Heat Transfer/Heat Exchanger

- How is the heat transfer?
- Mechanism of Convection
- Applications .
- Mean fluid Velocity and Boundary and their effect on the rate of heat transfer.
- Fundamental equation of heat transfer
- Logarithmic-mean temperature difference.
- Heat transfer Coefficients.
- Heat flux and Nusselt correlation
- Simulation program for Heat Exchanger

How is the heat transfer?

- Heat can transfer between the surface of a solid conductor and the surrounding medium whenever temperature gradient exists.

  Conduction

  Convection

  Natural convection

  Forced Convection

Natural and forced Convection

- Natural convection occurs whenever heat flows between a solid and fluid, or between fluid layers.
- As a result of heat exchange

  Change in density of effective fluid layers taken place, which causes upward flow of heated fluid.

If this motion is associated with heat transfer mechanism only, then it is called Natural Convection

Forced Convection

- If this motion is associated by mechanical means such as pumps, gravity or fans, the movement of the fluid is enforced.
- And in this case, we then speak of Forced convection.

Heat Exchangers

- A device whose primary purpose is the transfer of energy between two fluids is named a Heat Exchanger.

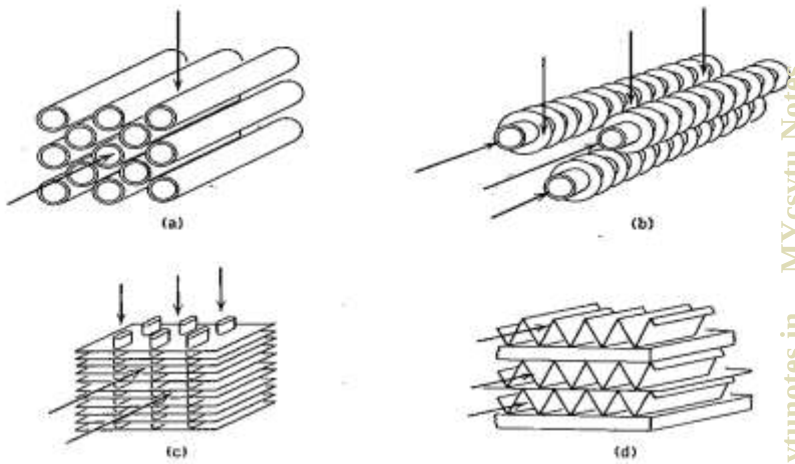


Figure 22.4 Compact heat-exchanger configurations.

Applications of Heat Exchangers

Heat Exchangers prevent car engine overheating and increase efficiency

Heat exchangers are used in AC and furnaces

Heat exchangers are used in Industry for heat transfer

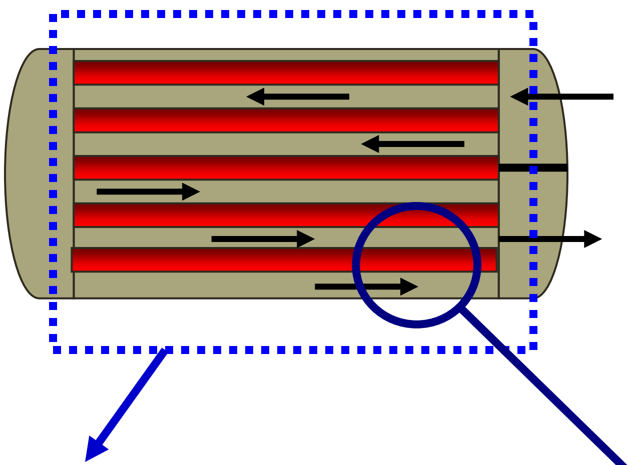
- The closed-type exchanger is the most popular one.
- One example of this type is the Double pipe exchanger.

- In this type, the hot and cold fluid streams do not come into direct contact with each other. They are separated by a tube wall or flat plate.

Principle of Heat Exchanger

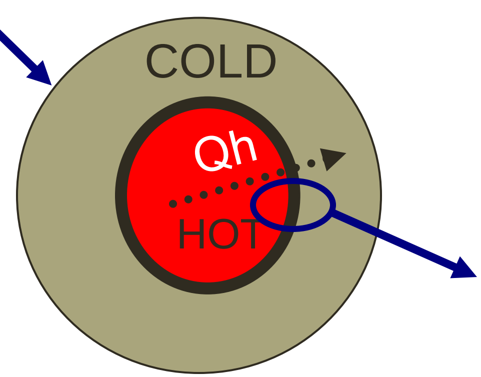
- First Law of Thermodynamic: *"Energy is conserved."*

$$\cancelto{0}{\frac{dE}{dt}} = \left(\sum_{in} \dot{m}.\hat{h}_{in} - \sum_{out} \dot{m}.\hat{h}_{out} \right) + \cancelto{0}{q} + \cancelto{0}{\dot{w}_s} + \cancelto{0}{\dot{e}_{generated}}$$



$$\left.\begin{aligned} Q_h &= A\dot{m}_h.C_p^h.\Delta T_h \\ Q_c &= A\dot{m}_c.C_p^c.\Delta T_c \end{aligned}\right\} \sum_{in} \dot{m}.\hat{h} = -\sum_{out} \dot{m}.\hat{h}$$

•Control Volume



Cross Section Area

Thermal Boundary Layer

THERMAL

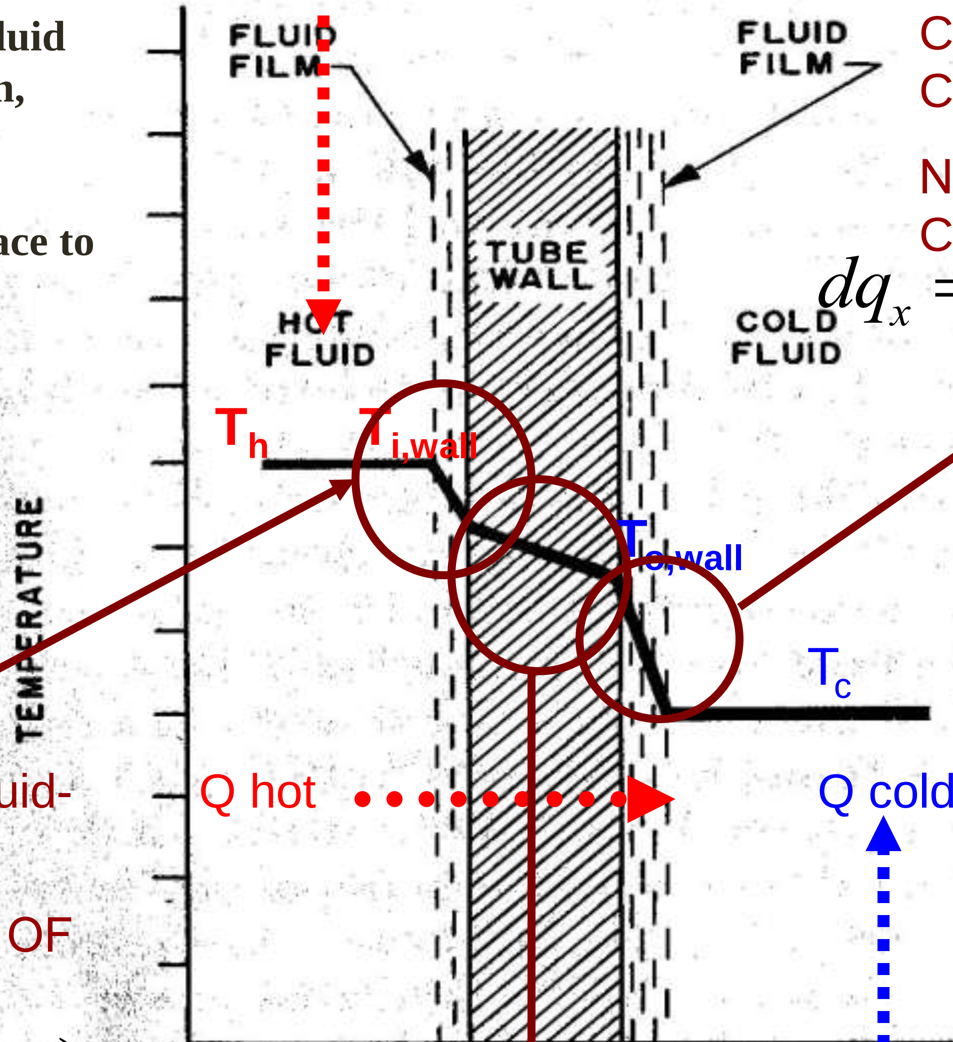
BOUNDARY LAYER

Energy moves from hot fluid to a surface by convection, through the wall by conduction, and then by convection from the surface to the cold fluid.

Region III: Solid – Cold Liquid Convection

NEWTON’S LAW OF CCOLING

$$dq_x = h_c.(T_{ow} - T_c).dA$$



Region I : Hot Liquid-Solid Convection

NEWTON’S LAW OF CCOLING

$$dq_x = h_h.(T_h - T_{iw}).dA$$

Region II : Conduction Across Copper Wall

FOURIER’S LAW

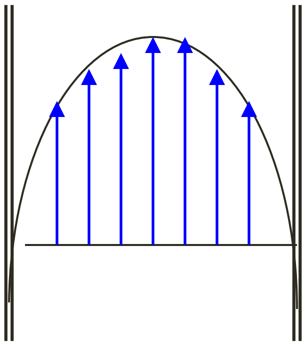
$$dq_x = -k.\frac{dT}{dr}$$

- **Velocity distribution and boundary layer**

When fluid flow through a circular tube of uniform cross-suction and fully developed,

The velocity distribution depend on the type of the flow.

In laminar flow the volumetric flowrate is a function of the radius.



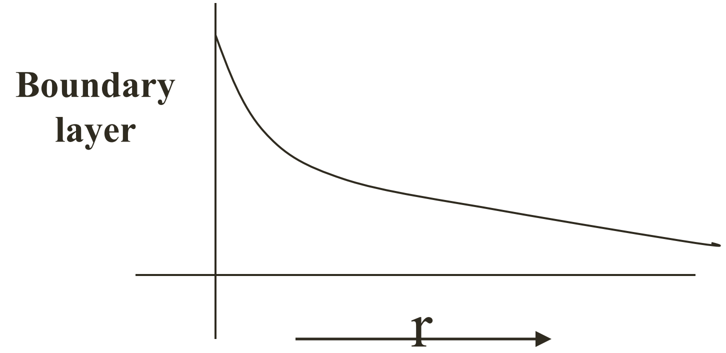
$$V = \int_{r=0}^{r=D/2} u 2\pi r \, dr$$

V = volumetric flowrate

u = average mean velocity

- In turbulent flow, there is no such distribution.

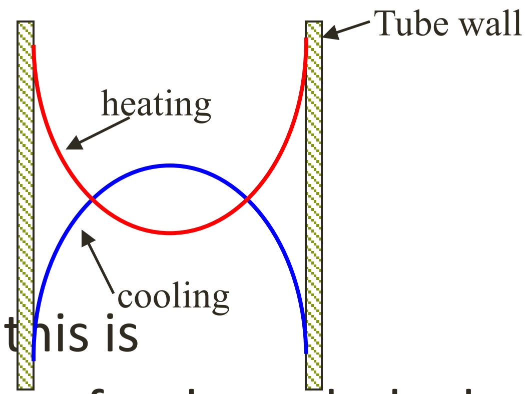
- The molecule of the flowing fluid which adjacent to the surface have zero velocity because of mass-attractive forces. Other fluid particles in the vicinity of this layer, when attempting to slid over it, are slow down by viscous forces.

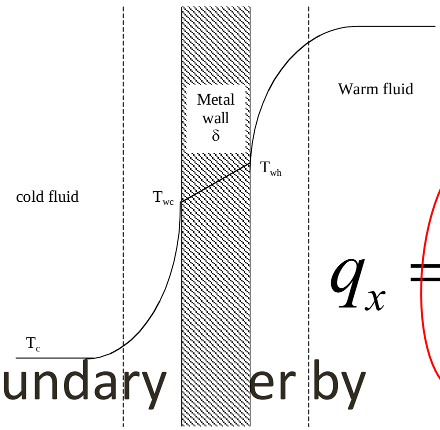


- Accordingly the temperature gradient is larger at the wall and through the viscous sub-layer, and small in the turbulent core.

$$q_x = hA\Delta T$$

$$q_x = hA(T_w - T)$$





$$q_x = \underbrace{\frac{k}{\delta}}_{h} A(T_w - T)$$

- The reason for this is

1) Heat must transfer through the boundary layer by conduction.

2) Most of the fluid have a low thermal conductivity (k)

3) While in the turbulent core there are a rapid moving eddies, which they are equalizing the temperature.

U = The Overall Heat Transfer Coefficient [W/m.K]

Region I : Hot Liquid – Solid Convection

$$q_x = h_{hot}.(T_h - T_{iw}).A \longrightarrow T_h - T_{iw} = \frac{q_x}{h_h.A_i}$$

Region II : Conduction Across Copper Wall

$$q_x = \frac{k_{copper}.2\pi L}{\ln\frac{r_o}{r_i}} \longrightarrow T_{o,wall} - T_{i,wall} = \frac{q_x.\ln\left(\frac{r_o}{r_i}\right)}{k_{copper}.2\pi L}$$

Region III : Solid – Cold Liquid Convection

$$q_x = h_c(T_{o,wall} - T_c)A_o \longrightarrow T_{o,wall} - T_c = \frac{q_x}{h_c.A_o}$$

+

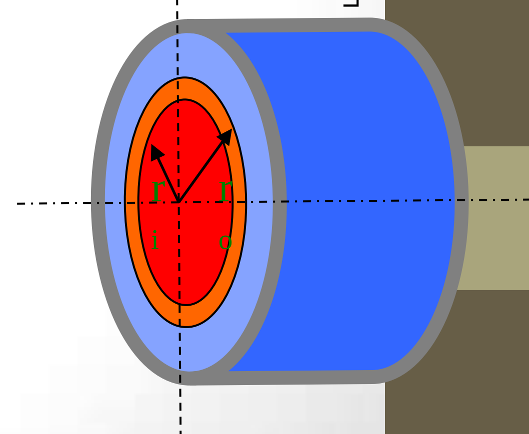
$$T_h - T_c = q_x\left[\frac{1}{h_h.A_i} + \frac{\ln\left(\frac{r_o}{r_i}\right)}{k_{copper}.2\pi L} + \frac{1}{h_c.A_o}\right]$$

$$T_h - T_c = \frac{q_x}{R_1 + R_2 + R_3}$$

$$q_x = U.A.(T_h - T_c)$$

$$U = \frac{1}{A.\Sigma R}$$

$$U = \left[\frac{r_o}{h_{hot}.r_i} + \frac{r_o.\ln\left(\frac{r_o}{r_i}\right)}{k_{copper}.r_i} + \frac{1}{h_{cold}}\right]^{-1}$$



Calculating U using Log Mean Temperature

Hot Stream : $dq_h = \dot{m}_h . C_p^h . dT_h$

Cold Stream: $dq_c = \dot{m}_c . C_p^c . dT_c$

$$d(\Delta T) = dT_h - dT_c$$
$$\Delta T = T_h - T_c$$

$$\rightarrow d(\Delta T) = \left(\frac{dq_h}{m_h . C_p^h} - \frac{dq_c}{m_c . C_p^c} \right)$$

$$dq = -dq_{hot} = dq_{cold}$$
$$-dq = -U.\Delta T.dA$$

$$d(\Delta T) = -U.\Delta T.dA.\left(\frac{1}{m_h . C_p^h} + \frac{1}{m_c . C_p^c} \right)$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U.\left(\frac{\Delta T_h}{q_h} + \frac{\Delta T_c}{q_c} \right).\int_{A_1}^{A_2} dA$$

$$\int_{\Delta T_1}^{\Delta T_2} \frac{d(\Delta T)}{\Delta T} = -U.\left(\frac{1}{m_h . C_p^h} + \frac{1}{m_c . C_p^c} \right).\int_{A_1}^{A_2} dA$$

$$\ln\left(\frac{\Delta T_2}{\Delta T_1} \right) = -\frac{U.A.}{q}\left(\Delta T_h + \Delta T_c \right) = -\frac{U.A}{q}\left[\left(T_h^{in} - T_h^{out} \right) - \left(T_c^{in} - T_c^{out} \right) \right]$$

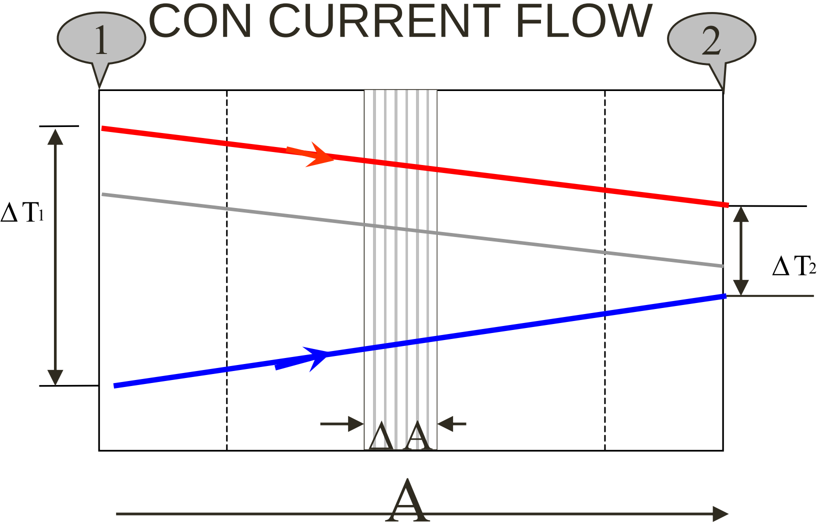
$$q = U.A \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1} \right)}$$

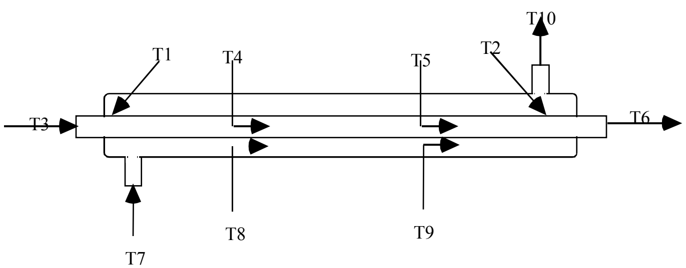
Log Mean Temperature

Log Mean Temperature evaluation

$$\Delta T_{Ln} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \longrightarrow U = \frac{\dot{m}_h . C_p^h .(T_3 - T_6)}{A.\Delta T_{Ln}} = \frac{\dot{m}_c . C_p^c .(T_7 - T_{10})}{A.\Delta T_{Ln}}$$

CON CURRENT FLOW



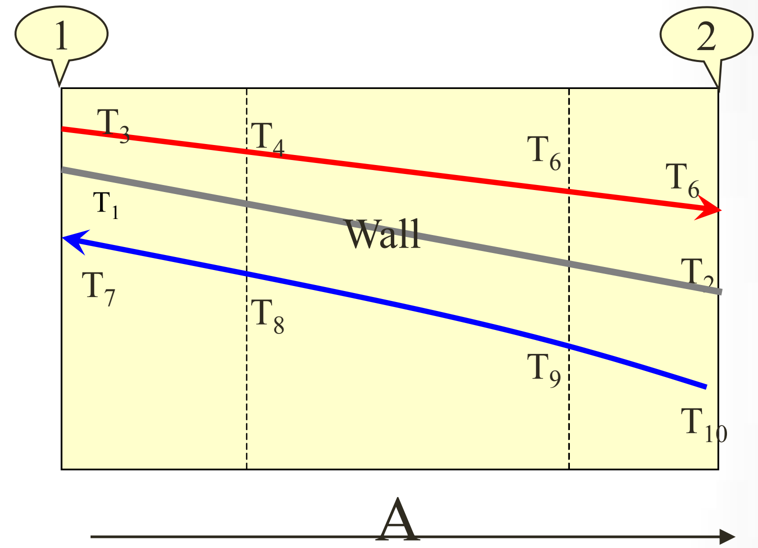


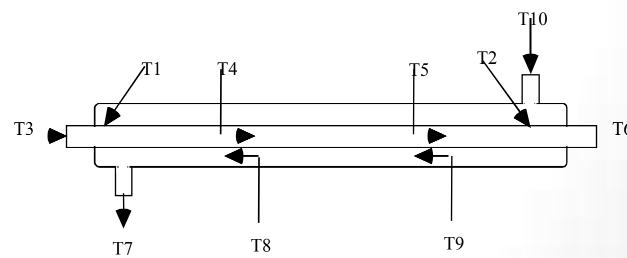
Parallel Flow

$$\Delta T_1 = T_h^{in} - T_c^{in} = T_3 - T_7$$

$$\Delta T_2 = T_h^{out} - T_c^{out} = T_6 - T_{10}$$

COUNTER CURRENT FLOW





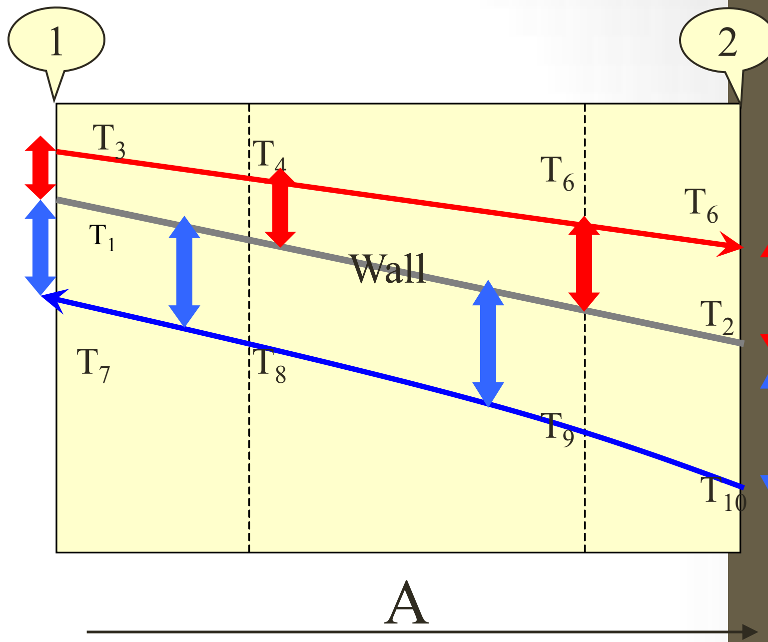
Counter - Current Flow

$$\Delta T_1 = T_h^{in} - T_c^{out} = T_3 - T_7$$

$$\Delta T_2 = T_h^{out} - T_c^{in} = T_6 - T_{10}$$

$$q = h_h \, A_i \, \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{(T_3 - T_1) - (T_6 - T_2)}{\ln \dfrac{(T_3 - T_1)}{(T_6 - T_2)}}$$



$$q = h_c \, A_o \, \Delta T_{lm}$$

$$\Delta T_{lm} = \frac{(T_1 - T_7) - (T_2 - T_{10})}{\ln \dfrac{(T_1 - T_7)}{(T_2 - T_{10})}}$$

DIMENSIONLESS ANALYSIS TO CHARACTERIZE A HEAT EXCHANGER

$$Nu = f(\mathrm{Re}, \mathrm{Pr}, L/D, \mu_b/\mu_o)$$

$$\frac{h.D}{k} \qquad \frac{v.D.\rho}{\mu} \qquad \frac{C_p.\mu}{k}$$

- **Further Simplification:**

$$Nu = a.\mathrm{Re}^b.\mathrm{Pr}^c$$

$$Nu = \frac{D}{\delta}$$

Can Be Obtained from 2 set of experiments

One set, run for constant Pr

And second set, run for constant Re

h

$$q = \frac{k}{\delta} A(T_w - T)$$

•Empirical Correlation

- For laminar flow

Nu = 1.62 (Re*Pr*L/D)

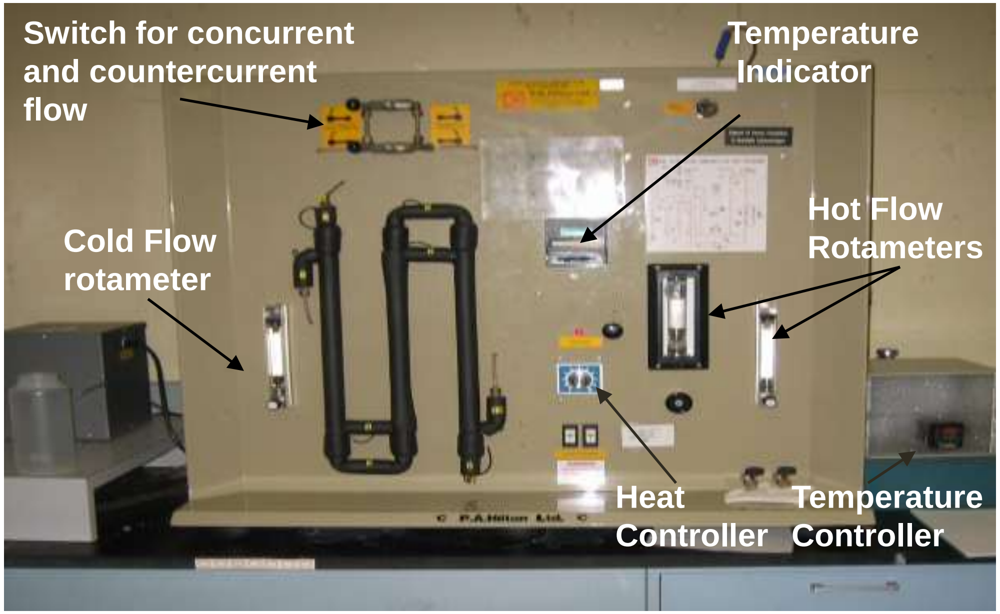
- For turbulent flow

$$Nu_{Ln} = 0.026.\,\mathrm{Re}^{0.8}.\,\mathrm{Pr}^{1/3}.\left(\frac{\mu_b}{\mu_o}\right)^{0.14}$$

- Good To Predict within 20%
- Conditions: L/D > 10
  0.6 < Pr < 16,700
  Re > 20,000

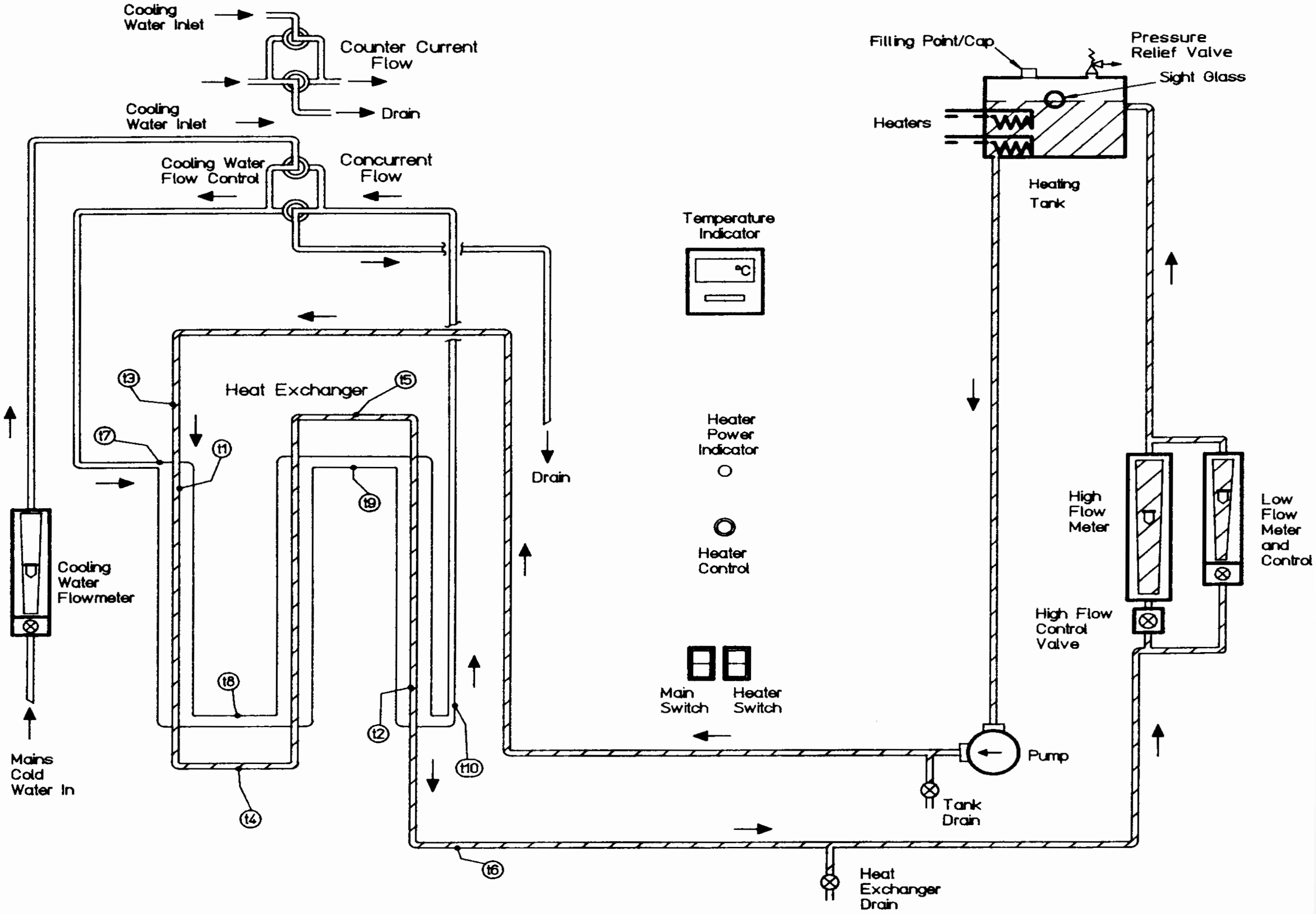
Experimental

Apparatus



- Two copper concentric pipes
  - Inner pipe (ID = 7.9 mm, OD = 9.5 mm, L = 1.05 m)
  - Outer pipe (ID = 11.1 mm, OD = 12.7 mm)
- Thermocouples placed at 10 locations along exchanger, T1 through T10

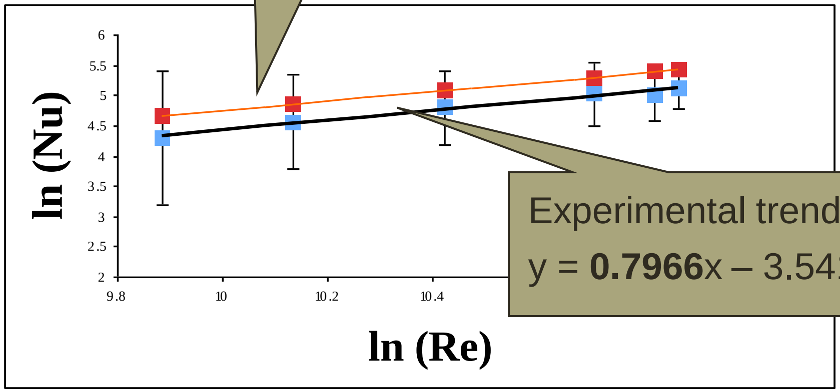
WATER–WATER TURBULENT FLOW HEAT EXCHANGER



s of Exp. Results

Theoretical trend

y = **0.8002**x – 3.0841



Experimental trend

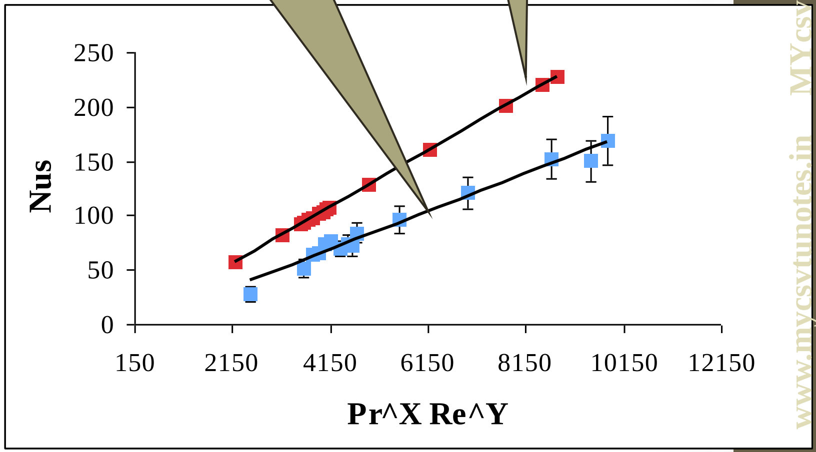
y = **0.7966**x – 3.5415

Theoretical trend

y = **0.026**x

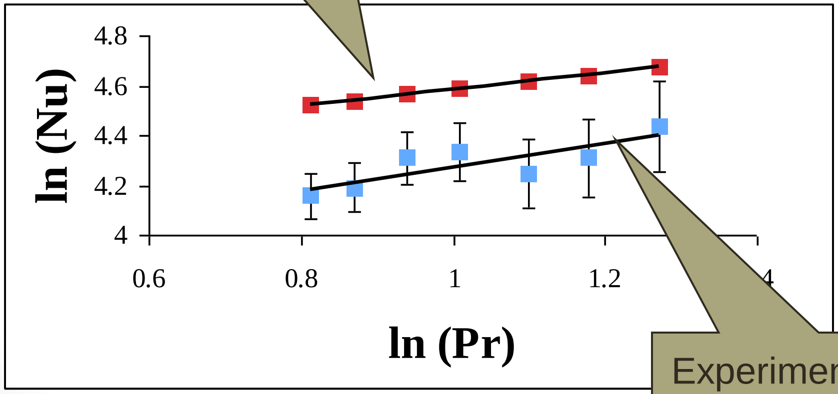
Experimental trend

y = **0.0175**x – 4.049



Theoretical trend

y = **0.3317**x + 4.2533



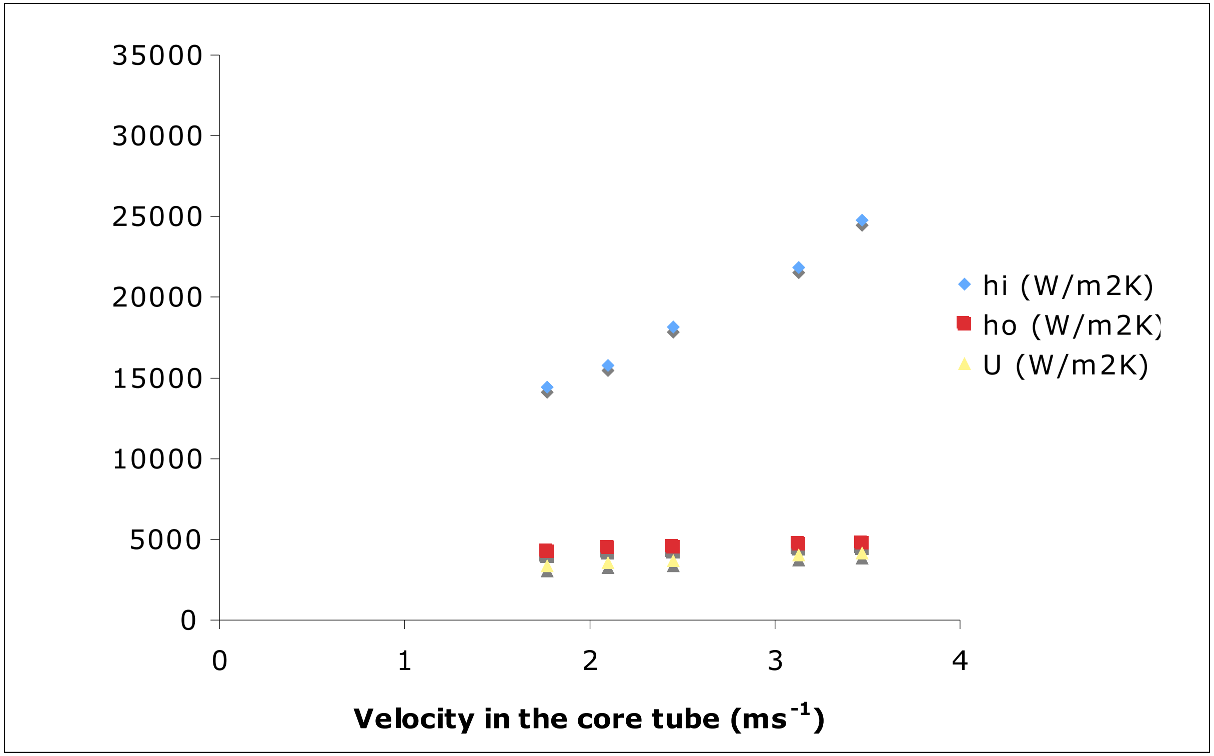
Experimental trend

y = **0.4622**x – 3.8097

Experimental $Nu = 0.0175Re^{0.7966}Pr^{0.4622}$

Theoretical $Nu = 0.026Re^{0.8}Pr^{0.33}$

Effect of core tube velocity on the local and over all Heat Transfer coefficients



Two-Phase Flow Boiling Heat Transfer

P M V Subbarao

Professor

Mechanical Engineering Department

Selection of Optimal Parameters for Healthy and Safe Furnace Walls with Frictional Flow…..

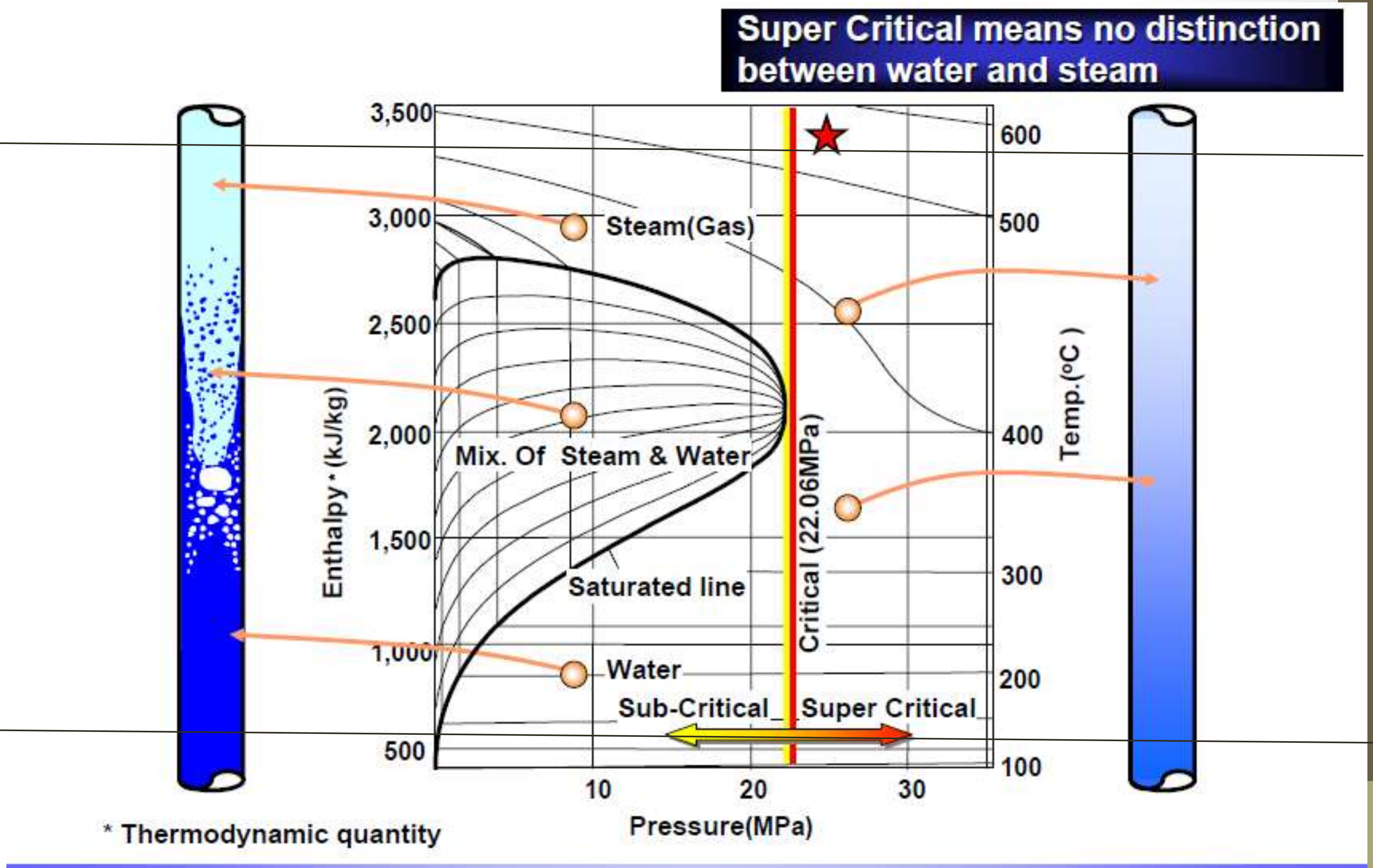
Heat Transfer in Liquid Region

$$q_s'' = h(T_s - T_{sat})$$

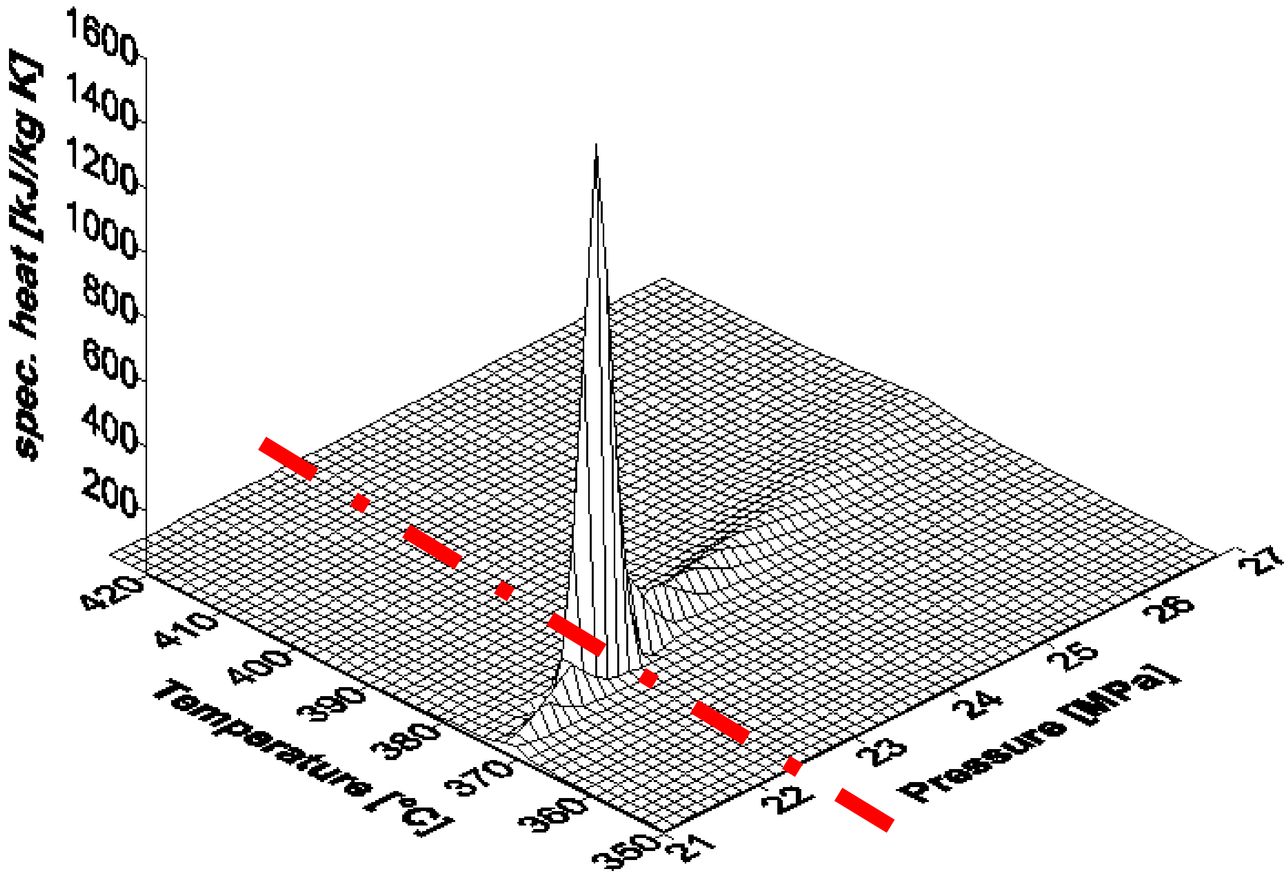
- The liquid in the channel may be in laminar or turbulent flow, in either case the laws governing the heat transfer are well established.
- Heat transfer in turbulent flow in a circular tube can be estimated by the well-known Dittus-Boelter (subcritical) equation.

$$\frac{hD_H}{k_f} = 0.023\,\mathrm{Re}^{0.8}\; pr^{1/3}\left(\frac{k}{d}\right)$$

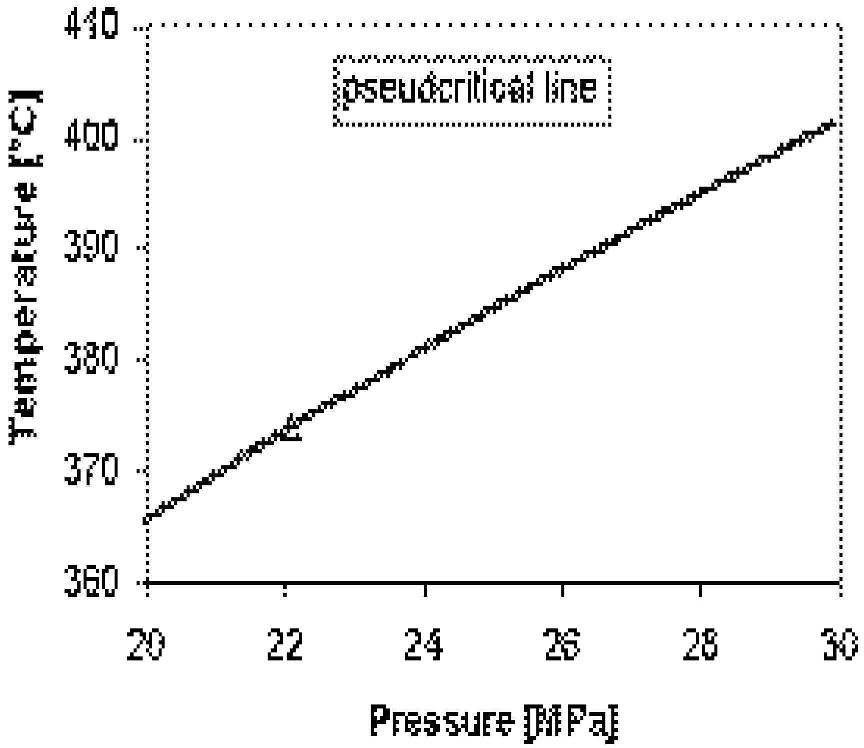
Religious to Secular Attitude

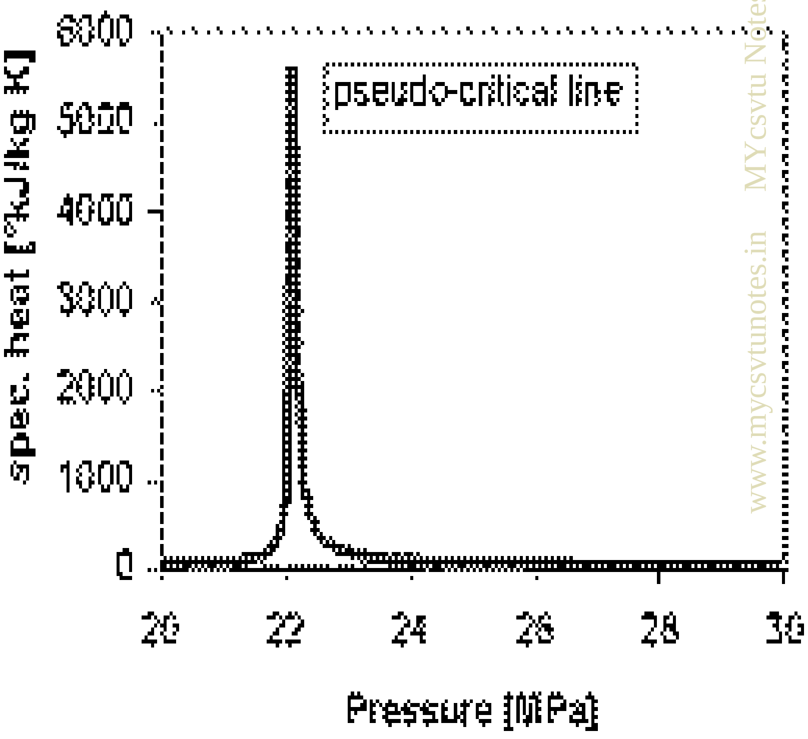


Specific heat of Supercritical Water

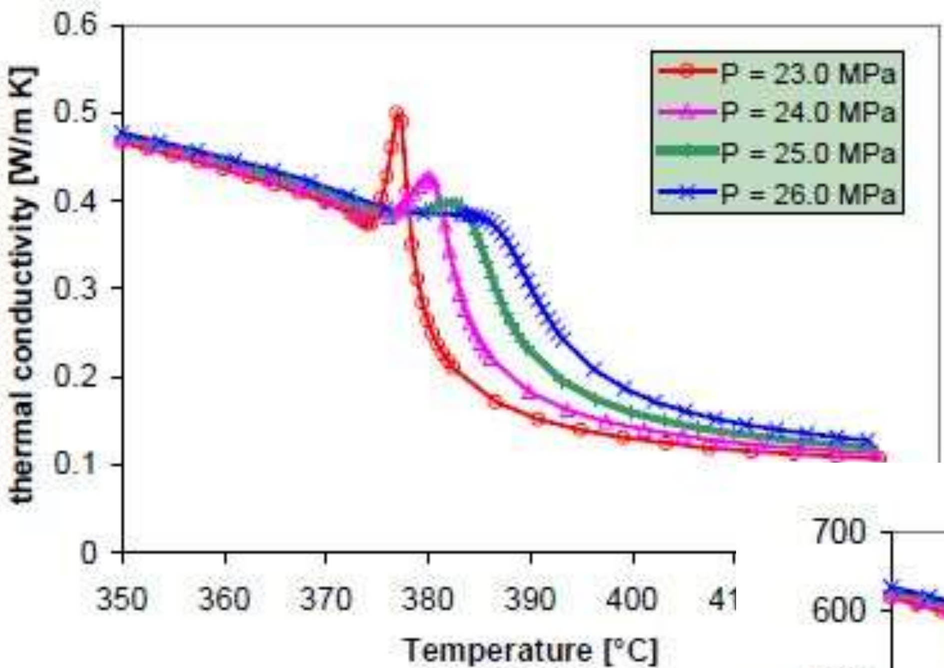


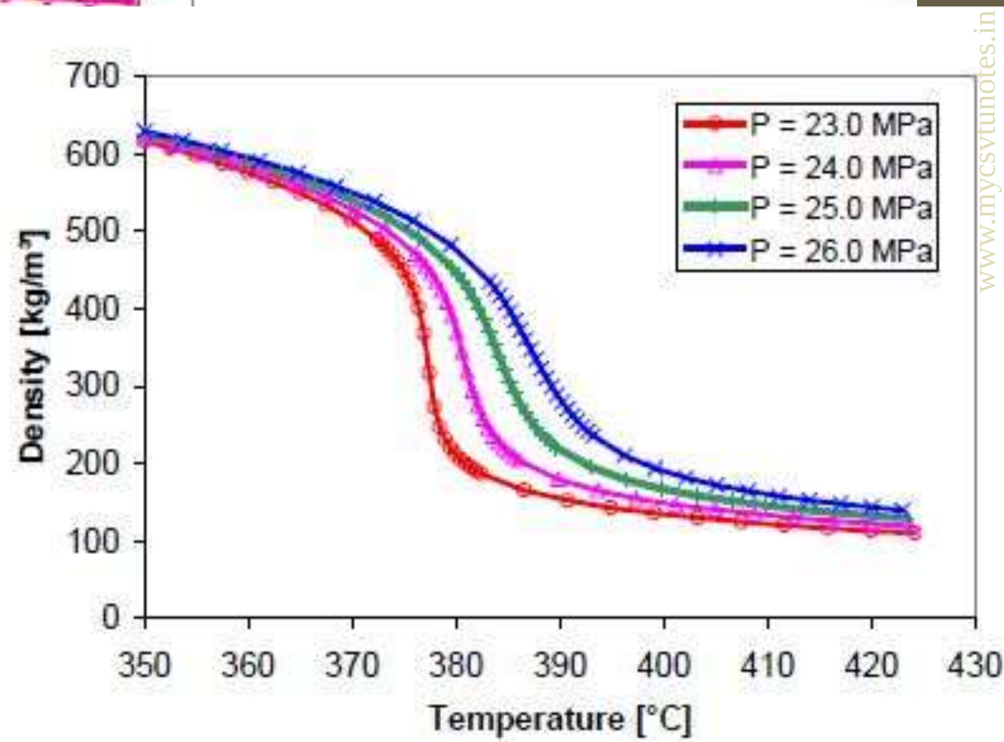
Pseudo Critical Line

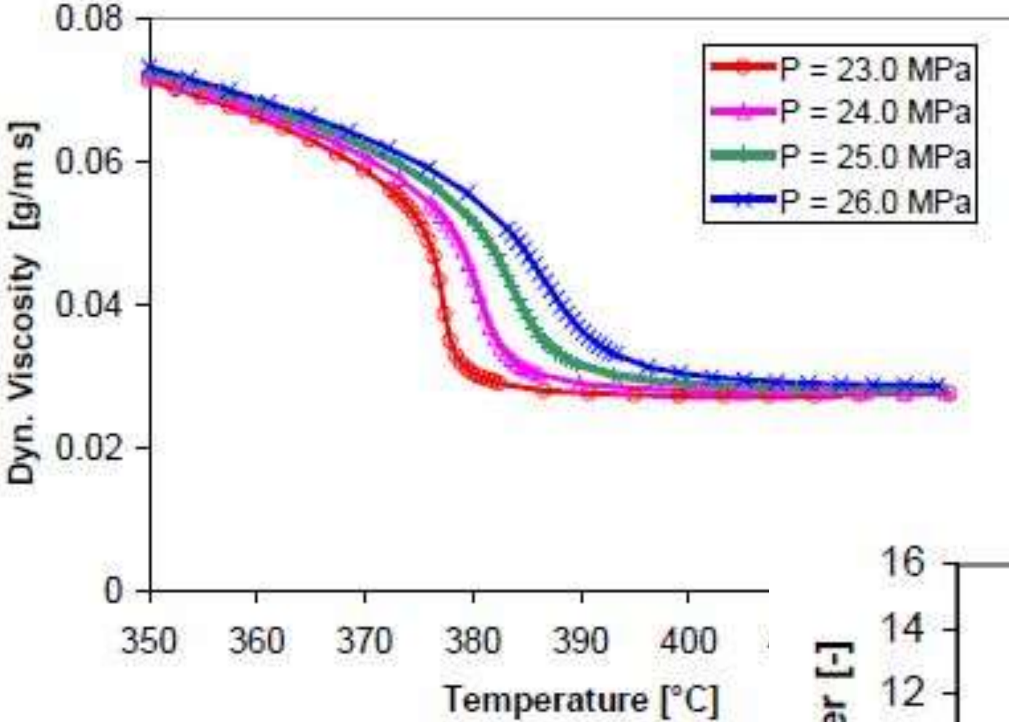


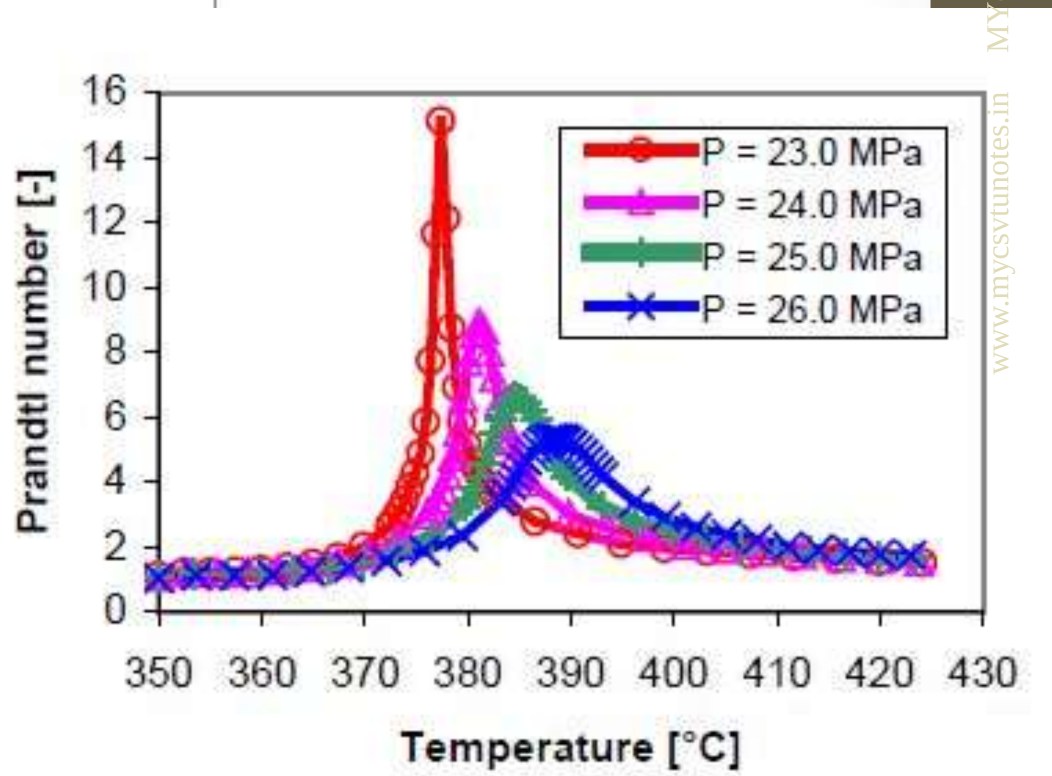


Thermo physical Properties at Super Critical Pressures



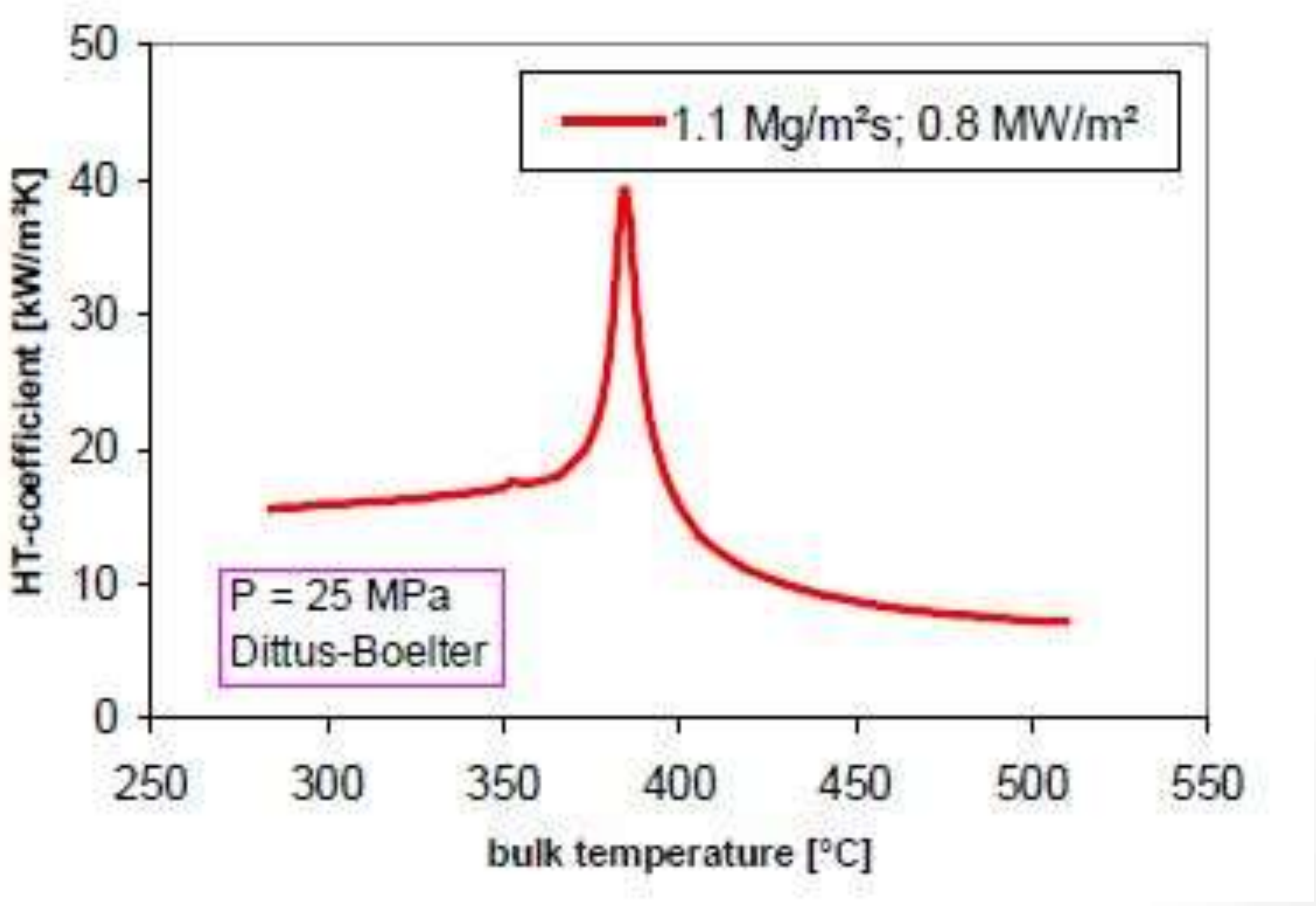



Dyn. Viscosity [g/m s]
0.08
0.06
0.04
0.02
0
350
360
370
380
390
400
Temperature [°C]
P = 23.0 MPa
P = 24.0 MPa
P = 25.0 MPa
P = 26.0 MPa

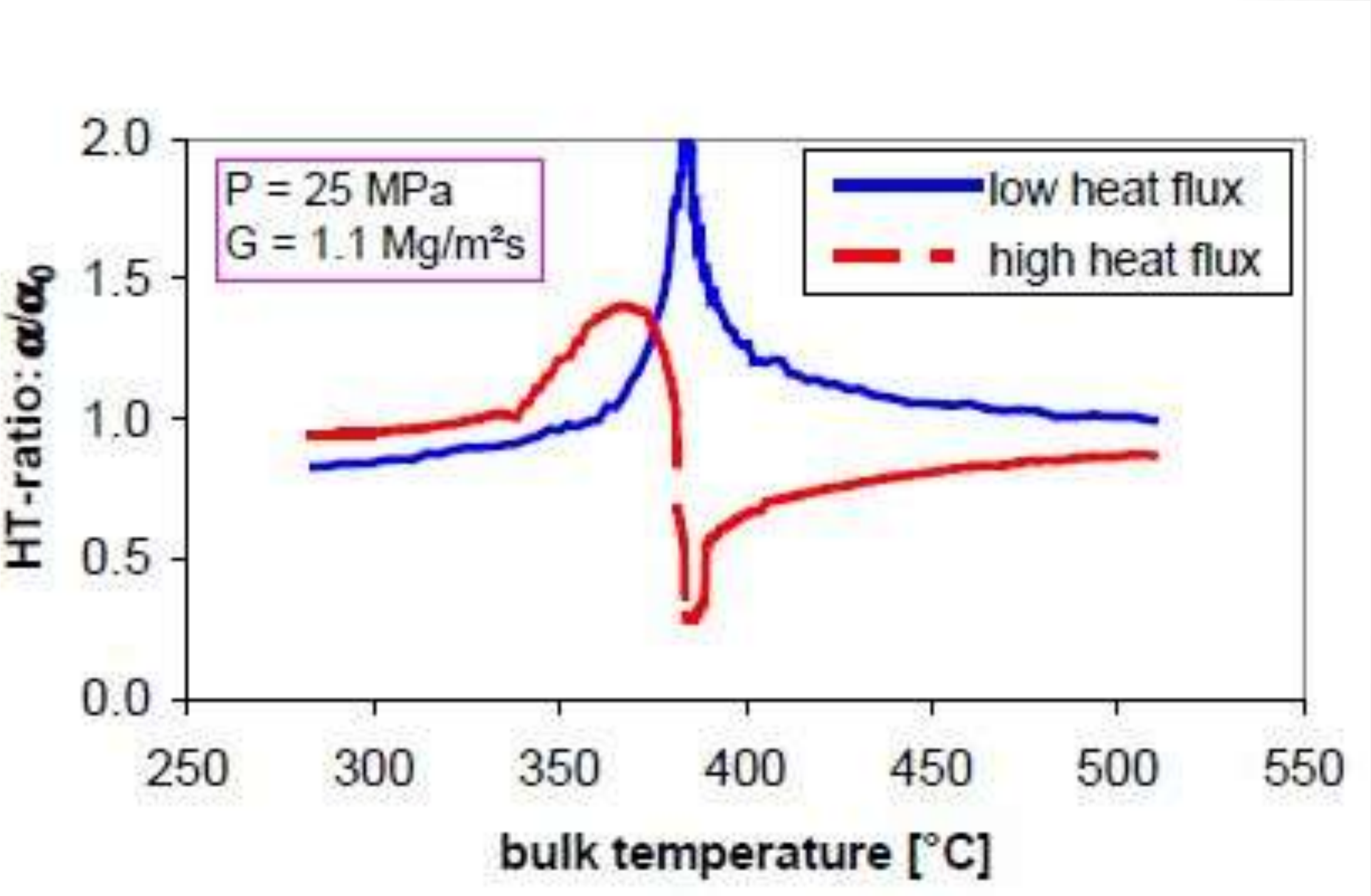

Prandtl number [-]
16
14
12
10
8
6
4
2
0
350
360
370
380
390
400
410
420
430
Temperature [°C]
P = 23.0 MPa
P = 24.0 MPa
P = 25.0 MPa
P = 26.0 MPa

Heat Transfer Coefficient

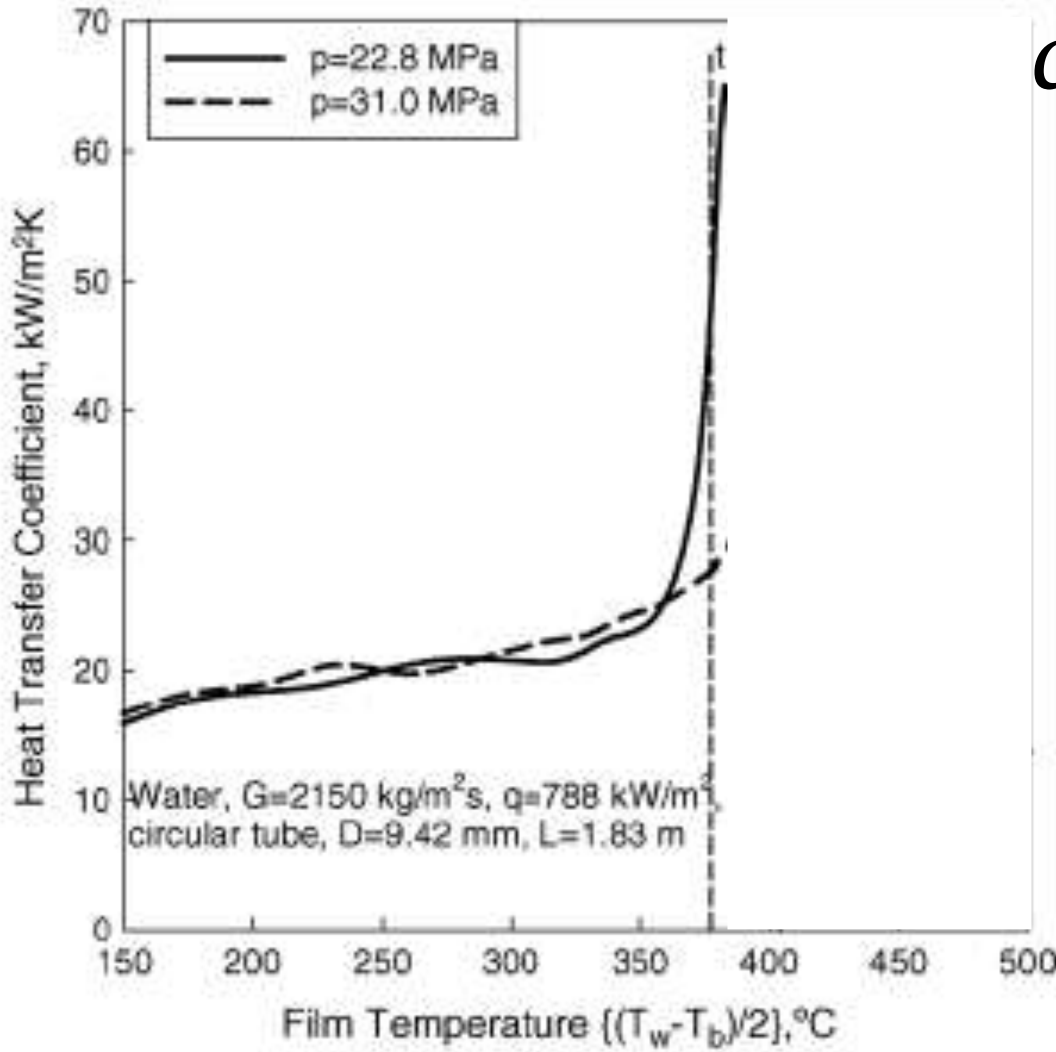
$$\frac{hd}{k_L} = 0.023\,\mathrm{Re}^{0.8}\ pr^{0.4}\left(\frac{k}{d}\right)$$



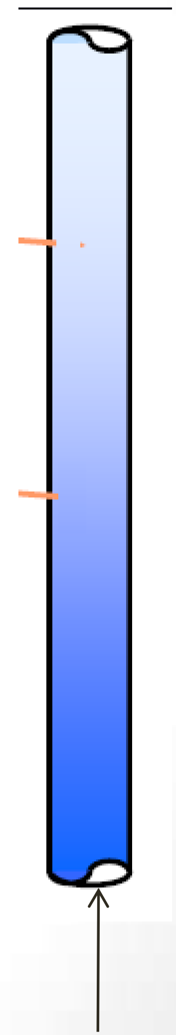
Actual Heat Transfer Coefficient of SC Water

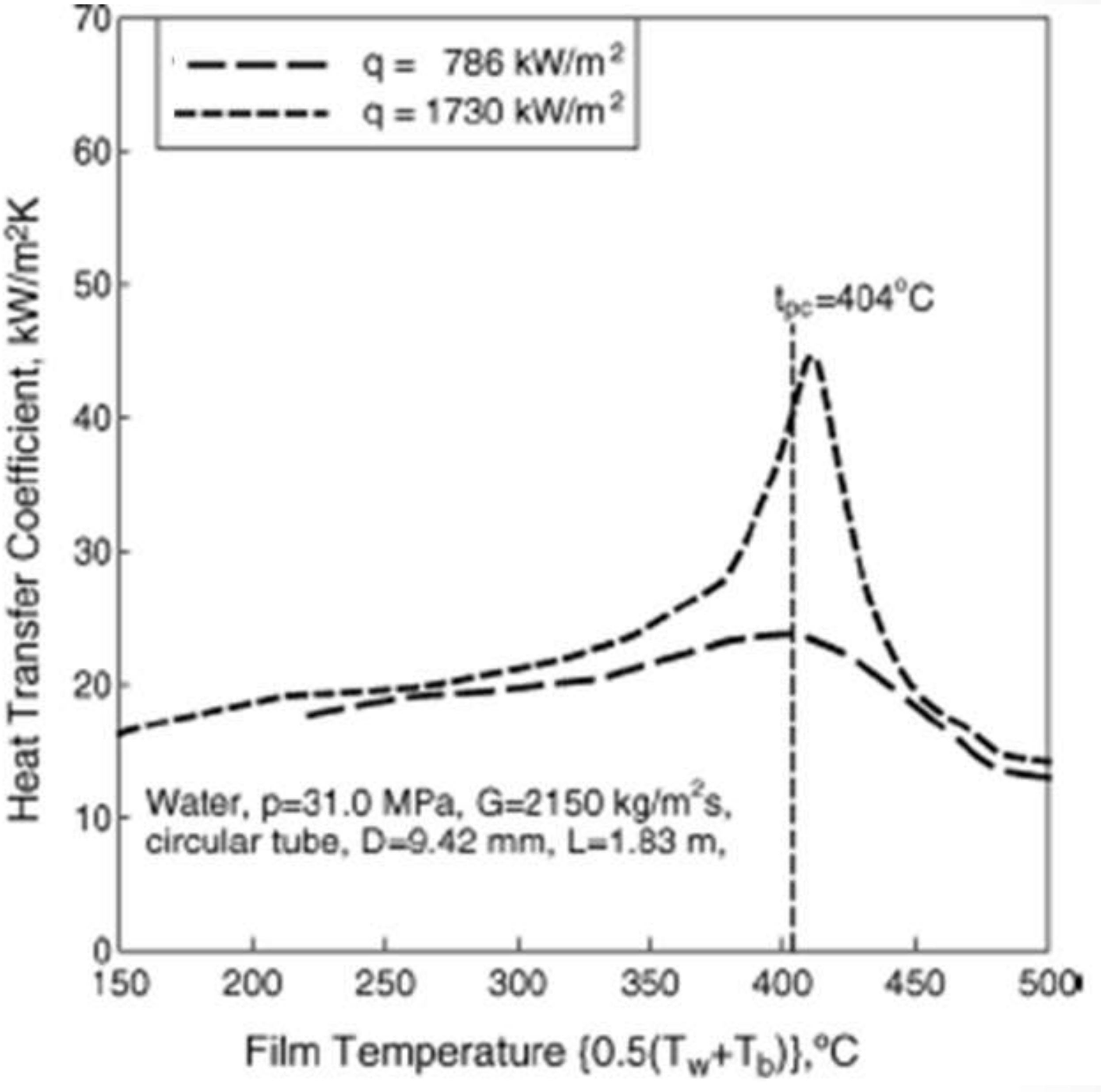


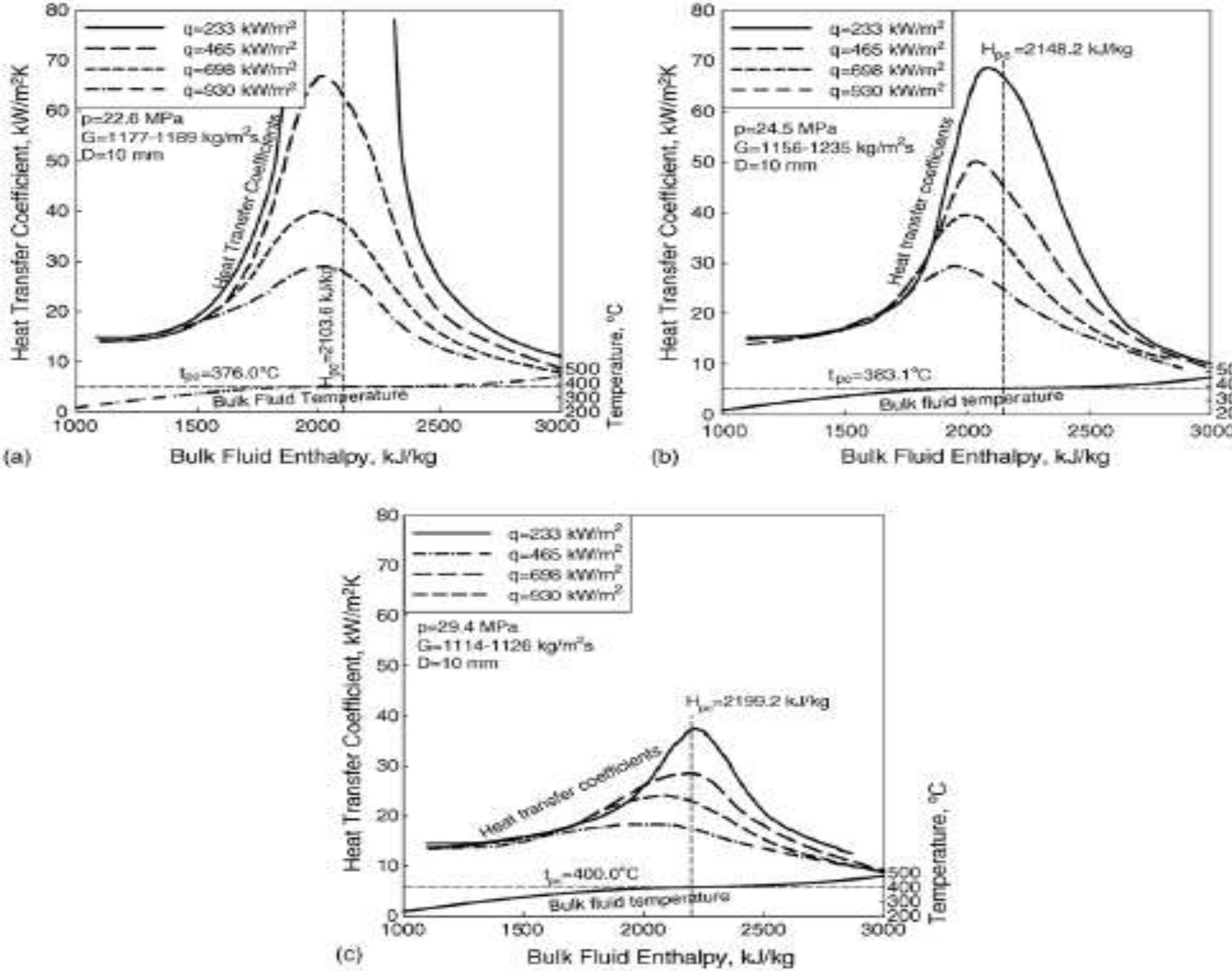
Study of Flow Boiling



$$q_s'' = h(T_s - T_{sat})$$




q = 786 kW/m2
q = 1730 kW/m2
tpc=404°C
Water, p=31.0 MPa, G=2150 kg/m2s,
circular tube, D=9.42 mm, L=1.83 m,
Heat Transfer Coefficient, kW/m2K
Film Temperature {0.5(Tw+Tb)},°C
0
10
20
30
40
50
60
70
150
200
250
300
350
400
450
500


(a)
Heat Transfer Coefficient, kW/m²K
Bulk Fluid Enthalpy, kJ/kg
Temperature, °C
q=233 kW/m²
q=465 kW/m²
q=698 kW/m²
q=930 kW/m²
p=22.6 MPa
G=1177-1189 kg/m²s
D=10 mm
Heat Transfer Coefficients
H_pc=2103.6 kJ/kg
t_pc=376.0°C
Bulk Fluid Temperature
(b)
Heat Transfer Coefficient, kW/m²K
Bulk Fluid Enthalpy, kJ/kg
Temperature, °C
q=233 kW/m²
q=465 kW/m²
q=698 kW/m²
q=930 kW/m²
H_pc=2148.2 kJ/kg
p=24.5 MPa
G=1156-1235 kg/m²s
D=10 mm
Heat transfer coefficients
t_pc=383.1°C
Bulk fluid temperature
(c)
Heat Transfer Coefficient, kW/m²K
Bulk Fluid Enthalpy, kJ/kg
Temperature, °C
q=233 kW/m²
q=465 kW/m²
q=698 kW/m²
q=930 kW/m²
p=29.4 MPa
G=1114-1126 kg/m²s
D=10 mm
H_pc=2199.2 kJ/kg
Heat transfer coefficients
t_pc=400.0°C
Bulk fluid temperature

Single-Phase Liquid Heat Transfer

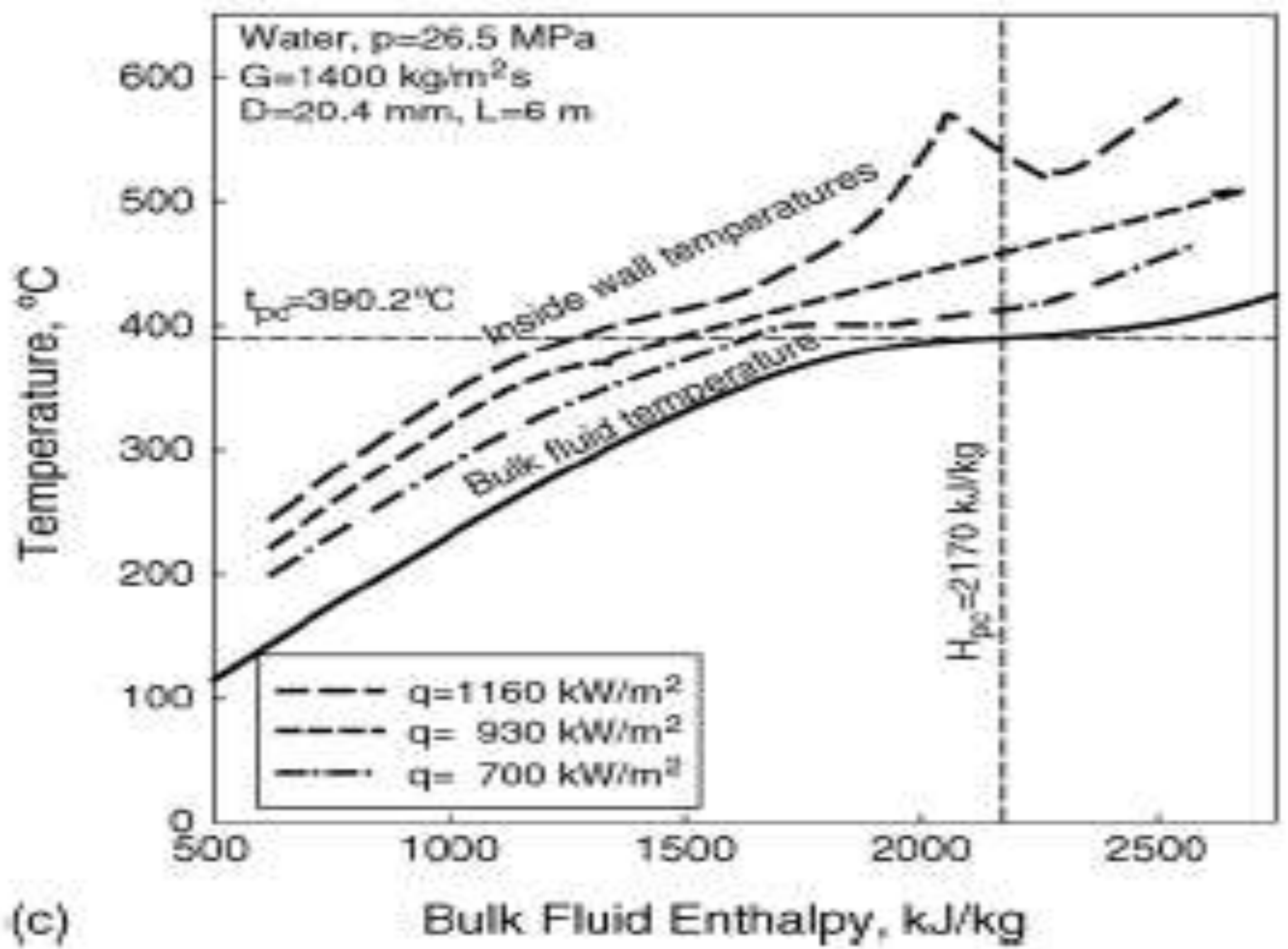
- Under steady state one-dimensional conditions the tube surface temperature is given by:

$$T_{wall}(z) = T_{fluid}(z) + \Delta T_{fw}$$

$$T_{fluid}(z) = T_{fluid,i} + \frac{q'' P z}{G A \dot{C}_{p,fluid}}$$

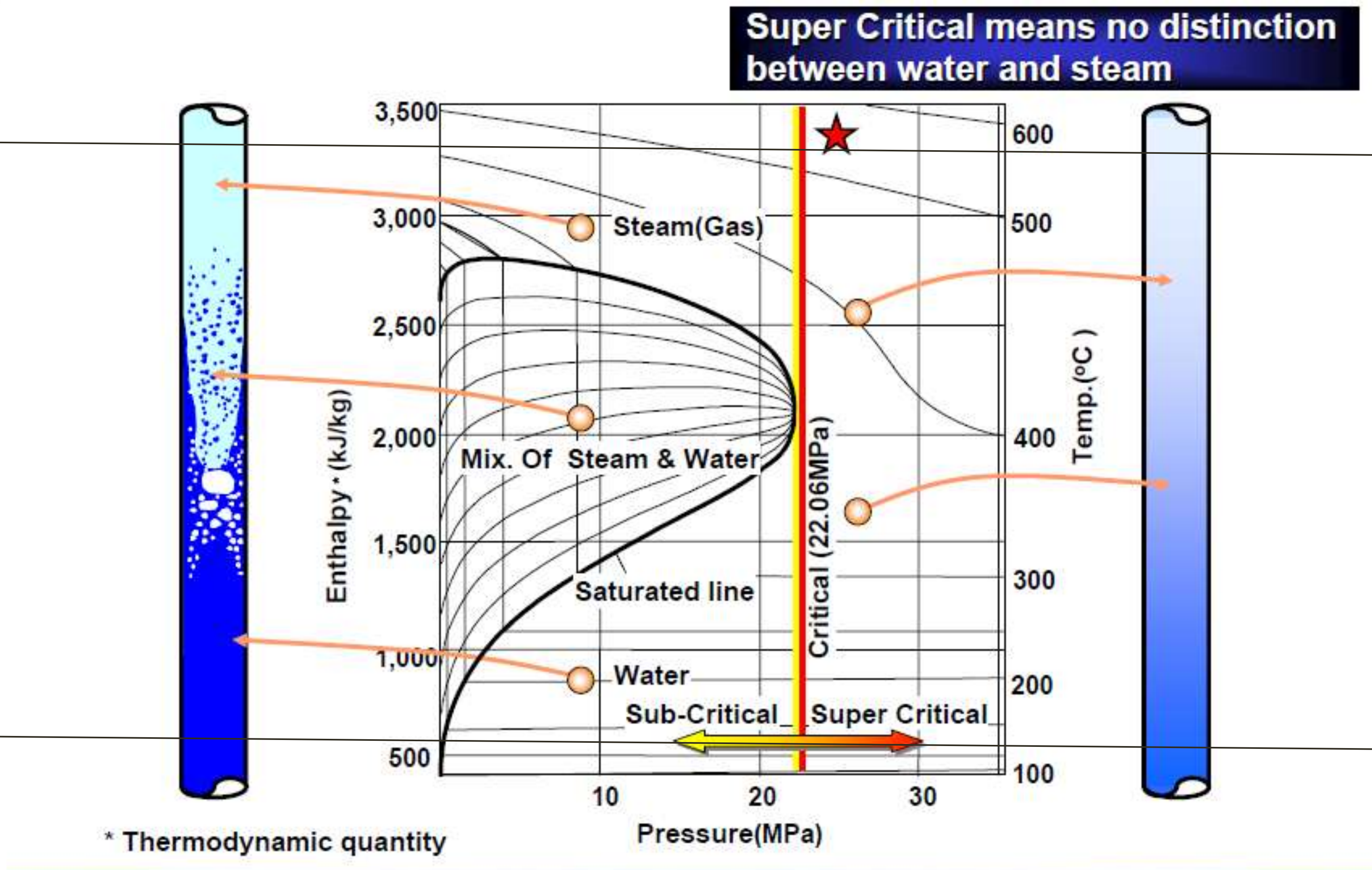
$$\Delta T_{fw} = \frac{q''}{h}$$

- and where
- *q''* is the heat flux,
- *P* is the heated perimeter,
- *G* is the mass velocity,
- *A* is the flow area.
- C_p is the liquid specific heat.
- ΔT_{fw} is the temperature difference.
- *h* is the heat transfer coefficient.



Water, p=26.5 MPa
G=1400 kg/m²s
D=20.4 mm, L=6 m
t_pc=390.2°C
Inside wall temperatures
Bulk fluid temperature
H_pc=2170 kJ/kg
q=1160 kW/m²
q= 930 kW/m²
q= 700 kW/m²
Temperature, °C
0
100
200
300
400
500
600
500
1000
1500
2000
2500
(c)
Bulk Fluid Enthalpy, kJ/kg

The Religious Attitude



The Onset of Nucleate Boiling

- If the wall temperature rises sufficiently above the local saturation temperature pre-existing vapor in wall sites can nucleate and grow.
- This temperature, T_{ONB}, marks the onset of nucleate boiling for this flow boiling situation.
- From the standpoint of an energy balance this occurs at a particular axial location along the tube length, Z_{ONB}.
- For a uniform flux condition,

$$T_{wall,ONB} = T_{wwi} + q'' \left[\frac{PZ_{ONB}}{\dot{m}AC_{pL}} + \frac{1}{h_{cb}} \right]$$

We can arrange this energy balance to emphasize the necessary superheat above saturation for the onset of nucleate boiling

$$T_{wall,ONB} - T_{sat} \equiv \Delta T_{ONB}$$

$$\Delta T_{ONB} = q^{"}\left[\frac{PZ_{ONB}}{\dot{m}AC_{pL}} + \frac{1}{h_{cb}}\right] - \left(T_{sat} - T_{wwi}\right)$$

Now that we have a relation between ΔT_{ONB} and Z_{ONB} we must provide a stability model for the onset of nucleate boiling.

one can formulate a model based on the metastable condition of the vapor nuclei ready to grow into the world.

There are a number of correlation models for this stability line of ΔT_{ONB}.

Bergles and Rohsenow (1964) obtained an equation for the wall superheat required for the onset of subcooled boiling.

Their equation is valid for water only, given by

$$\left(T_W - T_{SAT}\right)_{ONB} = 0.558 \frac{q''}{1082\, p^{1.158}} 0.463\, p^{0.0234}$$

$$\left(T_W - T_{SAT}\right)_{ONB} = \sqrt{\left(\frac{8\sigma q'' T_{SAT}}{k_L h_{fg} \rho_g}\right)}$$

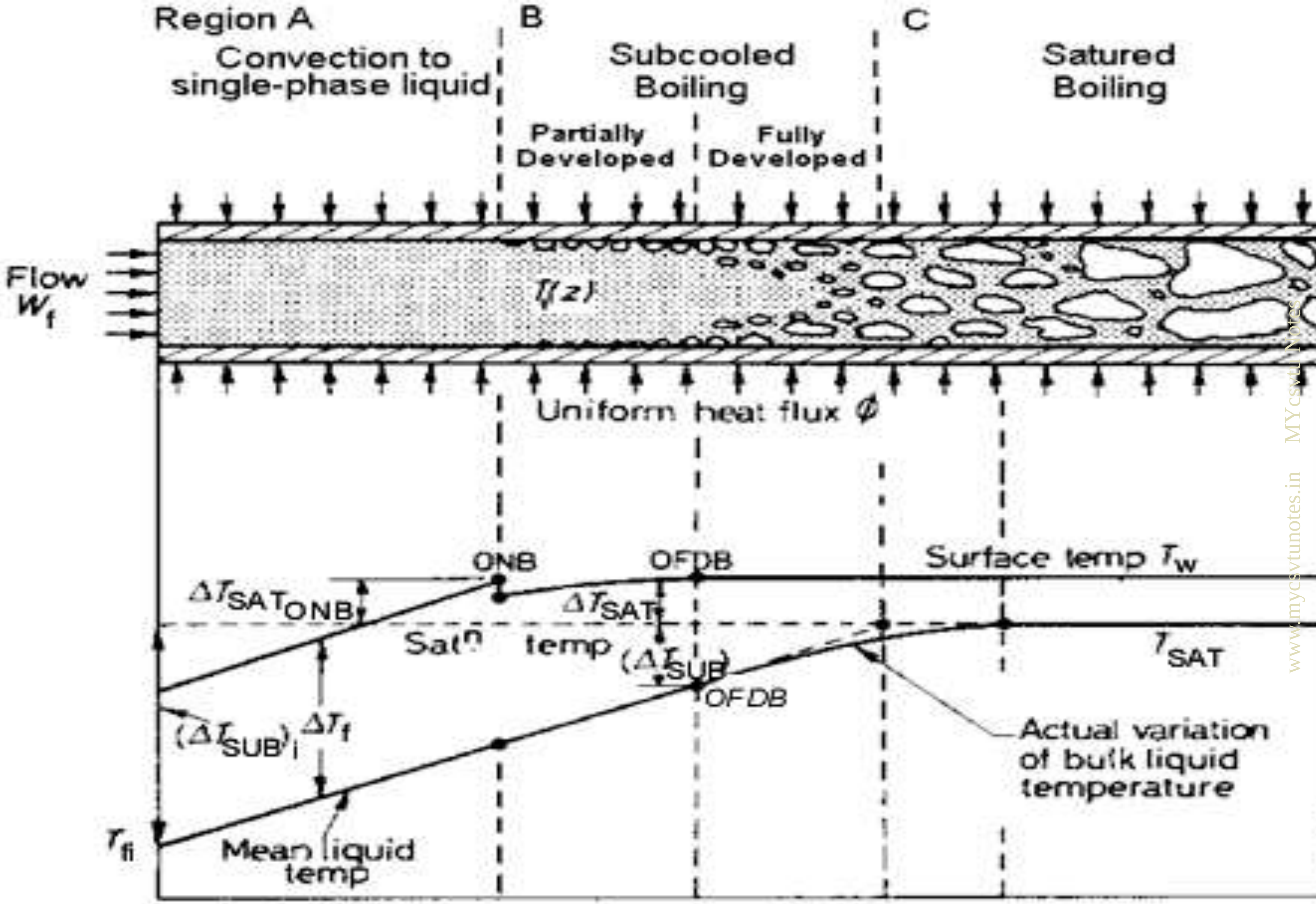


Figure 1. Schematic Representation of the Subcooled Boiling Regimen.

Subcooled Boiling

- The onset of nucleate boiling indicates the location where the vapor can first exist in a stable state on the heater surface without condensing or vapor collapse.
- As more energy is input into the liquid (i.e., downstream axially) these vapor bubbles can grow and eventually detach from the heater surface and enter the liquid.
- Onset of nucleate boiling occurs at an axial location before the bulk liquid is saturated.
- The point where the vapor bubbles could detach from the heater surface would also occur at an axial location before the bulk liquid is saturated.
- This axial length over which boiling occurs when the bulk liquid is subcooled is called the "subcooled boiling" length.
- This region may be large or small in actual size depending on the fluid properties, mass flow rate, pressures and heat flux.
- It is a region of inherent nonequilibrium where the flowing mass quality and vapor void fraction are non-zero and positive even though the thermodynamic equilibrium quality and volume fraction would be zero; since the bulk temperature is below saturation.