Talk
on
Elements of two-phase flows with emphasis on Liquid-Vapour phase change phenomenon

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Outline of topics covered

- Introduction to the topic
- Description of two-phase flow parameters
- Handling 1D governing equations
- Pressure drop model
  - Model for frictional pressure drop
  - Frictional pressure drop characterisation
- Flow pattern
  - Understanding various regimes
  - Typical flow pattern prediction model
- Boiling
  - Pool boiling
  - Bubble growth
  - Regime of boiling, heat transfer characteristics
  - Discussion on convective boiling
- Condensation
Two-phase flows

Boiling and condensation systems mainly encompass co-current gas-liquid flows

We would look at developing the methodologies for predicting, characterising gas–liquid configurations in pipe systems

Some highlights of our focus

✓ Our consideration shall be that the flow is Newtonian, but could be horizontal, vertical or inclined.
✓ Key issue - understanding the various configuration of gas-liquid flows (flow patterns) and also evolving predictable models for pressure drop and heat transfer predictions
✓ We shall restrict to understanding one dimensional treatment.
✓ Careful simplifications required- Even simple situation demand closer look!
Description of two-phase flow parameters

Phase could be solid/liquid/gas

Multiphase flow – Simultaneous flow of several phases

We would look at flows: (i) steam-water – single component two-phase flow
(ii) Air-water- two component two-phase flow.

Approaches

✓ Correlation: quick tool for industrial or even first hand sizing realisation of operational systems
✓ 1D semi analytical: Mixture model are used with appropriate closure relation through correlation

Some mixture models popularly known are:
   Homogeneous flow models
   Separated flow model
   Drift flux model

Note: Flow regimes identification for a particular flow situation seems to be a detrimental aspect to estimate the heat transfer and pressure drop characteristics
Description of two-phase flow parameters

Mass flow rate of the phases:
\[ \dot{m}_f = \rho_f A_f \bar{u}_f; \dot{m}_g = \rho_g A_g \bar{u}_g \]

Phase average velocity:
\[ \bar{u}_f = \frac{Q_f}{A_f}; \bar{u}_g = \frac{Q_g}{A_g} \]

Void Fraction (or) hold-up:
\[ \alpha \rightarrow 0 \text{to} 1 \]
\[ \langle \alpha \rangle = \frac{\int \int \alpha(\bar{r}, t) d\bar{r} dt}{\int d\bar{r} \int dt} \]
\[ \alpha = \frac{\bar{A}_g}{A} = \frac{\bar{V}_g}{V} \]

Void Fraction (or) hold-up:
For steady flow situation, it common to use simply an area or volume average

Slip between phases:
\[ s = \frac{\bar{u}_g}{\bar{u}_l} \]
\[ \bar{u}_r = \bar{u}_g - \bar{u}_l \]

Average velocity (or) volumetric flux
\[ j = j_l + j_g \]

Base velocity
\[ j = \left[ \frac{Q_l + Q_g}{A} \right] \]
Description of two-phase flow parameters

No slip hold-up: (volumetric flow fraction)
\[ \beta = \frac{\dot{V}_g}{\dot{V}_f + \dot{V}_g} \]

Following relations are useful:
\[ j_g = \alpha u_g ; j_l = (1-\alpha)u_l \]

Drift velocity:
\[ u_d = (u_g - j)or(u_f - j) \]

Drift flux:
\[ j_{gf} = \alpha u_{gj} \]

Note: \( u_{gj} \) is \( u_d \) (drift velocity)

Mass fluxes:
\[ G_g = Gx \]
\[ G_f = G(1-x) \]

Usual void–quality relation
\[ \Rightarrow \alpha = \frac{1}{\left(\frac{1-x}{x}\right)\frac{\rho_g}{\rho_f}} s + 1 \]

Momentum fluxes:
\[ G_g j_g = \frac{G^2 x^2}{\rho_g^2} \]
\[ G_f j_f = \frac{G^2 (1-x)^2}{\rho_l^2} \]
Description of two-phase flow parameters

Homogeneous model considerations:
- Both the phases modeled to have same velocity and same temperature
- All parameters are treated as average pseudo fluid properties- no description of flow pattern

Homogeneous void fraction:
\[ u_g = \frac{m_g x}{\rho_g \alpha}; \quad u_f = \frac{m_f (1-x)}{\rho_g (1-\alpha)} \]

Equal velocity means
\[ \alpha_H = \frac{\frac{x}{\rho_g} \frac{1-x}{\rho_f} + \frac{x}{\rho_g}}{1} \quad \Rightarrow \alpha_H = \frac{1}{\left(\frac{1-x}{x}\right) \frac{\rho_g}{\rho_f} + 1} \]

Mixture density (homogeneous)
\[ \rho_H = \rho_g \alpha + \rho_f (1-\alpha) \]

Usual void – quality relation
\[ \Rightarrow \alpha = \frac{1}{\left(\frac{1-x}{x}\right) \frac{\rho_g}{\rho_f} s + 1} \]

Void-vol.flow fraction- relation:
\[ \beta = \frac{\dot{V}_g}{\dot{V}_f + \dot{V}_g} \]
\[ = \frac{A_g u_g}{A_g u_g + A_f u_f} \]
\[ \beta = \alpha_H \text{ (No-slip case)} \]
\[ \text{else}, \alpha > \beta \]
Review of 1 D Governing Equation

➢ Conservation of mass

- Gas phase:
  \[
  \frac{\partial (\rho_g A_g)}{\partial t} + \frac{\partial m_g}{\partial s} = \pm \Gamma_g'
  \]

- Liquid:
  \[
  \frac{\partial (\rho_l A_l)}{\partial t} + \frac{\partial m_l}{\partial s} = \pm \Gamma_l'
  \]

In case of boiling (single comp. 2Φ) flow \( \Gamma_g' = -\Gamma_l' \)

➢ Conservation of momentum:

- Conservative form.
  \[
  \frac{\partial (\dot{m})}{\partial t} + \frac{\partial (\dot{m}u)}{\partial s} = -A \frac{\partial p}{\partial s} - \tau_w P_w - \rho A g \sin \theta
  \]

- Non-Conservative form (primitive variable form).
  \[
  \rho A \frac{\partial u}{\partial t} + \rho A u \frac{\partial u}{\partial s} = -A \frac{\partial p}{\partial s} - \tau_w P_w - \rho A g \sin \theta
  \]
Review of 1 D Governing Equation

➢ Conservation of Energy :

\[
\rho A \frac{\partial h}{\partial t} + \rho Au \frac{\partial h}{\partial s} = uA \frac{\partial p}{\partial s} + A \frac{\partial p}{\partial t} + q_{\text{surf}} P_h - \frac{\partial (q_{\text{axial}} A)}{\partial s} - W
\]

**General choice of Independent variables:** \( A, H, s, P_W, P_h, t \)

- Equations =3

Closing relations=5

\[ \rho = \rho(p,h) \]

\[ q_{\text{surface}} = \text{specified} \]

(or) \[ q_{\text{surface}} = f(u, \text{properties}) \]

\[ \tau_w = f(\rho, \text{geometry}, u, \mu) \]

\[ m = \rho Au \quad q_{\text{axial}} \rightarrow \text{neglected usually} \]

Appropriate to a particular situation, these take shape as equations through empirical relation or mechanistic relations.
A simple way of handling 2Φ pressure drop

- Consider the momentum conservation

\[ \rho A \frac{\partial u}{\partial t} + \rho A u \frac{\partial u}{\partial s} = -A \frac{\partial p}{\partial s} - \tau_w P_w - \rho A g \sin \theta \]

Let us rewrite this for pipe flow with a mass flux \( G \) for steady state:

\[ -\frac{dp}{dz} = \frac{4\tau}{d} + \rho g \sin \theta + G^2 \frac{d}{dz} \left( \frac{1}{\rho} \right) \]

Total Pressure Grad. = (Fric. Pressure. Grad.)+(Grav. Pressure Grad.)+(Acc. Pressure Grad.)

The question now is how do we have a mixture density and mixture viscosity defined?
A simple way of handling $2\Phi$ pressure drop

**Ref:**

- Define

$$f = \frac{\tau_w}{1 \frac{\rho u^2}{2}}$$

$$Re = \frac{Gd}{\mu}$$

$$\frac{1}{\sqrt{f}} = 3.48 - 4 \log_{10} \left[2e/D + 9.35/(Re \sqrt{f})\right].$$

$$f = \frac{\rho D \left[\frac{dp}{dz}\right]}{2G^2}$$

$$Re = \frac{DG}{\mu}$$

**Design of Concentrated Solar Thermal Systems** during December 16 - 18, 2013

17 Dec 2013
Difficulty in characterising $2\Phi$ – Frictional pressure drop

Simple $1\Phi$ situation:

- Cold condition: $\Delta p_{DPT} = (p_1 - p_2)$

$$\Delta p_{DPT} = (P_A + \rho_{im} g h_1) - (P_B + \rho_{im} g h_2)$$

$$\Delta p_{DPT} = (P_A - P_B) + \rho_{im} g h_1 - \rho_{im} g h_2$$

$$\Delta p_{DPT} = \Delta p_f + \rho_p g H - \left( \frac{\rho_p v_1^2}{2} - \frac{\rho_p v_2^2}{2} \right) - \left( \frac{\rho_p v_3^2}{2} - \frac{\rho_p v_1^2}{2} \right) + \rho_{im} g h_1 - \rho_{im} g h_2$$

$$\Delta p_{DPT} = \Delta p_f + \rho_p g H + \left( \frac{\rho_p v_3^2}{2} - \frac{\rho_p v_1^2}{2} \right) + \rho_{im} g h_1 - \rho_{im} g h_2$$

For cold condition, $\rho_{im} = \rho_p$  Also $H = (h_2 - h_1)$

$$\Delta p_{DPT} = \Delta p_f + \rho_p g h_2 - \rho_p g h_1 + \left( \frac{\rho_p v_3^2}{2} - \frac{\rho_p v_1^2}{2} \right) + \rho_{im} g h_1 - \rho_{im} g h_2$$

$$\Delta p_{DPT} = \Delta p_f + \frac{\rho_p v_1^2}{2} \left( \frac{v_3}{v_1} \right)^2 - 1$$

$$\Delta p_{DPT} = \Delta p_f + \frac{8m^2}{\rho_p \pi^2 d_1^4} \left( \frac{d_1}{d_3} \right)^4 - 1$$
Difficulty in characterising $2\Phi$ – Frictional pressure drop

**Simple $1\Phi$ situation:**

- **Hot condition:**
  
  For Hot condition, $\rho_{im} > \rho_p$
  
  $\Delta \rho = \rho_{im} - \rho_p$

  The expression for pressure drop sensed by DPT may be written as

  \[
  \Delta p_{DPT} = \Delta p_f + \frac{8m^2}{\rho_p \pi^2 d_1^4} \left( \frac{d_1}{d_3} \right)^4 - 1 \right] + \rho_p gH - \rho_{im} g(h_2 - h_1)
  \]

  \[
  \Delta p_{DPT} = \Delta p_f + \rho_p gH - \rho_{im} gH + \frac{8m^2}{\rho_p \pi^2 d_1^4} \left[ \left( \frac{d_1}{d_3} \right)^4 - 1 \right]
  \]

  $\Delta p_{DPT} = \Delta p_f - \Delta \rho gH + \frac{8m^2}{\rho_p \pi^2 d_1^4} \left[ \left( \frac{d_1}{d_3} \right)^4 - 1 \right]$  

  ➢ Now presence of $2\Phi$ imposes more complications in terms of density variation along the length which is not clearly known apriori
Frictional Pressure drop-Homogeneous model

When does this work well:

- For this density ratios to be valid, steam water mixtures are to have $p > 120 \text{ bar}$

$$\left(-\frac{dp}{dz}\right)_{fric} = \frac{4\tau_w}{d} = \frac{4}{d} f_h \frac{1}{2} \frac{G^2}{\rho_h}$$

- Usual practice is to work with Two-phase multipliers which are factors by which single phase (gas or liquid) pressure gradient must be multiplied to get the value for two-phase pressure gradient.

$$\left(-\frac{dp}{dz}\right)_{fric} = \left(-\frac{dp}{dz}\right)_{lo} \phi_{lo}^2$$

$$\rho_l < 10; \ \ G > 2000 \text{ kg/m}^2\text{s} \quad \frac{\rho_l}{\rho_g}$$
Frictional Pressure drop-
(Lockhart-Martinelli model)

Using the model:

➢ The frictional pressure drop is written as

\[
\left( -\frac{dp}{dz} \right)_{fric} = \left( -\frac{dp}{dz} \right)_t + C \left[ \left( -\frac{dp}{dz} \right)_l + \left( -\frac{dp}{dz} \right)_g \right]^{1/2} + \left( -\frac{dp}{dz} \right)_g
\]

\[
\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}; \quad \phi_g^2 = 1 + CX + CX^2
\]

➢ For a homogeneous cases a limiting value of C is calculated, where the phases interact well with same velocity.

\[
C = \left( \frac{\rho_l}{\rho_g} \right)^{1/2} + \left( \frac{\rho_g}{\rho_l} \right)^{1/2}
\]

Define

\[
X^2 = \frac{\left( \frac{dp}{dz} \right)_t}{\left( \frac{dp}{dz} \right)_g}
\]

<table>
<thead>
<tr>
<th>Liquid</th>
<th>Gas</th>
<th>Value of C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulent</td>
<td>Turbulent</td>
<td>20</td>
</tr>
<tr>
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<td>12</td>
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<td>10</td>
</tr>
<tr>
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<td>Laminar</td>
<td>05</td>
</tr>
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</table>
For Air-water flows, typically seen flow patterns in vertical flow: (Adiabatic flow)

Increasing air flow rate

Typical Cocurrent Air-water flow pattern (dia=32 mm) in vertical pipe - Roumy, 1969
For Air-water flows, typically seen flow patterns in horizontal flow: (Adiabatic flow)

- Bubble
- Slug
- Plug
- Annular
- Stratified
- Disperse
- Wavy

*Typical Cocurrent Air-water flow pattern in horizontal 5.1 cm diameter pipe—Weisman, 1983*
For steam–water flows, typically seen flow patterns **vertical heated boiler tube** as steam generation evolves: (Diabatic flow)

*For steam–water flows, typically seen flow pattern in **horizontal heated boiler tube** (Diabatic flow)*

Models for predicting flow patterns

- Knowledge of flow pattern is essential to know the flow regime of appropriate gas-liquid configuration
- This is importance as the magnitude of pressure drop & nature of heat transfer directly depend on this.

Operational Variables: \( \dot{m}_f \) & \( \dot{m}_g \)  
Geometric Variables: \( d \) (say pipeφ) & \( \theta \) (inclination)

Known fluid properties: \( \rho_1, \rho_g, \mu_1, \mu_g, \sigma \)

Co-ordinates for flow pattern map: 
Though many researchers have come out with many co-ordinates, Superficial phase velocities (or) momentum fluxes are commonly found to be used.

Some early popular flow pattern maps:
- Baker (1952)
- Hewitt & Roberts (1969)
- Taitel & Duckler (1976)- Horizontal flow
- Taitel, Bornea & Duckler (1980)- Vertical flow
Mechanistic flow pattern map
(2Φ-vertical upward flow through tubes)

Ref:

Seems to work reasonably well!

**Transition criterion A** (Bubbly to slug transition)

Presence of bubbly flow is postulated only above 25 mm diameter
The basic idea in deriving this criterion is to consider the velocities of small bubbles and large Taylor bubble. The typical criterion is

This is constant for a pipe diameter;
**note**: \(u_s = u_g - u_l\) is the slip /bubble rise velocity

\[
 u_s = 1.53 \left[ \frac{g(\rho_l - \rho_g)\sigma_l}{\rho_l^2} \right]^{1/4}
\]

\(u_{tb} = 0.35\sqrt{gd}\)

\[
 j_g = 0.25 u_s + 0.333 j_l
\]

\[
 u_s = 1.53 \left[ \frac{g(\rho_l - \rho_g)\sigma_l}{\rho_l^2} \right]^{1/4}
\]

\(\alpha_{crit} = 0.35\)

Note for \(u_s > u_{tb}\), the small bubbles easily catch up and coalesce and we see the transition to slug flow
Mechanistic flow pattern map
(2\(\Phi\)-vertical upward flow through tubes)

Let us look at one more criteria as discussed in the paper,

**Transition criterion E** (churn to Annular flow transition)

The balance between the gravity force \(F_g\) and drag force \(F_d\) acting on the entrained liquid droplets (of dia. \(d\)) is used as the criterion for transition.

\[
F_D = 0.5c_D \frac{\pi d^2}{4} \rho_g u_g^2 \\
F_G = \frac{\pi d^3}{6} g (\rho_l - \rho_g)
\]

For transition \(F_B > F_g\)

\[
j_g > 3.1 \left[ \frac{g \sigma_l (\rho_l - \rho_g)}{\rho_g^2} \right]^{1/4}
\]

One more reference you could rely on for vertical flows:

Flooding phenomenon

This represents a limit to counter-current flow where the gas phase is flowing upwards and liquid phase downwards. This is a kind of gross instability when fairly well defined flow rates are exceeded.

(typically occurs in slug flow regime)

Just before flooding

Bottle being emptied

Bottle being filled
(Submerged in a pool of water)

Interesting Flooding Situations

Gas flow rate increased to initiate flooding
Flooding phenomenon

Model given by Wallis (1961) seems to predict the occurrence fairly well.

\[
\left[ j_g^* \right]^{1/2} + \left[ j_l^* \right]^{1/2} = C
\]

\[ C = 0.88 (\text{smooth tubes}) \]
\[ C = 0.725 (\text{sharp ended tubes}) \]

Non-dimensionalised liquid & gas superficial velocities are

\[ j_g^* = \frac{j_g \rho_g^{1/2} \sim (\text{inertial})}{\left[ gD(\rho_l - \rho_g) \right]^{1/2} \sim (\text{gravity})} \]

Analogous to Froude number

\[ j_l^* = \frac{j_l \rho_l^{1/2}}{\left[ gD(\rho_l - \rho_g) \right]^{1/2}} \]
Boiling

Bubble generation, associated dynamics is the key to the augmented heat transfer characteristics. Hence understanding the nature and characteristics of the bubble on the hot surface is important.

Mechanism
- Nucleate boiling.
- Film boiling

Geometric
- pool boiling
- flow boiling

Heat could be added to water either through a heat flux controlled arrangement or through a temperature controlled arrangement.
Pool boiling

In a bowl of water that is heated – to a first appx. boiling can be said to occur when $T_w > T_{\text{sat}}$ & $T_b \leq T_{\text{sat}}$. The boiling with bubble generation that occurs when $T_b < T_{\text{sat}}$ is called subcooled boiling.

**Heterogeneous nucleation**

This is what we see normally on the heated solid surface.

Bubbles are not seen till the wall temp exceeds saturation temperature of the fluid.

**Homogeneous nucleation**

When water is heated on a extremely smooth surface (like a polished rotating dish of mercury), no physical nucleation sites are present on the surface.

A random process- highly energetic molecules coalesce to form very localised “vapour” region.

Water @ 1bar is seen to show nucleation only around 321ºC.
Bubble growth

The bubble growth is governed by

(i) In the initial phase - governed by surrounding liquid - Inertial controlled

(ii) In the later part – governed by Thermal diffusion through bubble boundary - Heat transfer controlled

Inertial controlled

Consider the condition of spherically symmetric 1 D flow

\[
\frac{1}{r^2} \frac{\partial}{\partial r} \left( r^2 V_r \right) = 0
\]

At \( r = R_B \), \( V_r = \frac{dR_B}{dt} = \dot{R}_B \)

\[ V_r = \frac{R_B^2 \dot{R}_B}{r^2} \]

Now from first law with no heat transfer, the liquid kinetic energy is equated to the work done.

\[
\dot{R}_B = \sqrt{\frac{2}{3} \frac{h_{fg} v_f}{T_{sat} v_g} \left( T_W - T_{sat} \left( P_{\infty} \right) \right)}
\]

This tells us bubble radius is proportional to time in this region.
**Bubble growth**

**Heat transfer controlled**

The latent heat of evaporation must be supplied by hot liquid. The infinite pool of liquid can be viewed as the semi-infinite region.

Energy balance gives

\[ \frac{q'' 4\pi R_B^2}{h_{fg}} = \frac{dm_g}{dt} = \frac{d}{dt} \frac{4}{3} \rho_g \pi r R_B^3 = 4 \pi \rho_g R_B^2 \frac{dR_B}{dt} \]

One can systematically arrive at

\[ R_B = 2Ja \sqrt{\frac{\alpha_t}{\pi}} + c \]

where

\[ Ja = \frac{\rho_l c_{pl}(T_s - T_i)}{h_{ve} \rho_v} \]

Using BC of \( R_B = 0 \) at \( t = 0 \), implies \( c = 0 \)

\[ R_B = 2Ja \sqrt{\frac{\alpha_t t}{\pi}} \]

\[ \Rightarrow \dot{R}_B = Ja \sqrt{\frac{\alpha_t}{\pi t}} \]

This tells us bubble radius is proportional to square root of time in this region.
Pool boiling curve

- Nukiyama’s experiment- 1934- heat flux controlled

Nukiyama’s heat flux controlled expt.
Leidenfrost Phenomenon

Typical nucleate boiling

Hemispherical liquid drop model at Leidenfrost condition

Using simple heat transfer and poiseuille flow consideration one can get predictive estimate of lifetime of the droplet as function of initial radius of bubble and fluid properties.
Convective boiling

- Systems are prone to micro (local) and macro (gross) instabilities
- Nucleation begins before the bulk fluid saturation temperature is reached - subcooled region
Convective boiling in tubes and channels

- Flow may be either through forced circulation or by natural circulation
- To maximise the heat transfer, we need to ensure the $2\Phi$ flow is in the nucleate boiling region - the flow need to be predominantly bubbly or slug

It is important that for convective boiling systems, we have predictive estimates of the way we pump heat in the system
Convective boiling in tubes and channels

Boiling considerations in constant heat flux controlled heated tube

Define

\[
(\Delta T_{sub})_i = (T_{sat} - T_{fi})
\]

\[
(\Delta T_{sat})_{ONB} = (T_{sat} - T_{sat})_{ONB}
\]

- Z is the co-ordinate along the heated length
- No heat loss is assumed

Simple energy balance would get us the expression track the thermodynamic quality

\[
x_e(z) = x_{in} + \frac{p_h}{GAh_{fg}} \int_0^z q''(z)dz
\]

Tracking wall temp.

\[
T_w = T_f(z) + \Delta T_{fw}
\]

\[
\Delta T_{fw} = \frac{q''}{h_{fo}}
\]

Tracking fluid temp.

\[
T_f(z) = T_{fi} + \left[ \frac{q''p_h}{mc_{pg}} \right] Z
\]
**Convective boiling in tubes and channels**

**Subcooled boiling**

Davis & Anderson - 1966

\[
\frac{T_w - T_{sat}}{T_w - T_f} \geq \frac{8\sigma T_{sat} v_{fg} h}{k_f h_{fg}}
\]

- For known heat flux

\[
T_f(z) = -\frac{q''}{h} + T_w
\]

- Limiting criterion would be

\[
T_f(z) = T_{sat} + \left[\sqrt{\frac{8\sigma T_{sat} v_{fg} q''}{k_i h_{fg}}} \right] - \frac{q''}{h}
\]

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Condensation occurs when $T_w < T_{sat}$ & $T_b >= T_{sat}$. When $T_b > T_{sat}$, the vapour must be cooled to $T_{sat}$ before liquid appears. This is referred to as heated condensation.

**Mechanism**
- Film condensation.
- Dropwise condensation

- What is desired is dropwise
- Practically the practical system end up operating in the film condensation mode.

*For a first level understanding, without non-condensable gases, Nusselt analysis would give an estimate of the film thickness and hence the heat transfer can be worked out*
Condensation inside tubes

The process of flow pattern changes as condensation proceeds is complex

- At point A wall is wet first with condensate
- At B liquid particles get torn off the liquid film into the superheated vapour
- From C onwards it goes through

Bakers map showing condensation path
Actual condensation

Actual condensation is different from what is different from Nusselt approximation as

- The film is never smooth and has effect in augmenting heat transfer.
- Shear effects make the film thinner and flow turbulence effects also augment heat transfer
- Multi-component effects (presence of non-condensibles)- This severely retards the heat transfer
- When non-condensible gases are present, the prediction of heat transfer and hence condensation is involved.
- Typical model requires the consideration of the flow configuration and for a known bulk non condensible fluid temperature and inlet mixture concentration, what is required is to work out the heat transfer through the non condensible vapour barrier.
Thank you for patient hearing