Sustainable Energy Transformation Technologies, SH2706

Lecture No 12

Title:

Heat Convection in ETS

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Outline

- Basic definitions in heat convection theory
- Single phase convection heat transfer
  - Forced convection
  - Natural convection
- Boiling heat transfer
  - Pool boiling
  - Convective boiling
    - Onset of Nucleate Boiling (ONB)
  - Boiling crisis – Critical Heat Flux (CHF)
    - Dryout
    - Departure from Nucleate Boiling (DNB)
  - Post-CHF heat transfer

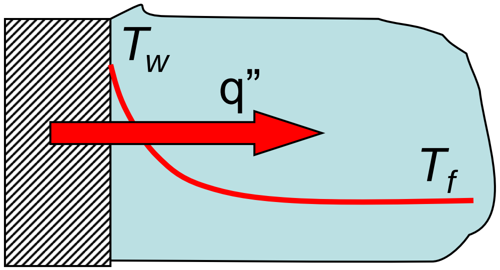
Heat Convection (1)

- Convection is one of the most common modes of heat transfer.
- The characteristic feature of the convective heat transfer is that heat is transported through large-scale (as compared to fluid molecules) fluid motion.
- Depending on the fluid flow character, the convective heat transfer is further classified in several categories:
  - **natural convection**, when fluid flow is driven by the buoyancy forces
  - **forced convection**, when fluid flow is driven by external forces, e.g. due to a pressure difference developed by a pump
  - **laminar convection**, when fluid flow is laminar
  - **turbulent convection**, when fluid flow is turbulent
  - **mixed convection**, when forced and natural convection co-exist

Heat Convection (2)

- Convective heat transfer from a solid surface to the fluid is often described by **Newton's equation of cooling**

$$q'' = h\left(T_w - T_f\right)$$



- where q'' [W m^{-2}] is the heat flux at the wall surface, h [W m^{-2} K^{-1}] is the convective heat transfer coefficient, T_w [K] is the wall surface temperature and T_f [K] is the fluid bulk temperature.
- The convective heat transfer coefficient depends upon the fluid properties and flow conditions.

Heat Convection (3)

- The value of the heat transfer coefficient can change on the heated surface.
- It is thus important to distinguish between the local value of the coefficient, valid on a small surface surrounding a given point, and the mean value, typically obtained from a proper averaging of the local values.
- Such averaging can be achieved as,

$$h = \frac{1}{A_H} \int_{A_H} h_{loc} dA_H$$

h_{loc} is the local heat transfer coefficient and A_H is the heat transfer area

Heat Convection (4)

- If the mean heat transfer coefficient between given solid and fluid is known, then the amount of heat exchanged through a surface with area A_H can be found as

$$q = hA_H \Delta T$$

ΔT – proper mean temperature difference between the solid surface and the fluid

Heat Convection (5)

- Often it can be assumed that the heat transfer coefficient is a function of distance only, as is the case for flow along a plate or flow in pipes.
- For such cases the mean heat transfer coefficient can be found as

$$h = \frac{1}{L}\int_0^L h_z dz$$

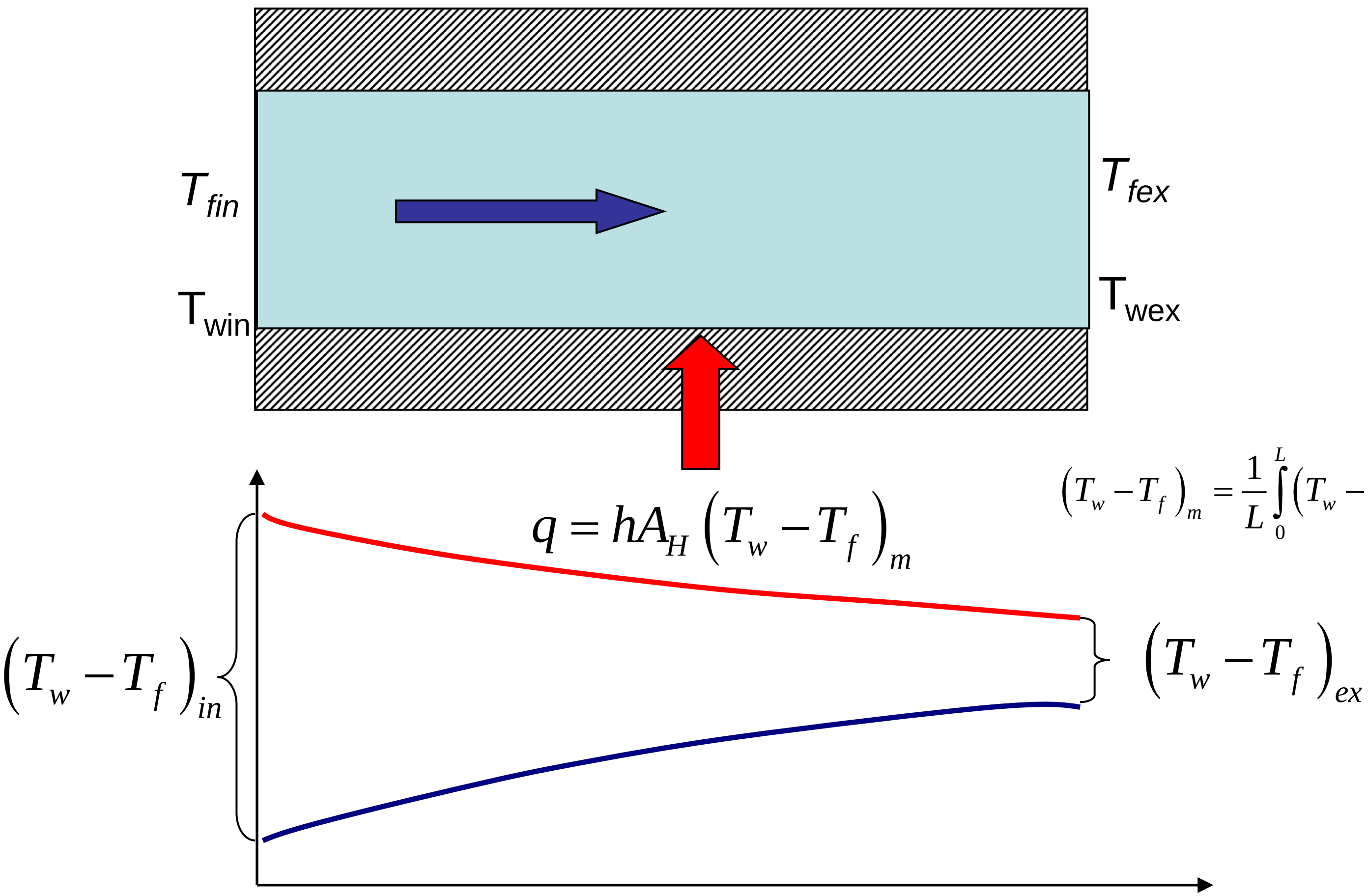
- and the exchanged heat is found as

$$q = hA_H\left(T_w - T_f\right)_m$$

Logarithmic mean temperature difference

$$\left(T_w - T_f\right)_m = \frac{1}{L}\int_0^L \left(T_w - T_f\right)dz = \frac{\left(T_w - T_f\right)_{in} - \left(T_w - T_f\right)_{ex}}{\ln\frac{\left(T_w - T_f\right)_{in}}{\left(T_w - T_f\right)_{ex}}}$$

Heat Convection (6)



$$\left(T_w - T_f\right)_m = \frac{1}{L}\int_0^L \left(T_w - T_f\right) dz = \frac{\left(T_w - T_f\right)_{in} - \left(T_w - T_f\right)_{ex}}{\ln \frac{\left(T_w - T_f\right)_{in}}{\left(T_w - T_f\right)_{ex}}}$$

Heat Convection (7)

- Newton's equation of cooling is applicable to all kinds of flows: laminar and turbulent; for both gases and liquids, and describes a heat transfer mechanism which is termed the **convection**.
- When the fluid motion is caused by pumps, fans, or in general by external pressure gradients, the **forced convection** takes place.
- Otherwise when fluid motion is caused be density gradients in the fluid, the **natural convection** occurs.

- The simplicity of Newton's equation could be deceiving, however.

- In fact, the heat transfer coefficient h is hiding the complexity of the solid-fluid heat transfer phenomena, and, as can be expected, a proper correlation must be used to obtain the coefficient which will be valid for specific flow conditions only.

Heat Convection (8)

- The dimensional analysis of involved governing equations leads further to a general relationship to determine the Nusselt number.

$$\mathrm{Nu} = \frac{h \cdot L}{\lambda}$$

L – characteristic dimension

λ – thermal conductivity of fluid

- In the case of the forced convection the relationship is as follows,

$$\mathrm{Nu} = f\left(\mathrm{Re}, \mathrm{Pr}, \ldots\right)$$

Re – Reynolds number
Pr – Prandtl number

- whereas for the natural (or free) convection the relationship becomes

$$\mathrm{Nu} = f\left(\mathrm{Gr}, \mathrm{Pr}, \ldots\right)$$ or $$\mathrm{Nu} = f\left(\mathrm{Ra}, \ldots\right)$$

Gr – Grashof number
Ra - Rayleigh

Heat Convection (9)

- Here we introduced additional important dimensionless numbers applicable to convective heat transfer:

Reynolds number $\mathrm{Re} = \dfrac{U \cdot L}{\nu}$

Grashof number $\mathrm{Gr} = \dfrac{\beta g L^3}{\nu^2} \Delta T$

Rayleigh number $\mathrm{Ra} = \dfrac{\beta g L^3}{a\nu} \Delta T$

Prandtl number $\mathrm{Pr} = \dfrac{\nu}{a} = \dfrac{c_p \mu}{\lambda}$

ν – kinematic viscosity, [$m^2\ s^{-1}$]

L – characteristic length [m]

β – volumetric expansion coeff. [K^{-1}]

$\Delta T = T_w - T_f$ - temperature difference

g – gravity acceleration

a – thermal diffusivity $=\lambda/(\rho c_p)$

μ – dynamic viscosity, [Pa.s]

c_p – specific heat [$J\ kg^{-1}\ K^{-1}$]

λ - thermal conductivity [$W\ m^{-1}\ K^{-1}$]

Laminar Forced Convection (1)

- In laminar flow the heat transfer in fluid is resulting from conduction only, and the temperature distribution can be obtained from the energy equation.
- Let's investigate laminar flow in a pipe.
- In the cylindrical system of coordinates and with an assumption of axisymmetric flow the energy conservation equation for liquid becomes,

$$\frac{w}{a}\frac{\partial T}{\partial z} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{\partial^2 T}{\partial z^2}$$

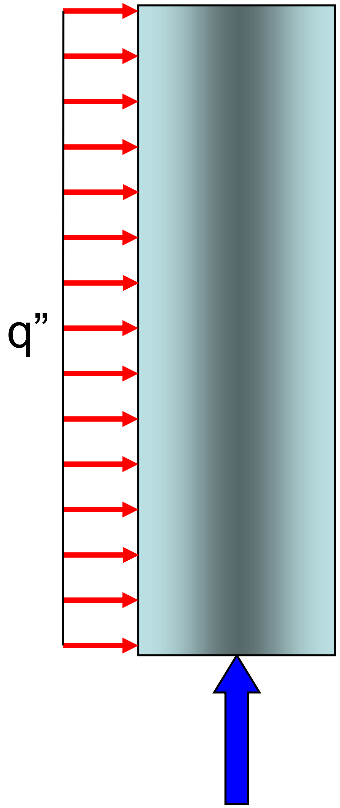
w(r) – local flow velocity
a – thermal diffusivity
T – temperature
r,z – radial & axial coordinate

Laminar Forced Convection (2)

- The term

$$\frac{\partial^2 T}{\partial z^2}$$

- describes heat conduction in the axial (flow) direction and is important only for liquid metals. For water (as it is assumed here), this term can be dropped.
- Assuming further constant heat flux on the pipe wall surface and a constant heat transfer coefficient, the axial temperature derivative will be



$$\frac{\partial T}{\partial z} = \frac{dT_w}{dz} = \frac{dT_m}{dz}$$

T_m – mean liquid temperature

Laminar Forced Convection (3)

- Thus, the equation to be solved is as follows

$$\frac{w}{a}\frac{dT_m}{dz} = \frac{1}{r}\frac{d}{dr}\left(r\frac{dT}{dr}\right)$$

- or, using the known parabolic solution for velocity distribution in a pipe:

$$w = 2U\left[1-\left(\frac{r}{R}\right)^2\right]$$

U – mean flow velocity
R – pipe inner radius

- we get:

$$\frac{2U}{a}\left[1-\left(\frac{r}{R}\right)^2\right]\frac{dT_m}{dz} = \frac{1}{r}\frac{d}{dr}\left(r\frac{dT}{dr}\right)$$

Laminar Forced Convection (4)

- Double-integration yields,

$$T = T_w - \frac{2UR^2}{a}\left[\frac{3}{16} - \left(\frac{r}{2R}\right)^2 + \left(\frac{r}{2R}\right)^4\right]\frac{dT_m}{dz}$$

- The mean temperature in the pipe cross-section can be found as:

$$T_m \equiv \frac{\int_A wTdA}{\int_A wdA} = \frac{2}{UR^2}\int_0^R wTrdr = T_w - \frac{11}{96}\frac{2UR^2}{a}\frac{dT_m}{dz}$$

Laminar Forced Convection (5)

- Now the heat flux can be expressed as

$$q'' = h\left(T_w - T_m\right) = h\,\frac{11}{96}\,\frac{2UR^2}{a}\,\frac{dT_m}{dz}$$

- The same heat flux can be obtained from the temperature gradient of the fluid in the vicinity of the pipe wall as,

$$q'' = \lambda\left(\frac{dT}{dr}\right)_{r=R} = \frac{\lambda UR}{2a}\,\frac{dT_m}{dz}$$

Laminar Forced Convection (6)

- Combining the two expressions for heat flux, yields:

$$h = \frac{48}{11}\frac{\lambda}{2R} \Rightarrow \mathrm{Nu} = \frac{h \cdot 2R}{\lambda} = \frac{48}{11} \approx 4.364$$

- It should be recalled that the above value of the Nusselt number has been obtained for a laminar convective heat transfer in a pipe with <u>uniform distribution of heat flux</u>.
- Similar analysis can be performed for a case with a <u>uniform distribution of the wall temperature</u>, and the resultant Nusselt number is as follows,

$$\mathrm{Nu} = 3.658$$

Laminar Forced Convection (7)

- As can be seen, for such simplified cases, Nusselt number has a constant value and does not depend on Reynolds and Prandtl numbers

- This is due to the assumptions that have been adopted in the derivation and solution of equations

- For more complex cases and to obtain better accuracy, several correlations for the Nusselt number have been developed

Laminar Forced Convection (8)

- One such correlation was given by Sieder and Tate:

$$\mathrm{Nu}_{\mathrm{ar}} = 1.86\left(\mathrm{Re}\,\mathrm{Pr}\,\frac{D_h}{L}\right)^{1/3}\left(\frac{\mu_f}{\mu_w}\right)^{0.14} \qquad \mathrm{Nu}_{ar} = h_{ar} D_h / \lambda$$

- where:

μ_f viscosity calculated at mean arithmetic between inlet and outlet temperature

μ_w viscosity calculated at wall temperature

L channel length

D_h channel hydraulic diameter

Nu_{ar} Nusselt number based on heat transfer coefficient defined for arithmetic mean temperature

Laminar Forced Convection (9)

- The product of the Reynolds and the Prandtl number often occurs in various expressions, and can be replaced by the **Peclet number** as follows

$$\mathrm{Pe} = \mathrm{Re}\,\mathrm{Pr} = \frac{UD_h}{\nu}\frac{\nu}{a} = \frac{\rho c_p U D_h}{\lambda}$$

Laminar Forced Convection (10)

- Since in laminar flows the buoyancy effects can play a significant role, in some correlations this effect is accounted for by introduction of the **Grashof number**.
- Michiejev proposed the following expression, valid for various channels with $L/D_h > 50$

$$\mathrm{Nu} = 0.15\,\mathrm{Re}^{0.33}\,\mathrm{Pr}_f^{0.43}\,\mathrm{Gr}^{0.1}\left(\frac{\mathrm{Pr}_f}{\mathrm{Pr}_w}\right)^{0.25} \qquad Gr = \frac{g\beta D_h^3 \Delta T}{\nu^2} \qquad \Delta T = T_w - T_f$$

- The reference temperature, used to determine the fluid properties, is the arithmetic mean of the channel inlet and outlet temperature, and the reference linear dimension is the hydraulic diameter of the channel, D_h. Pr_f and Pr_w are Prandtl numbers calculated at mean fluid temperature and wall temperature, respectively

Turbulent Forced Convection (1)

- In essentially all cases of practical importance for heat transfer analysis in energy transformation systems, the working fluid is under turbulent flow conditions.
- Satisfactory predictions of heat transfer coefficients in long, straight channels of uniform cross section can be made on the assumption that the only variables involved are
  - mean velocity of the fluid,
  - diameter (or equivalent diameter) of the channel,
  - fluid density,
  - fluid heat capacity,
  - fluid viscosity,
  - thermal conductivity of the fluid.

Turbulent Forced Convection (2)

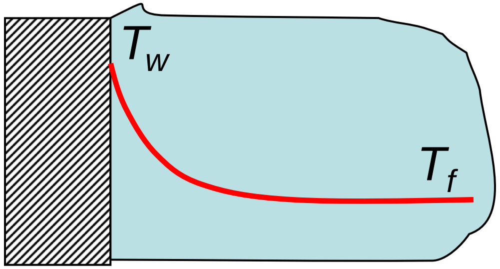
- Using the dimensional analysis of the governing equations it can be shown that the heat transfer processes for turbulent flow conditions can be expressed in terms of an expression that mainly contains three dimensionless numbers:
  - the Reynolds number (Re),
  - the Nusselt number (Nu)
  - the Prandtl number (Pr),
- and the general form of the expression is as follows

$$\mathrm{Nu} = f\left(\mathrm{Re}, \mathrm{Pr}, \ldots\right)$$

Turbulent Forced Convection (3)

- The temperature and the velocity profile in turbulent flows are more flat as compared with laminar flows.
- This fact influences the choice of the reference temperature that is used to calculate the fluid properties.
- Often as the reference temperature it is taken the arithmetic mean between the fluid and the wall temperature

$$T_{ref} = \frac{T_f + T_w}{2}$$



- Occasionally

$$T_{ref} = T_f + 0.4\left(T_w - T_f\right) \quad \text{or} \quad T_{ref} = T_f + 0.6\left(T_w - T_f\right)$$

Turbulent Forced Convection (4)

- **Dittus-Boelter** proposed the following correlation for forced-convective heat transfer in long straight pipes:

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{n}$$

  – valid for $L/D_h > 60$, $\mathrm{Re} > 10000$ and $0.7 < \mathrm{Pr} < 100$.
  – The exponent n is equal to 0.4 for heating and 0.3 for cooling of the fluid
  – The fluid properties should be calculated with the average fluid temperature

Turbulent Forced Convection (5)

- Colburn introduced the Stanton number to replace the Nusselt number as follows

$$\mathrm{St} = \frac{\mathrm{Nu}}{\mathrm{Re}\,\mathrm{Pr}} = \frac{hD_h\mu\lambda}{\lambda\rho U D_h \mu c_p} = \frac{h}{\rho U c_p}$$

- and the correlation is given as

$$\mathrm{St} = 0.023\,\mathrm{Re}^{-0.2}\,\mathrm{Pr}^{-2/3}$$

- Or using the Nusselt number:

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}$$

Note: it is very similar to the Dittus-Boelter correlation, but properties are calculated differently!

Turbulent Forced Convection (6)

- The Colburn correlation is valid for L/D_h > 60, Re > 10000 and 0.7 < Pr < 160.
- The reference temperature is defined as the mean arithmetic of the wall and the average fluid temperature.
- Specific heat c_p should be calculated at the average fluid temperature, but all other properties should be calculated at the reference temperature.

Natural Convection (1)

- Natural (or free) convection occurs when fluid flow is driven by buoyancy forces, usually due to gravity.
- In the vicinity of the body that exchanges heat with fluid there are temperature gradients that cause density changes in the fluid.
- The density changes are causing the buoyancy force, which result in local fluid motion.
- To capture this effect, the buoyancy force must be introduced into the momentum equation for fluid.
- This force per unit volume is equal to

$$g\left(\rho_\infty - \rho\right)$$

ρ_∞ - fluid density far from wall

Natural Convection (2)

- The expression for the buoyancy force may be transformed through introduction of the coefficient of the thermal expansion for fluid

$$\beta = \frac{1}{\upsilon_\infty}\left(\frac{\partial \upsilon}{\partial T}\right)_p \qquad \upsilon = 1/\rho \text{ – specific volume}$$

- Since

$$\upsilon \cong \upsilon_\infty + \left(\frac{\partial \upsilon}{\partial T}\right)_p (T - T_\infty) = \upsilon_\infty \left[1 + \beta (T - T_\infty)\right]$$

- Then

$$g(\rho_\infty - \rho) = g\rho\left(\frac{\rho_\infty}{\rho} - 1\right) \cong g\rho\beta(T - T_\infty) = g\rho\beta\Delta T$$

Natural Convection (3)

- The mathematical description of the natural convection is based on formulation of the conservation equations for mass, momentum and energy.
- Introducing a few practical approximations,
  - constant density in the continuity equation,
  - neglect of the pressure gradient in the momentum equation,
  - neglect of energy dissipation function,
  - constant viscosity,
- the conservation equations are as follows

$$\nabla \cdot \mathbf{v} = 0$$

$$\nabla \cdot (\mathbf{v}\mathbf{v}) = \nu \nabla^2 \mathbf{v} + \mathbf{g}\beta\Delta T$$

$$\mathbf{v}\nabla T = a\nabla^2 T$$

Natural Convection (4)

- Performing a dimensional analysis of equations, it can be shown that the general expression describing the free convection heat transfer is as follows

$$\mathrm{Nu} = f\left(\mathrm{Gr}, \mathrm{Pr}, \ldots\right) \quad \text{or} \quad \mathrm{Nu} = f\left(\mathrm{Ra}, \ldots\right)$$

  – where

$$\mathrm{Nu} = \frac{h \cdot D_h}{\lambda}$$ Nusselt number

$$\mathrm{Gr} = \frac{\beta g L^3}{\nu^2} \Delta T$$ Grashof number

$$\mathrm{Pr} = \frac{\nu}{a} = \frac{c_p \mu}{\lambda}$$ Prandtl number

$$\mathrm{Ra} = \frac{\beta g L^3}{\nu a} \Delta T = \mathrm{Gr} \cdot \mathrm{Pr}$$

Rayleigh number

Natural Convection (5)

- Various empirical expressions are provided for prediction of the Nusselt number, derived specifically for laminar or turbulent flows in different geometry configurations.

- For laminar natural convection about cylinder or sphere:

$$\mathrm{Nu} = \begin{cases} const & \text{for} \quad \mathrm{Gr}\,\mathrm{Pr} < 10^{-3} \\ 1.18\left(\mathrm{Gr}\,\mathrm{Pr}\right)^{1/8} & \text{for} \quad 10^{-3} < \mathrm{Gr}\,\mathrm{Pr} < 5\cdot 10^{2} \\ 0.54\left(\mathrm{Gr}\,\mathrm{Pr}\right)^{1/4} & \text{for} \quad 5\cdot 10^{2} < \mathrm{Gr}\,\mathrm{Pr} < 2\cdot 10^{7} \end{cases}$$

const = 0.5 for an infinite cylinder;

= 2 for a sphere

Natural Convection (6)

- For turbulent natural convection (Michiejev):

$$\mathrm{Nu} = 0.135(\mathrm{Gr}\,\mathrm{Pr})^{1/3}$$

  - valid for

$$2 \cdot 10^{7} < \mathrm{Gr}\,\mathrm{Pr} < 10^{13}$$

  - with reference temperature

$$T_{ref} = \frac{T_w + T_\infty}{2}$$

Boiling Heat Transfer (1)

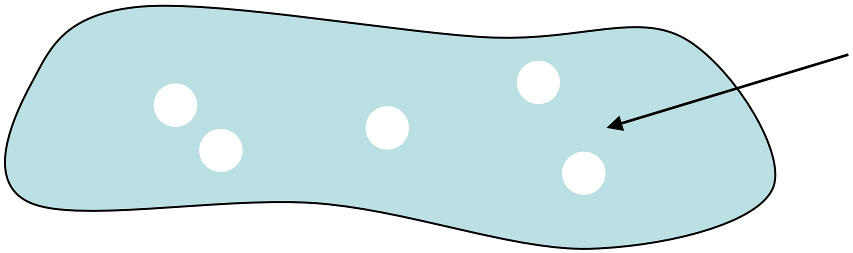
- Boiling heat transfer occurs when the heated fluid undergoes phase change from liquid to vapor.
- This is a very complex process due to phenomena that occurs both at the solid-fluid and liquid-vapor interfaces.

- One of the characteristic features of the boiling heat transfer is its high heat transfer coefficient.

- This fact makes the boiling heat transfer an interesting topic since it enables transfers of large heat fluxes, which are required in many practical applications.

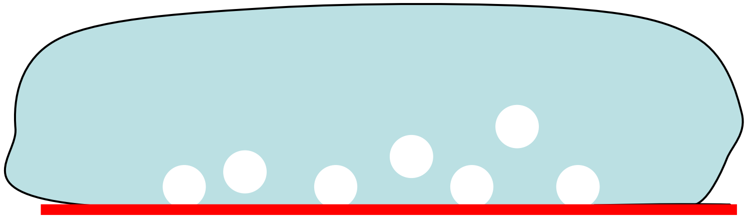
Boiling Heat Transfer (2)

- Depending on the applied heating, boiling can be:
  - **homogeneous**, when heat is supplied in the whole liquid volume, and vapor is generated in any point in the volume



boiling takes place in the whole volume

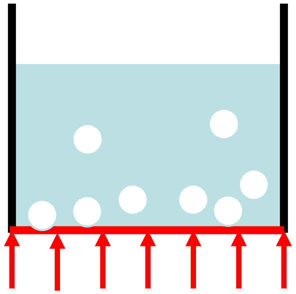
  - **heterogeneous**, when heat is supplied through a solid wall and vapor is generated on the solid surface.



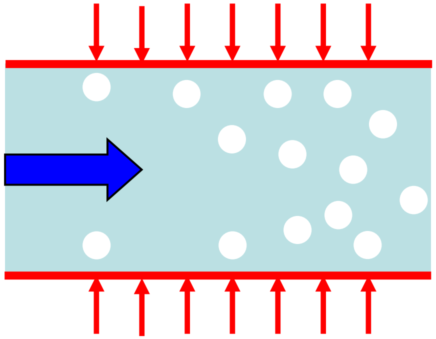
boiling takes place at the heated surface

Boiling Heat Transfer (3)

- Another classification reflects the fluid behavior during boiling. Based on that boiling can be:
  - **pool boiling**, when fluid is at rest. This case corresponds to the free-convection heat transfer considered for single-phase flows.

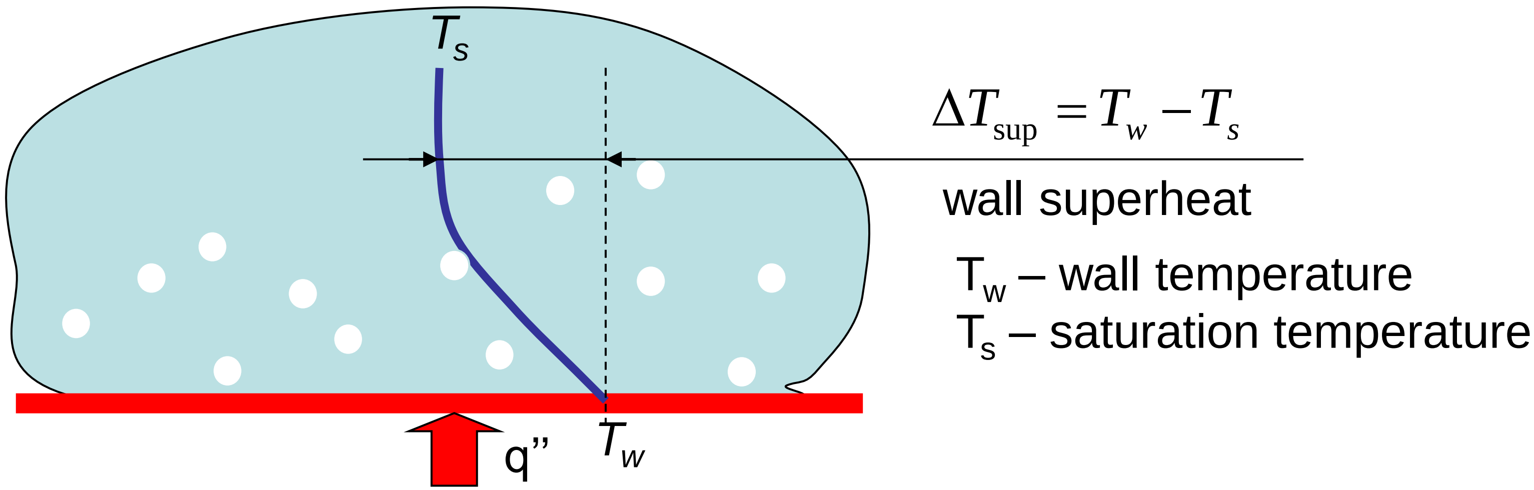


  - **flow (convective) boiling in heated channels**, when fluid is in motion in channels due to external agitation, created with fans or pumps. This type of boiling corresponds to the forced convection heat transfer considered for single-phase flows.



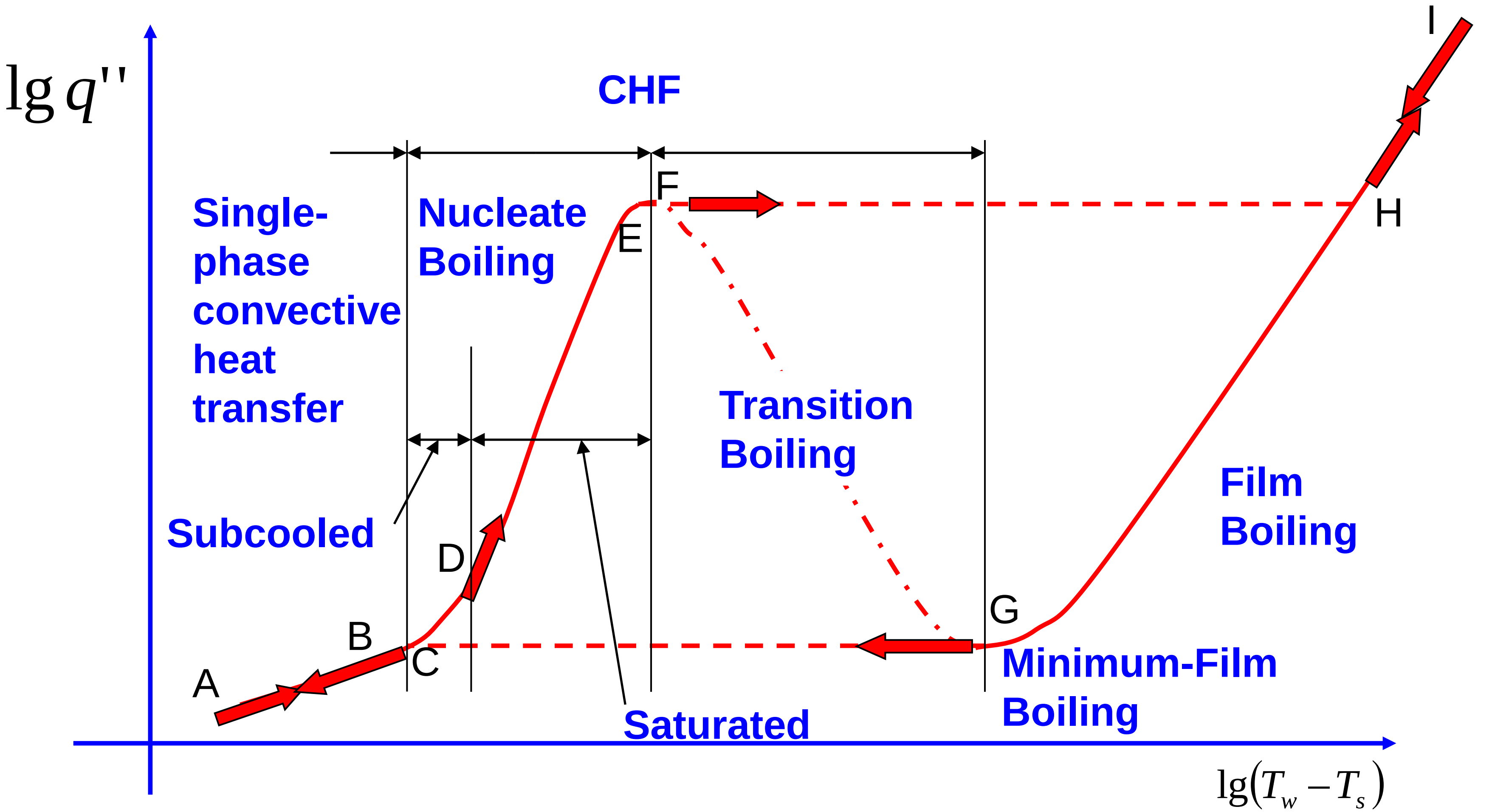
Boiling Heat Transfer (4)

- Boiling heat transfer is often presented graphically on a plane, where the heat flux is shown in function of the wall superheat.



- An example of the boiling curve with pertinent heat transfer modes are shown in the next slide.

Boiling Curve (1)



Boiling Curve (2)

- for heat-flux controlled boiling with increasing heat flux the curve *ABCDEFHI* will be followed

- With the wall-temperature controlled boiling with increasing temperature the curve *ABCDEFGHI* will be followed.

- In the reversed process, when the heat flux is decreased from point *I*, the curve *IHGCBA* will be followed, and when the temperature is decreased from point *I* the curve *IHGFEDCBA* will be followed.

Nucleate Pool Boiling (1)

- From experimental data it is evident that the heat transfer coefficient for pool boiling is proportional to the heat flux as,

$$h = C\left(q''\right)^{n}$$

  - where *C* is a constant which depends on pressure and type of fluid. The exponent *n* has a value 0.67÷0.7, depending to a certain extend on the surface roughness.

- Forster and Zuber proposed a semi-empirical expression for heat transfer coefficient valid for water,

$$h = 0.56\left(q''\right)^{0.7} p^{0.15}$$

h is heat transfer coefficient [W·m^{-2}·K^{-1}], *q"* is heat flux [W·m^{-2}] and *p* is pressure [Pa].

Nucleate Pool Boiling (2)

- Most of the theoretical expressions derived for the heat transfer coefficient in nucleate pool boiling use the following relationship as the starting point:

$$\mathrm{Nu}_b = a\,\mathrm{Re}_b^{m_1}\,\mathrm{Pr}_f^{m_2}$$

  - here a is a constant coefficient, m_1 and m_2 are constant exponents
  - Re_b – Reynolds number for pool boiling
  - Nu_b – Nusselt number for pool boiling
  - Pr_f – Prandtl number for fluid

Nucleate Pool Boiling (3)

- As an example, Rohsenow proposed the following:

$$\mathrm{Re}_b = \frac{q''}{i_{fg}\mu_f}\left[\frac{\sigma}{g\left(\rho_f - \rho_g\right)}\right]^{0.5}$$

$$\mathrm{Nu}_b = \frac{q''}{\Delta T_{\mathrm{sup}}\lambda_f}\left[\frac{\sigma}{g\left(\rho_f - \rho_g\right)}\right]^{0.5}$$

  – and constants, valid for boiling water on mechanically polished stainless steel are: a = 75.8, m_1 = 2/3 and m_2 = -0.7.

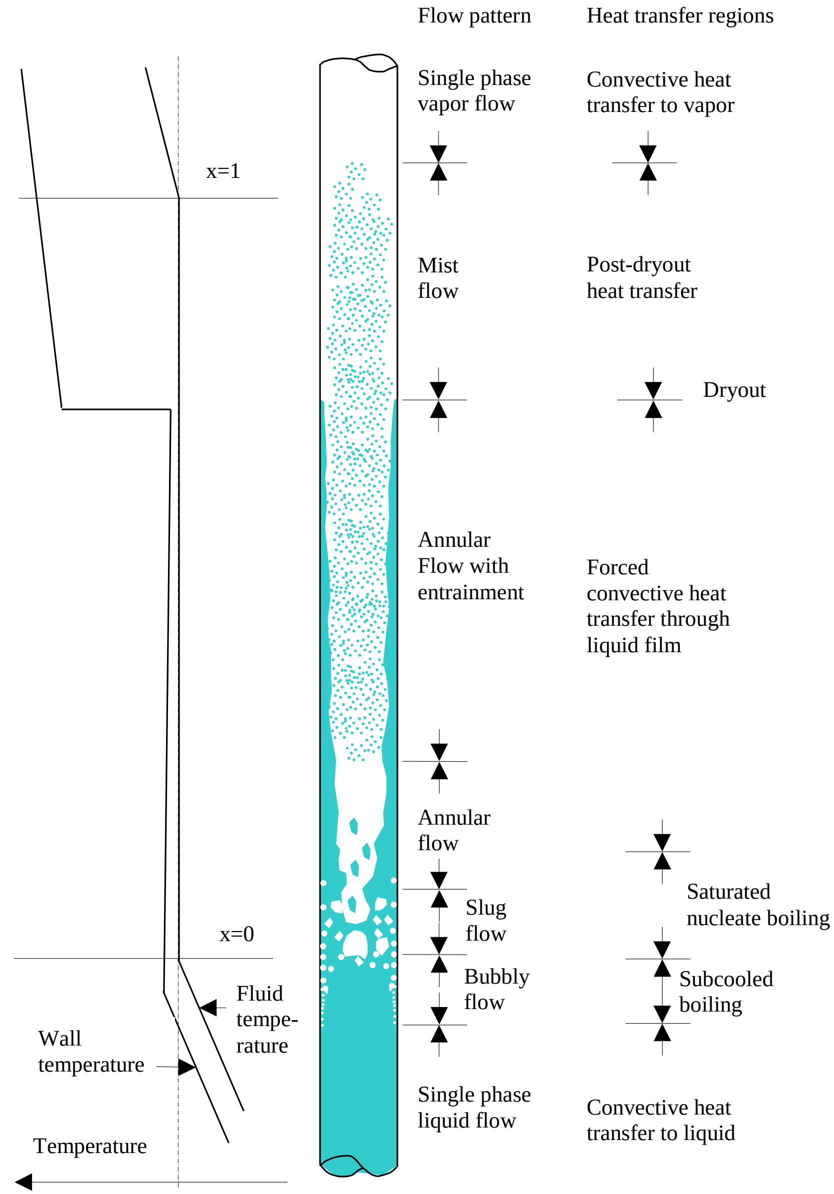
Convective Boiling in Heated Channels

- Convective boiling in heated channels occurs in various types of boilers, including the reactor core of light water reactors or collectors of solar thermal power.

- This is a very complex type of boiling, since flow conditions change along the heated channel due to the increase of enthalpy and the resulting phase change.

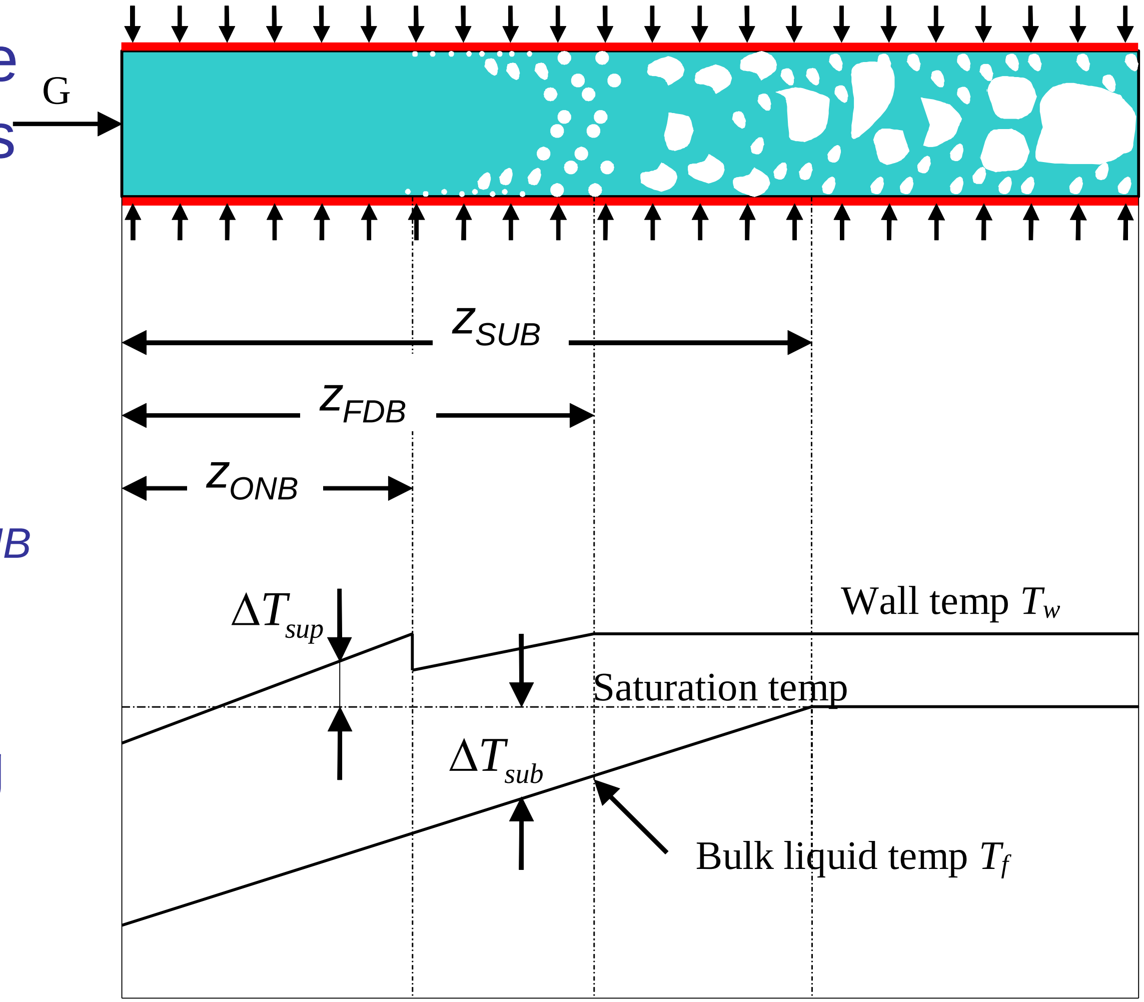
- The boiling regimes may change from the nucleate subcooled and saturated boiling to evaporating film, and possibly to the film and mist-flow boiling.

Flow and Heat Transfer Regimes in a Boiling Channel



Onset of Nucleate Boiling (1)

- Onset of Nucleate Boiling (ONB) – is a point where boiling first appears in the channel. It is located at $z = z_{ONB}$ from the inlet
- z_{FDB} – fully developed boiling
- z_{SUB} – subcooled length



Onset of Nucleate Boiling (2)

- The coolant temperature in a uniformly heated channel is a linear function of the axial distance

$$T_f(z) = T_{fin} + \frac{q'' P_H z}{c_p G A}$$

T_{fin} – inlet fluid temperature

- From which, the length of the subcooled region can be readily obtained as

$$z_{SUB} = \frac{c_p G A}{q'' P_H}\left(T_{sat} - T_{fin}\right) = \frac{c_p G A}{q'' P_H}\Delta T_{subi}$$

ΔT_{subi} Inlet subcooling

T_{sat} – saturation temperature

Onset of Nucleate Boiling (3)

- The Newton equation of cooling is as follows

$$T_w - T_f = q''/h$$

T_w wall surface temperature

- Thus, the wall surface temperature becomes

$$T_w(z) = T_f(z) + \frac{q''}{h} = T_{fin} + q''\left(\frac{P_H z}{c_p G A} + \frac{1}{h}\right)$$

- Or, introducing so-called **wall superheat** $\Delta T_{sup}(z)$, it can be expressed as a function of *z*-coordinate as follows:

$$\Delta T_{\sup}(z) \equiv T_w(z) - T_{sat} = -\Delta T_{subi} + q''\left(\frac{P_H z}{c_p G A} + \frac{1}{h}\right)$$

Onset of Nucleate Boiling (4)

- Clearly, there is no boiling when the wall superheat is less than zero
- Bowring suggested that at the onset-of-nucleate-boiling point the wall superheat is equal to that which result from a subcooled boiling correlation
- Experiments indicate that in subcooled boiling the wall superheat and the applied heat flux are related as

$$\Delta T_{\text{sup}} = \psi \cdot \left(q''\right)^n$$

n and ψ - parameters

Onset of Nucleate Boiling (5)

- Thus Bowring's expression for the local superheat for onset of nucleate boiling is

$$\Delta T_{\sup}(z)\Big|_{ONB} \equiv T_w(z_{ONB}) - T_{sat} = -\Delta T_{subi} + q''\left(\frac{P_H z_{ONB}}{c_p GA} + \frac{1}{h}\right) = \psi \cdot \left(q''\right)^n$$

- From which, the coordinate of onset of nucleate boiling z_{ONB} is found as

$$\boxed{z_{ONB} = \left[\frac{\Delta T_{subi} + \psi \cdot \left(q''\right)^n - \dfrac{q''}{h}}{q'' P_H}\right] c_p GA}$$

Onset of Nucleate Boiling (6)

- Examples of subcooled boiling correlations:

  – Jens-Lottes

$$\Delta T_{\sup} = 25\left(\frac{q''}{10^6}\right)^{0.25} e^{-p/62}$$

p – pressure, bar

q'' – heat flux, W/m^2

  – Thom et al.

$$\Delta T_{\sup} = 22.65\left(\frac{q''}{10^6}\right)^{0.5} e^{-p/87}$$

p – pressure, bar

q'' – heat flux, W/m^2

Saturated Flow Boiling (1)

- In subcooled boiling region the wall superheat is found as a function of heat flux using one of the earlier-mentioned correlations (Jens&Lottes, Thom *et al.*, etc)

- In saturated flow boiling other correlations, specifically developed for this heat transfer regime, should be used

- One of such correlations has been developed by Chen, who proposed to partition the heat transfer coefficient into the microscopic (boiling) and macroscopic (flow) contributions

Saturated Flow Boiling (2)

- That is, in the Chen correlation

$$h = h_{mic} + h_{mac}$$

- Where

$$h_{mac} = 0.023\left(\frac{\lambda_f}{D}\right)\mathrm{Re}^{0.8}\,\mathrm{Pr}_f^{0.4}\cdot F \qquad \mathrm{Re} = \frac{G(1-x)D}{\mu_f} \qquad \mathrm{Pr}_f = \frac{\mu_f c_{pf}}{\lambda_f}$$

x – local quality

$$S = \left(1 + 2.56\cdot 10^{-6} F^{1.463}\cdot \mathrm{Re}^{1.17}\right)^{-1}$$

$$F = \begin{cases} 1 & X_{tt}^{-1} \le 0.1 \\ 2.35\left(0.213 + \dfrac{1}{X_{tt}}\right)^{0.736} & X_{tt}^{-1} \ge 0.1 \end{cases} \qquad X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_f}\right)^{0.5}\left(\frac{\mu_f}{\mu_g}\right)^{0.1}$$

$$h_{mic} = 0.00122\left[\frac{\lambda_f^{0.79} c_{pf}^{0.45} \rho_f^{0.49}}{\sigma^{0.5}\mu_f^{0.29} i_{fg}^{0.24} \rho_g^{0.24}}\right]\Delta T_{\sup}^{0.24}\left(p_s(T_w) - p_f\right)^{0.75}\cdot S$$

$p_s(T_w)$ is the saturation pressure at wall temperature

Saturated Flow Boiling (3)

given q’’, x, p, G, D; guess ΔT_{sup}

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_f}\right)^{0.5} \left(\frac{\mu_f}{\mu_g}\right)^{0.1}$$

$$\mathrm{Re} = \frac{G(1-x)D}{\mu_f}$$

$$F = \begin{cases} 1 & X_{tt}^{-1} \le 0.1 \\ 2.35\left(0.213 + \frac{1}{X_{tt}}\right)^{0.736} & X_{tt}^{-1} \ge 0.1 \end{cases}$$

$$S = \left(1 + 2.56 \cdot 10^{-6} F^{1.463} \cdot \mathrm{Re}^{1.17}\right)^{-1}$$

$$h_{mac} = 0.023 \left(\frac{\lambda_f}{D}\right) \mathrm{Re}^{0.8}\, \mathrm{Pr}_f^{0.4} \cdot F$$

$$h_{mic} = 0.00122 \left[\frac{\lambda_f^{0.79} c_{pf}^{0.45} \rho_f^{0.49}}{\sigma^{0.5} \mu_f^{0.29} i_{fg}^{0.24} \rho_g^{0.24}}\right] \Delta T_{\sup}^{0.24} \left(p_s(T_w) - p_f\right)^{0.75} \cdot S$$

$$h = h_{mic} + h_{mac}$$

$$\Delta T_{\sup}^{new} = q''/h$$

$$\left\| \Delta T_{\sup}^{new} - \Delta T_{\sup} \right\| < \varepsilon$$

No

$$\Delta T_{\sup} = \Delta T_{\sup}^{new}$$

Yes

Critical Heat Flux (1)

- The value of pool boiling CHF may be evaluated from a proper correlation valid for given conditions.
- Zuber derived an expression for CHF by analyzing the stability of a flux of vapor bubbles generated at the heated surface.
- The correlation is as follows,

$$q''_{cr} = \frac{\pi}{24} i_{fg} \rho_g \left[\frac{\sigma g \left(\rho_f - \rho_g \right)}{\rho_g^2} \right]^{1/4} \left[\frac{\rho_f}{\rho_f + \rho_g} \right]^{1/2}$$

$$i_{fg} = i_g - i_f$$

σ – surface tension

Occurrence of Critical Heat Flux (1)

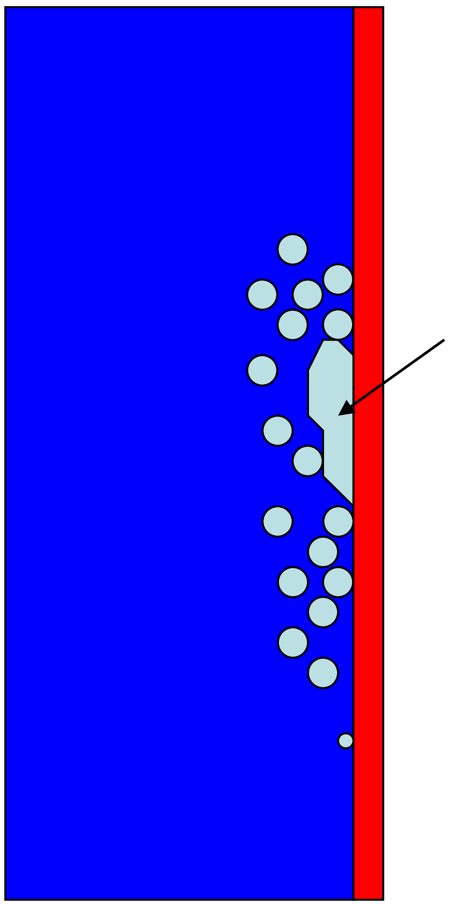
- Heat flux at which the wall temperature rises due to dramatically deteriorated heat transfer conditions is termed as the **Critical Heat Flux** (CHF)

- There are two major mechanisms of CHF in boiling channels:
  - Departure from Nucleate Boiling (DNB)
  - Dryout

- The physics of CHF is still not fully understood and mainly correlations are used for predictions of CHF occurrence

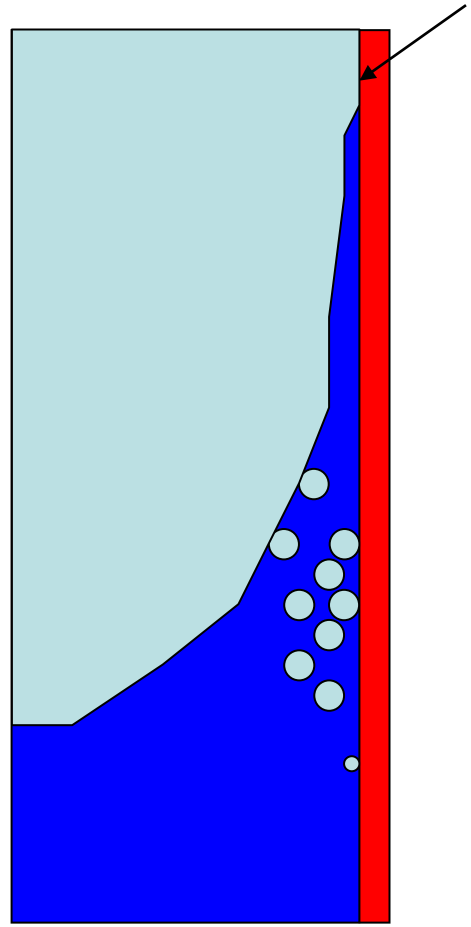
Occurrence of Critical Heat Flux (2)

Two main types of CHF: DNB and Dryout



Drypatch, where wall temperature increases significantly, due to poor heat transfer to liquid

This type of CHF is called “Departure from Nucleate Boiling”

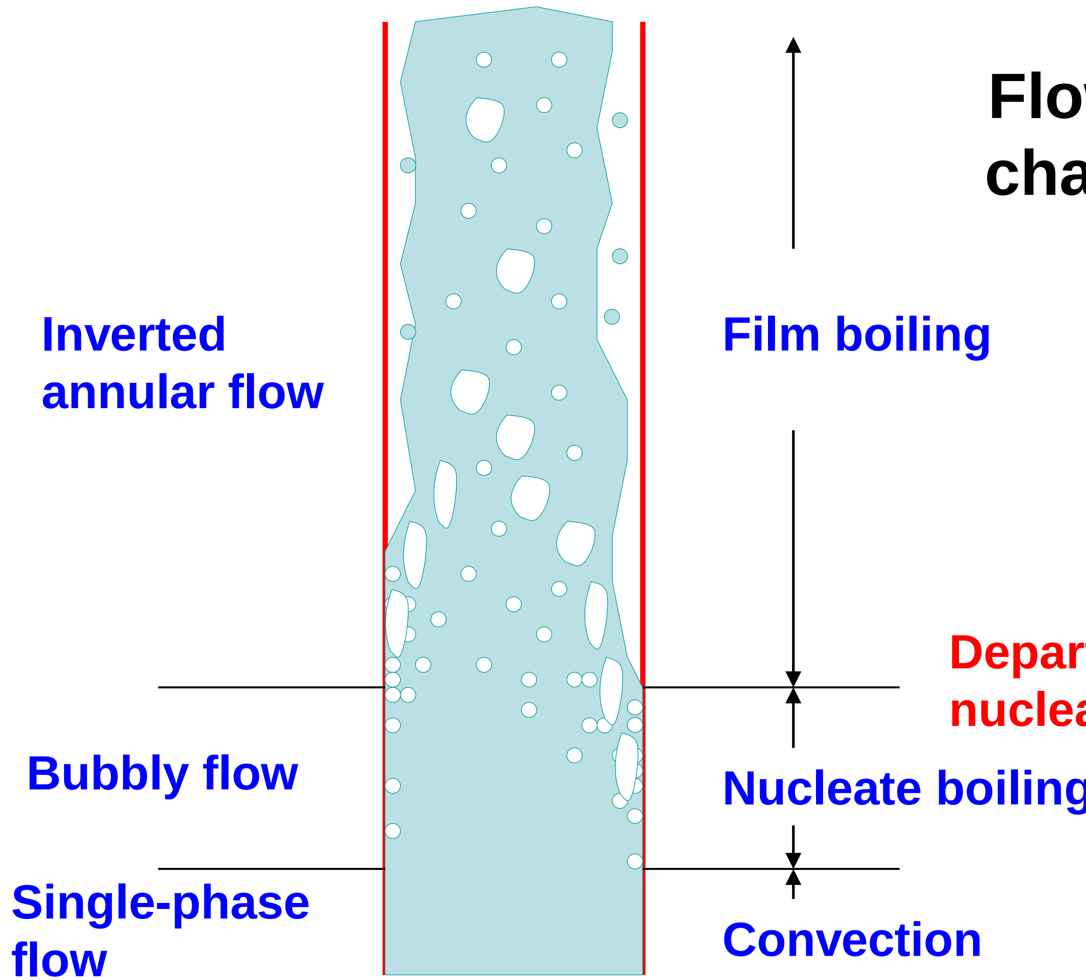


Drypatch, where liquid film dries out on the heated wall, leading to poor heat transfer.

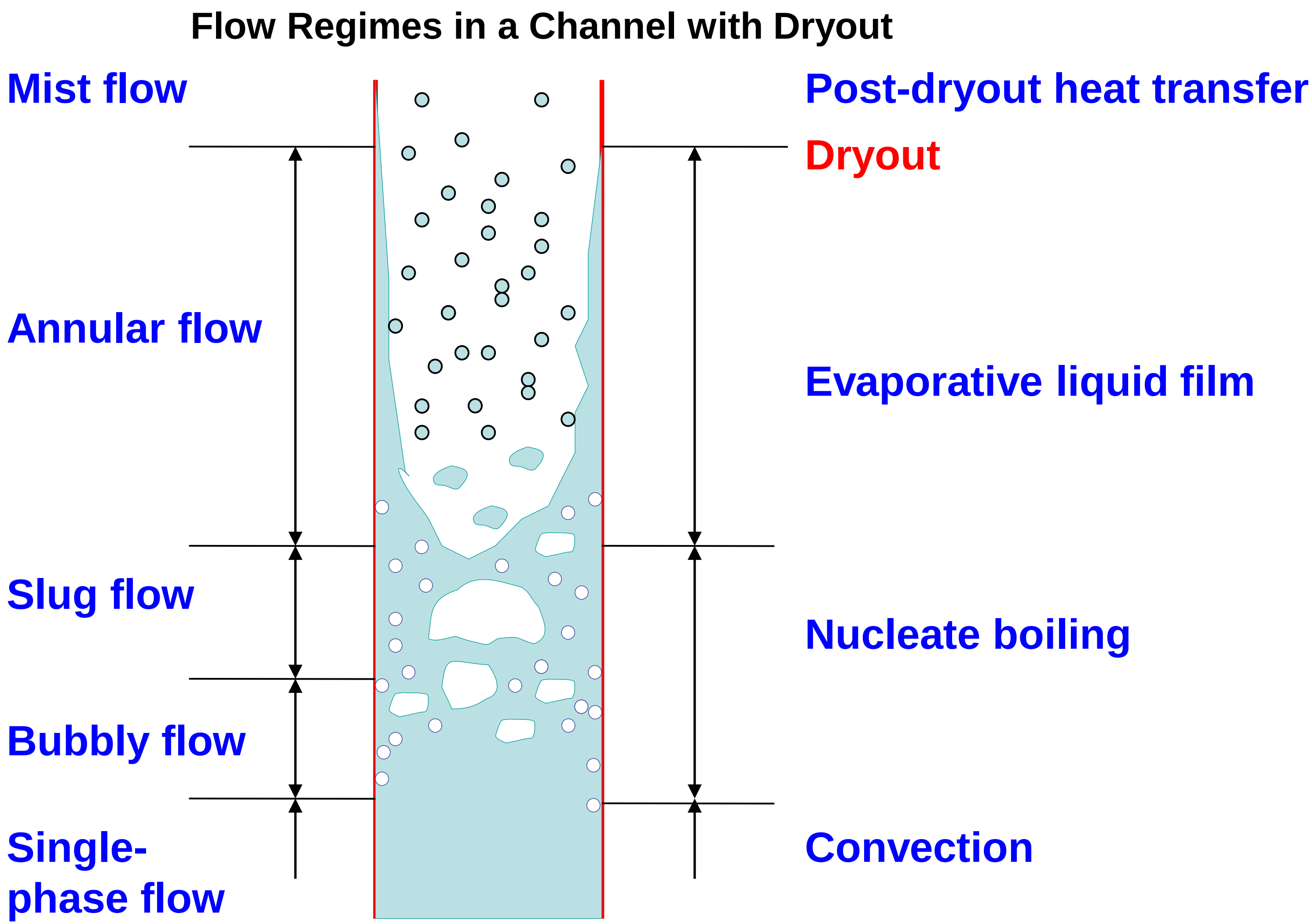
This type of CHF is called “Dryout”

Occurrence of Critical Heat Flux (3)

Flow regimes in a channel with DNB



Occurrence of Critical Heat Flux (4)



Occurrence of Critical Heat Flux (5)

- Examples of correlations:
  - Levitan and Lantsman correlation for DNB in 8mm pipes

$$q''_{cr} = \left[10.3 - 7.8\frac{p}{98} + 1.6\left(\frac{p}{98}\right)^2\right]\left(\frac{G}{1000}\right)^{1.2\left\{\left[0.25(p-98)/98\right]-x\right\}} e^{-1.5x}$$

750<G<5000
29.4<p<196

  - Levitan and Lantsman correlation for dryout in 8 mm pipes

$$x_{cr} = \left[0.39 + 1.57\frac{p}{98} - 2.04\left(\frac{p}{98}\right)^2 + 0.68\left(\frac{p}{98}\right)^3\right]\left(\frac{G}{1000}\right)^{-0.5}$$

750<G<3000
9.8<p<166.6

q_{cr} [MW/m^2], p [bar], G [kg/m^2.s], x – equilibrium quality, x_{cr} – critical quality

Occurrence of Critical Heat Flux (6)

- Levitan and Lantsman correlations can be used for other pipe diameters using the following corrections:
  - For DNB

$$q''_{cr} = q''_{cr}\Big|_{8mm} \cdot \left(\frac{8}{D}\right)^{0.5}$$

  - For dryout

$$x_{cr} = x_{cr}\Big|_{8mm} \cdot \left(\frac{8}{D}\right)^{0.15}$$

D – diameter, mm

Post-CHF Heat Transfer (1)

- Post-CHF heat transfer is heat transfer regimes that exists following the occurrence of CHF
  - DNB is typically followed by **convective film boiling** in the so-called inverted annular patter
  - Dryout is followed by heat transfer regime known as **mist flow evaporation**

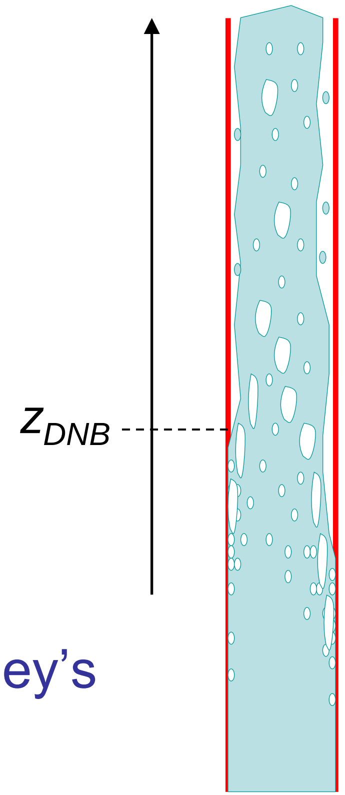
Post-CHF Heat Transfer (2)

- Convective film boiling (following DNB):
  - Laminar flow heat transfer coefficient

$$h = \left[\frac{\rho_g \left(\rho_f - \rho_g \right) g i_{fg} \lambda_g^3}{4\left(z - z_{DNB} \right) \mu_g \left(T_w - T_{sat} \right)} \right]^{1/4}$$

T_w – wall temperature

z_{DNB} – axial location of DNB



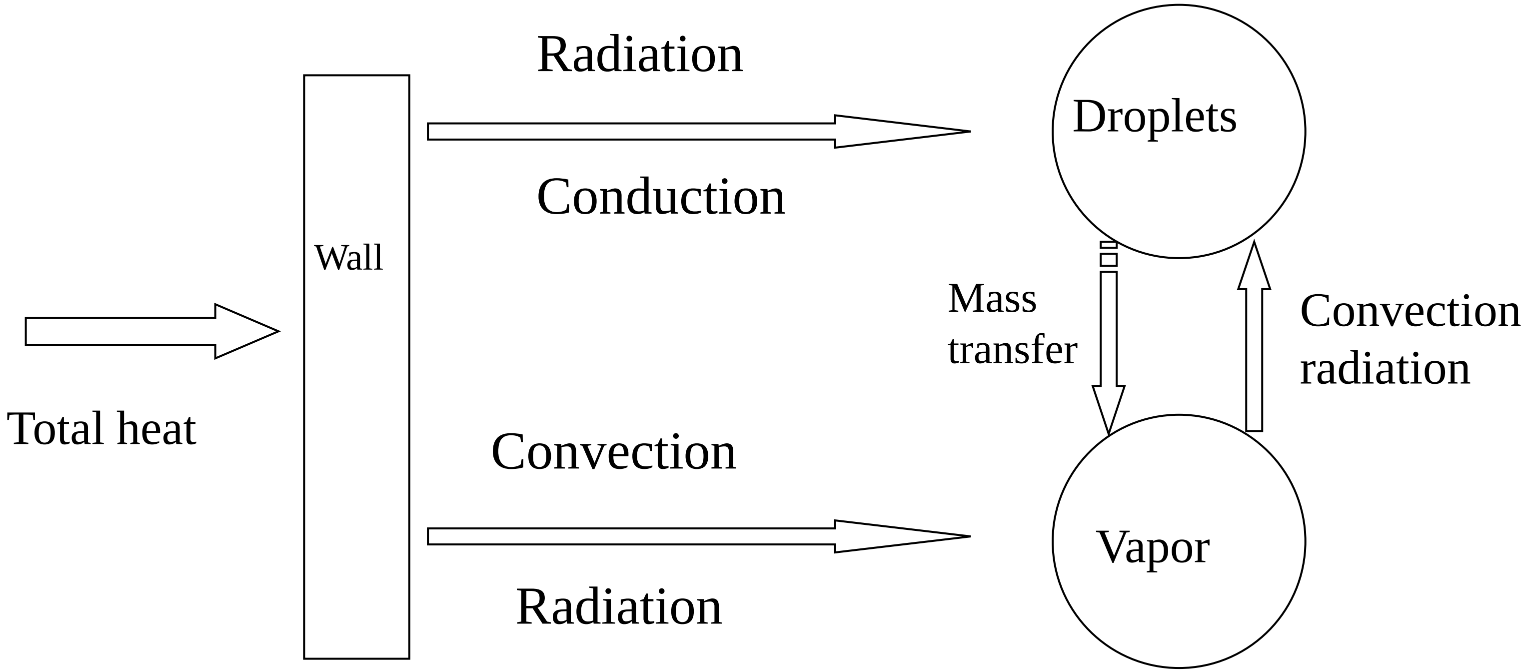
  - Turbulent flow heat transfer coefficient (using Bromley's approach)

$$h = 0.62 \left[\frac{\lambda_g^3 \rho_g \left(\rho_f - \rho_g \right) g i_{fg}}{\mu_g \left(T_w - T_{sat} \right) \lambda_H} \right]^{1/4} \qquad \lambda_H = 16.24 \left[\frac{\sigma^4 i_{fg}^3 \mu_g^5}{\rho_g \left(\rho_f - \rho_g \right)^5 g^5 \lambda_g^3 \left(T_w - T_{sat} \right)^2} \right]^{1/2}$$

Post-CHF Heat Transfer (3)

- In mist flow evaporation (following dryout) several different heat transfer mechanisms may play role:
  - Convective heat transfer from the wall to the vapor
  - Convective heat transfer from the vapor to the entrained droplets
  - Evaporation of droplets that collide with the wall and wet its surface
  - Evaporation of droplets that come to close proximity to the wall but do not wet the surface
  - Radiation heat transfer from the wall to the droplets
  - Radiation heat transfer from the wall to the vapor
  - Radiation heat transfer from vapor to droplets

Post-CHF Heat Transfer (4)



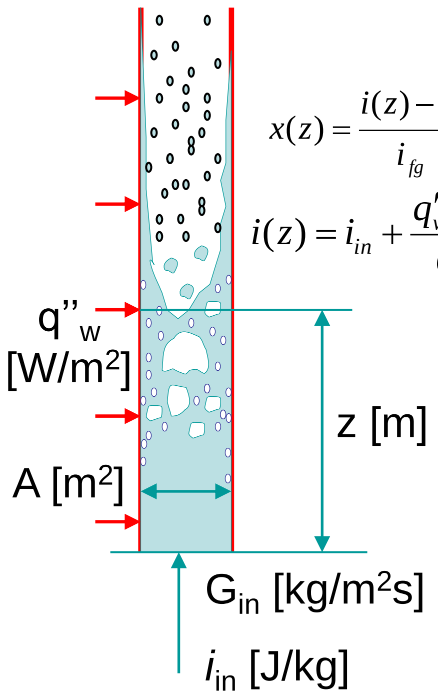
Post-CHF Heat Transfer (5)

- Mechanistc models are taking into account all heat transfer mechanisms but they are very complex
- Simplified approach can be based on using a proper correlation for heat transfer coefficient
- Groeneveld proposed the following correlation for heat transfer in the dispersed flow regime

$$\mathrm{Nu_g} = \frac{hD}{\lambda_g} = a\left[\left(\frac{GD}{\mu_g}\right)\left(x + \frac{\rho_g}{\rho_f}(1-x)\right)\right]^b \mathrm{Pr}_{g,w}^c Y^d \quad \text{where} \quad Y = 1 - 0.1\left(\frac{\rho_f}{\rho_g} - 1\right)^{0.4}(1-x)^{0.4}$$

- Coefficients *a-d* as well as validity ranges are given in the compendium. Note that $\mathrm{Pr}_{g,w}$ is the vapor Prandtl number evaluated at the wall temperature

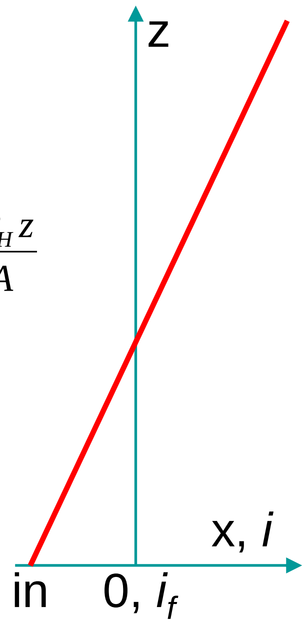
Boiling regimes with constant wall heat flux



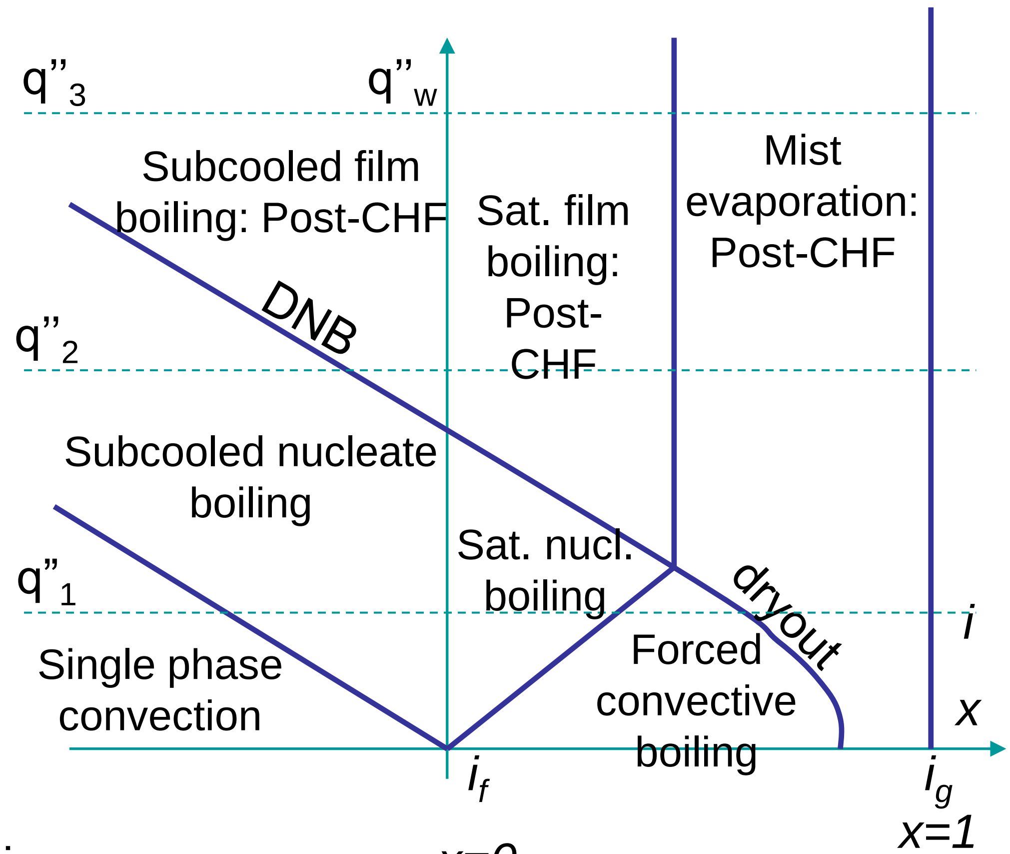
$$x(z) = \frac{i(z) - i_f}{i_{fg}}$$

$$i(z) = i_{in} + \frac{q''_w P_H z}{GA}$$

Boiling channel



Enthalpy distribution



Channel heat transfer regime map