. Thus,

$$\bar{h}_{fg}^* = h_{fg} + 0.68 \rho_1 (T_{sat} - T_s) + \rho_1 (T_v - T_{sat})$$

Then,

$$Q = h A_s (T_{sat} - T_s) = m \bar{h}_{fg}^*$$

rearranging, and substituting in in Re ,

$$Re = \frac{A Q}{\rho_1 \bar{h}_{fg}^*} = \frac{A A_s h (T_{sat} - T_s)}{\rho_1 \bar{h}_{fg}^*}$$

Based on Re , different regimes can be identified.

$Re \leq 30$ smooth and wave-free liquid-vapor interface

$30 < Re < 1800$ wavy laminar condensate flow

$1800 \leq Re$ turbulent condensate flow

For vertical plates, with $Re \leq 30$

$$\delta(x) = \left[\frac{A M_1 k_1 (T_{sat} - T_s) x}{g \rho_1 (\ell_1 - \ell_+) h_{fg}} \right]^{1/4}$$

and using

$$q_x = h_x (T_{sat} - T_s) = \frac{k_1}{\delta} (T_{sat} - T_s)$$

$$h_x = k_1 / \delta$$

can be derived analytically
(see section 10.7 (Incropera)
or section 10.5 (Angel's))

$$h_x = \left[\frac{g \rho_s (l_1 - l_v) h_{fg} k_s^3}{A u_s (T_{sat} - T_s) x} \right]^{1/4}$$

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and

$$h = \frac{1}{L} \int_0^L h_x dx = \frac{4}{3} h_{x=L} = 0.943 \left[\frac{g \rho_s (l_1 - l_v) h_{fg} k_s^3}{u_s (T_{sat} - T_s) L} \right]^{1/4}$$

With h_{fg} replacing h_{fg} in the above expression to take into account the temperature profile, and the cooling of the liquid below T_{sat} .

~~A~~ Here h_{fg}, ρ_s are evaluated at T_{sat} , while other liquid properties are evaluated at film temperature $T_f = \frac{T_{sat} + T_s}{2}$

Correlation for $30 < Re < 1800$

$$h = \frac{Re k_s}{1.08 Re^{1/22} - 5.2} \left(\frac{g}{v_i^2} \right)^{1/3} \quad l_v \leq l_1$$

or

— Kutateladze, 1963

$$h_{wavy} = 0.8 Re^{0.11} h_{smooth}$$

and

$$Re_{wavy} = \left\{ A \cdot 81 + 3.70 L k_s (T_{sat} - T_s) \left(\frac{g}{v_i^2} \right)^{1/3} \right\}^{0.820}$$

Correlation for $1800 \leq Re$

$$h_{turbulent} = \frac{Re k_s}{8750 + 58 P_r^{0.5} (Re^{0.75} - 273)} \left(\frac{g}{v_i^2} \right)^{1/3}$$

with

$$Re_{turbulent} = \left\{ \frac{0.0690 L k_s P_r^{0.5} (T_{sat} - T_s) \left(\frac{g}{v_i^2} \right)^{1/3}}{\mu_l h_{fg}^*} - 151 P_r^{0.5} + 283 \right\}^{1/3}$$

For vertical tubes with tube diameter larger than the film thickness, flat vertical plate relations can be used.

For inclined plates $h_{\text{inclined}} = h_{\text{vertical}} (\cos \theta)^{1/4}$
For horizontal tubes upto $\theta \leq 60^\circ$

$$h_{\text{horizontal}} = 0.729 \left[\frac{g \rho_c (P_s - P_v) W g k_s^3}{\mu_s (T_{\text{sat}} - T_s) D} \right]^{1/4}$$

For spheres, 0.729 is replaced by 0.815 in the above equation

For horizontal tube banks vertically stacked,

$$h_{\text{horizontal, } N \text{ tubes}} = \frac{1}{N^{1/4}} h_{\text{horizontal, 1 tube}}$$

↑
above equation

Example! Condensation of steam on horizontal tubes

The condenser of a steam power plant operates at a pressure of 7.38 kPa. Steam at this pressure condenses on the outer surfaces of horizontal tubes through which cooling water circulates. The outer diameter of the pipes is 3 cm, and the outer surface of the tubes is maintained at 30°C.

Determine the rate of heat transfer to the cooling water, and the rate of condensation of steam per unit length of a tube.

Assume steady state with isothermal tube.

Water at saturation temperature of 40°C (at 7.38 kPa) (158)

$$h_{fg} = 2407 \times 10^3 \text{ J/kg} \quad \rho_v = 0.05 \text{ kg/m}^3$$

$$\text{Film temperature} = 35^\circ\text{C} = \frac{T_{\text{sat}} + T_s}{2}$$

Water (liquid) properties

$$\rho_l = 994 \text{ kg/m}^3 \quad C_p = 4178 \text{ J/kg K}$$

$$\mu_l = 0.720 \times 10^{-3} \text{ kg/m.s} \quad k_l = 0.623 \text{ W/m.K}$$

$$h_{fg}^* = h_{fg} + 0.68 C_p (T_{\text{sat}} - T_s) = 2435 \times 10^3 \text{ J/kg}$$

$$\rho_v = 0.05 \text{ kg/m}^3 \text{ and } \rho_l = 994 \text{ kg/m}^3 \Rightarrow \rho_v \ll \rho_l$$

Thus,

$$h_{\text{horizontal}} = 0.729 \left[\frac{g \rho_l (\rho_l - \rho_v) h_{fg}^* k_l^3}{\mu_l (T_{\text{sat}} - T_s) D} \right]^{1/4}$$
$$= 9294 \text{ W/m}^2 \cdot \text{K}$$

$$A_s = \pi D L = 0.09425 \text{ m}^2$$

$$Q = h A_s (T_{\text{sat}} - T_s) = 8760 \text{ W}$$

$$\dot{m} = \frac{Q}{h_{fg}^*} = 0.00360 \text{ kg/s}$$