CHAPTER 3 THEORY

3.1 Heat recirculation combustion

Heat recirculating combustion is an effective technology not only for combustion of low calorific fuels but also for fuel conservation. Reactants are preheated prior to the flame zone by recycled heat from hot products. This results in superadiabatic combustion conditions (Budzianowski and Miller, 2009), i.e. the flame temperature can be substantially higher than that of simple non-recirculating adiabatic combustion. In superadiabatic combustion the flammability limits can be extended. Therefore, the heat recirculating combustion provides increased thermal efficiency and reduced harmful emissions. Moreover, recirculating combustion becomes more flexible in regard to calorific value of fuels.

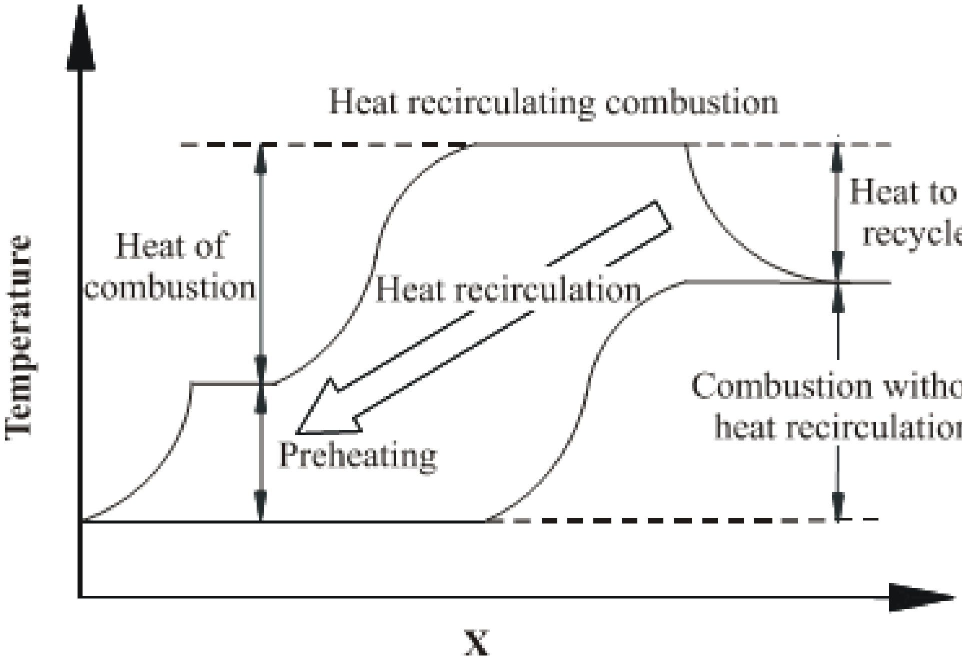


Figure 3.1 Schemetic temperature profiles in heat recirculation combustion of premixed reactants in one-dimensional adiabatic system.

Hardesty and Weinberg 1974, proposed the concept of “borrowed” enthalpy from the combustion products to preheat the combustible mixture. The schematic temperature profiles in heat recirculating combustion of premixed reactants in one-dimension adiabatic system are compared in Fig. 3.1 for the case with and without heat recirculation. In case of combustion without heat recirculation or normal flame (under line), the initial unburnt premixed reactants is ambient temperature. Combustion heat release raises the temperature to the adiabatic flame temperature, which depend on quality of mixture, thereafter remaining constant. While the initial reactants and flame temperatures in case of heat-recirculating combustion (upper line) are higher than normal flame. Because the additional enthalpy having been “borrowed” from the products are used to preheat reactant before entering to the reaction zone. The flame temperature in heat-recirculating combustion is determined by the amount of recycled heat that is independent of the equivalence ratio of mixture. Accordingly, it has been held that heat-recirculating combustion brings a temperature rise throughout combustion processes in proportion to the amount of recycled heat.

3.2 Internal heat recirculation combustion by porous burner

Combustion within porous media can be characterized as a heat-recirculating device in which the reactants or combustion air alone are preheated using heat “borrowed” from beyond the flame zone without mixing the two steams. Combustion systems of this kind which take advantage of heat recirculation are sometimes being referred as “excesses enthalpy,” “super-adiabatic flame temperature” or “filtration” combustion.

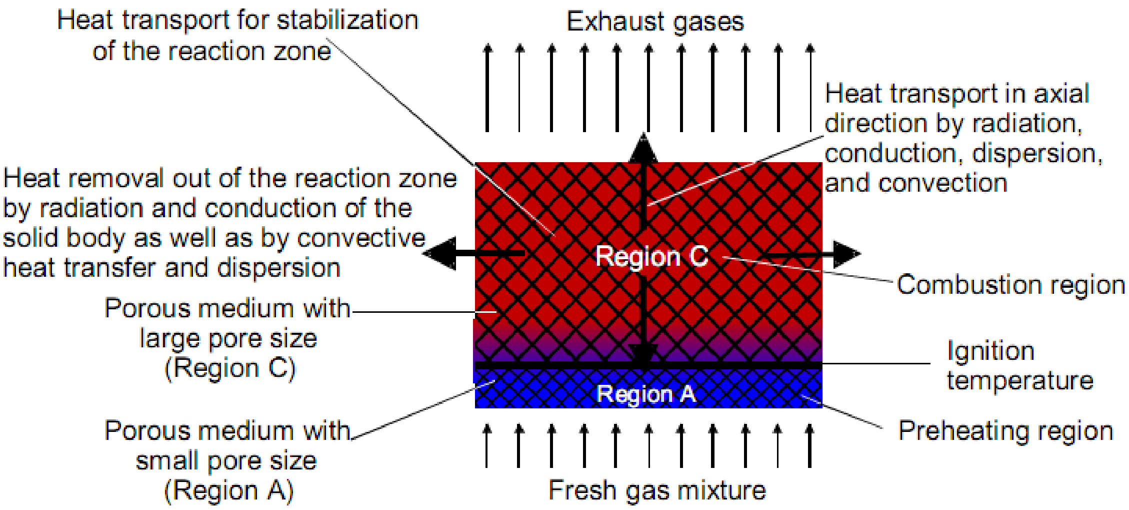


Figure 3.2 Heat transport in porous burner (Durst and Trimis 2002).

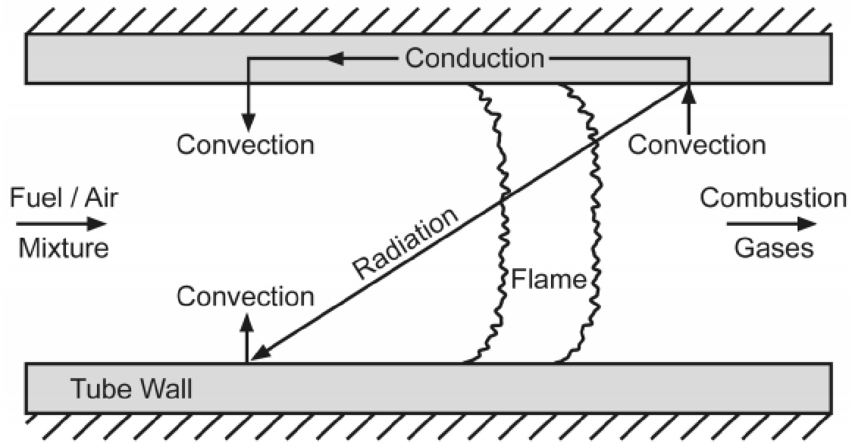


Figure 3.3 Schematic representation of heat recirculation in a porous medium idealized as an insulated refractory tube (Wood and Harris 2008).

The combined of conduction, convection and radiation heat transfer causes to heat recirculation in a porous medium show in Fig.3.2 and Fig. 3.3. In a porous burner, at post-flame zone, heat is transferred from the burned gases to the solid matrix by convection. Immediately, heat is recirculated by solid to solid conduction and radiation from the post-flame zone to the pre-flame zone. Thus, heat is transferred convectively from the hot solid to the incoming gas mixture at the pre-flame zone. This results to increasing flame temperature and burning velocity. In the other word, excess enthalpy or super-adiabatic combustion in the porous medium can be realized. The temperature profiles for the solid and gas of super-adiabatic combustion within porous media is shown in Fig.3.4. Moreover, many studies indicate that the porous burner can unlock the limitation of a general burner due to its providing a high radiant output with low hazardous emission (e.g. NO and CO). By the reason, in the last few decades, combustion in porous media has been the subject of a significant amount of research and development. There are numerous numerical studies for broaden knowledge in this area because the modeling has many advantages e.g. low cost to study and short time period requirement. The conservation equations for mass, gas energy, solid energy, and gas species were presented by Kaviany (1995) and Henneke and Ellzey (1999). In this thesis, the model has been adapted from that described in Henneke and Ellzey (1999) for combustion within porous medium. The problem considered in the present study is a one-dimensional reacting flow of constant-density fluids within porous media. To simplify the problem, the assumptions were used in the model were presented in section 4.1.2.

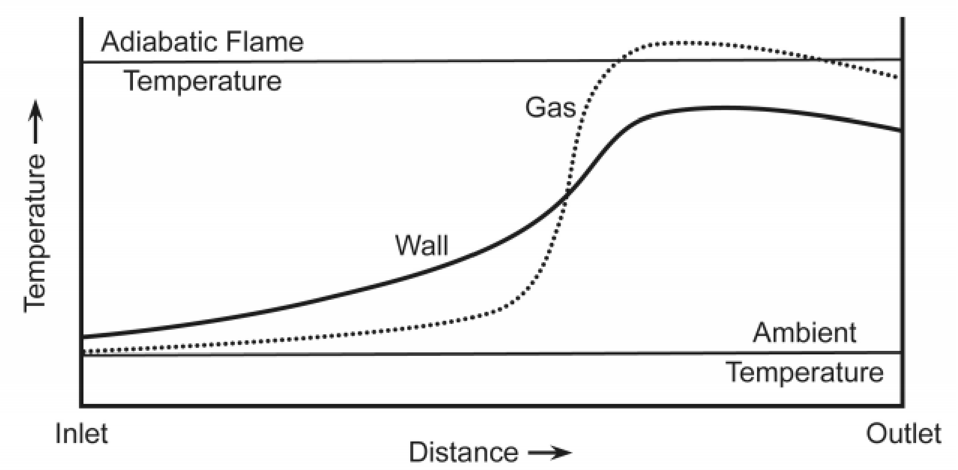


Figure 3.4 The corresponding variation in gas and tube wall temperature with distance (Wood and Harris 2008).

3.3 Boiling (Satish et al. 1999)

Heat transfer to boiling liquid is a convection process involving a change of phase from liquid to vapor. The phenomena of boiling heat transfer are considerably more complex than those of convection without phase change. There are two basic types of boiling: pool boiling and flow boiling. When a surface is exposed to a liquid, boiling may occur, and the heat flux will depend on the difference between the surface and the saturation temperature. When the heated surface is submerged below a free surface of liquid, the process is referred to as pool boiling. Pool boiling refers to boiling under natural convection conditions, whereas in forced flow boiling, liquid flow over the heater surface imposed by external means.

3.3.1 Pool boiling

Pool boiling occurs when a heater is submerged in a pool of initially stagnant liquid. When the surface temperature of the heater exceeds the saturation temperature of the liquid by a sufficient amount, vapor bubbles nucleate on the heater surface. The bubbles grow rapidly in the superheated liquid layer next to the surface until they depart and move out into the bulk liquid. While rising as the result of buoyancy, they either collapse or continue their growth, depending on whether the liquid is locally subcooled or superheated. If the temperature of the liquid is below the saturation temperature, the process is called subcooled, or local boiling. If the liquid is maintained at saturation temperature, the process is known as saturated or bulk boiling. Thus, in pool boiling, a complex fluid motion around the heater is initiated and maintained by the nucleation, growth, departure, and collapse of bubbles, and by natural convection. A thorough understanding of the process of boiling heat transfer in both pool boiling and flow boiling requires investigation of, first, the thermodynamics of the single bubble and, second, the hydrodynamics of the flow pattern resulting from many bubbles departing from a heated surface.

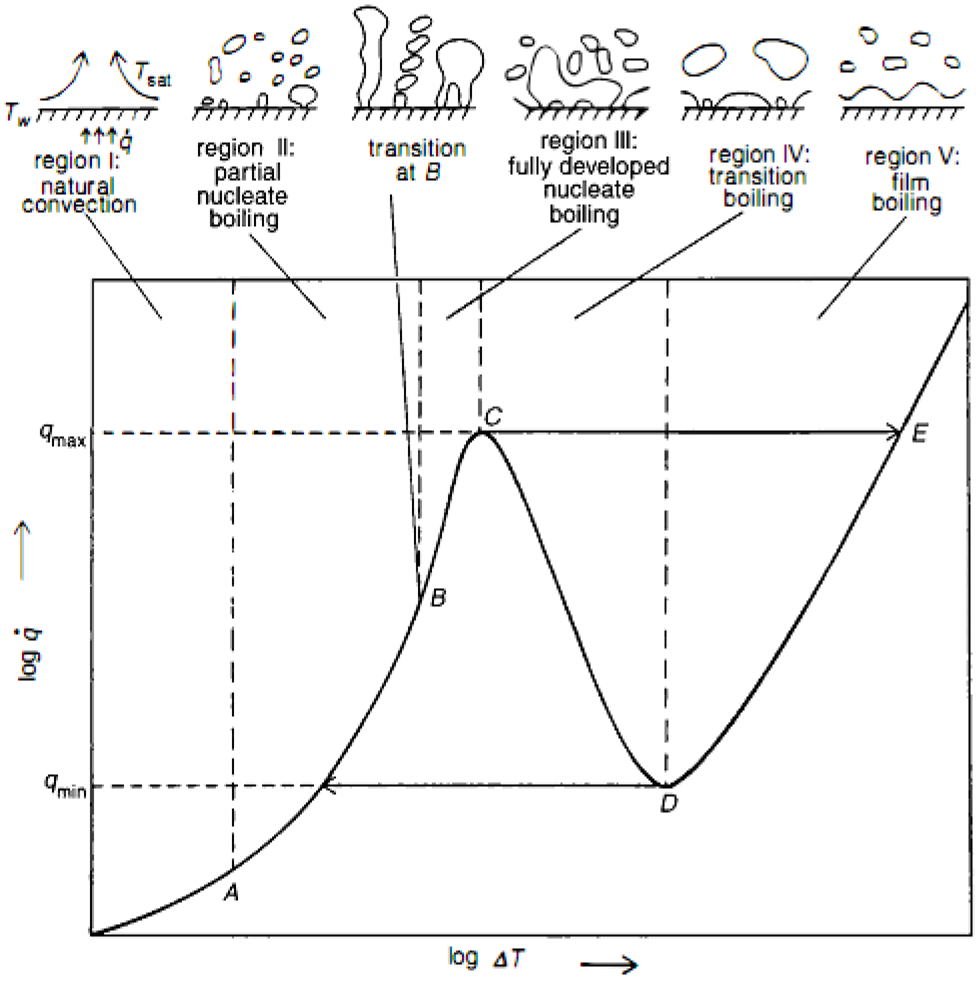


Figure 3.5 Typical boiling curves, showing qualitatively the dependence of the wall heat flux on the wall superheat ΔT, defined as the difference between the wall temperature T_w and the saturation temperature T_{sat} of the liquid. Schematic drawings show the boilimg process in regions I-V. These regions and the transition points A-E are discussed in the text.

Figure 3.5 shows, qualitatively, the boiling curve, i.e., dependence of the wall heat flux $\dot{q}$ on the wall superheat on a surface submerged in a pool of saturated liquid. The wall superheat, ΔT, is defined as the difference between the wall temperature and the saturation temperature of the liquid at the system pressure. The plotted curve is for a flat plate or a horizontal wire to which the heat input rate is controlled. As the heat-input rate to the surface is increased, the first mode of heat transfer to appear in a gravitational field is natural convection. At a certain value of the wall superheat (Point A), vapor bubbles appear on the heater surface. This is the onset of nucleate boiling. The bubbles from on cavities or scratches on the surface that contain pre-existing gas/vapor nuclei. In liquids that wet the surface well, the onset of nucleation may be delayed. For these liquids, a sudden activation of a large number of cavities at an increased wall superheat causes a reduction in the surface temperature while the heat flux remains constant. This behavior is not observed when the boiling curve is obtained by reducing the heat flux; thus, hysteresis results. After inception, a dramatic increase in the slope of the boiling curve is observed. In partial nucleate boiling, corresponding to region II (curve AB) in Figure 3.5, discrete bubbles are released from randomly located active sites on the heater surface. The density of active sites and the frequency of bubble release increase with heat flux or wall superheat. The transition from isolated bubbles to fully developed nucleate boiling (region III) occurs when bubbles at a given site begin to merge in the vertical direction. Vapor appears to leave the heater in the form of jets. The condition of the formation of jets also coincides (approximately) with the merger of vapor bubbles at the neighboring sites. After lateral merger, vapor structures appear like mushroom type bubbles with several stems (Gaertner, 1965).

3.3.2 Flow boiling

A boiling curve similar to that in pool boiling is obtained when flow occurs over the heater surface (external boiling) or inside a heated tube (internal boiling). In internal flow boiling, the vapor/liquid flow configuration change due to the addition of vapor along the flow direction. Figure 3.6 shows a typical forced flow boiling curve for a tube in which subcooled liquid enters the tube. The curve is drawn for a fixed total flow rate and system pressure. The first mode of heat transfer as subcooled liquid enters the tube is forced convection. The magnitude of the heat transfer coefficient increases with flow velocity. Since the heat transfer coefficient depends very weakly on the wall superheat in single phase forced convection, the heat flux linearly with wall superheat. After inception, a marked improvement in heat transfer coefficient occurs as wall superheat is increased. The region A′B′, spanning wall superheats from inception to fully developed nucleate boiling, is termed partial nucleate boiling. In this region, bubbles are formed discretely on the heated surface, and both single phase convection and phase change heat transfer contribute to the total heat transfer rate. As such, the functional dependence of heat flux on wall superheat is stronger than that for forced convection, but it is weaker than that for fully developed nucleate boiling.

Fully developed nucleate boiling curves (B′C′) at different velocityes generally overlap each other, suggesting the dominance of the heat removal by phase change. In fully developed nucleate boiling; bubble merger occurs at the heated surface. However, the phasic structure is influenced by the flow regime that exists in the bulk. Possible flow regimes are bubbley, slug, and annular. In annular flows, when liquid film on the wall becomes very thin, nucleate boiling may be suppressed. Now, the heat removal is through evaporation at the liquid vapor interface of the thin film. The heat transfer coefficient in very thin film annular flow may exceed that given by extension of the pool boiling curve. As a result, for a fixed heat flux, the wall superheat in very thin film flow may become smaller than that for fully developed nucleate boiling.

Under low flow/heat flux conditions, the critical heat flux occurs when the thin liquid film in annular flow dries out. Such heat flux is called dryout heat flux, and this condition generally occurs in steam generators. However, under high flow/heat flux conditions, the critical heat flux occur under conditions similar to those for pool boiling, i.e., sufficient liquid is available away from the wall or in the bulk. The critical heat flux condition results when vapor removal rate from the heater surface falls short of the vapor generation rate or liquid at the heater surface is not replenished fast enough to compensate for the evaporation rate. In the literature, critical heat flux condition has also been referred to as a departure from nucleate boiling, boiling crisis, and burnout heat flux. The magnitude of the critical heat flux depends on the local quality, mass velocity, and system pressure. For a surface subjected to axially uniform heat flux, the critical heat flux condition will first occur at the exit where the local quality is the highest. After occurrence of the critical heat flux, the temperature (for a heat flux controlled surface) raises very rapidly, and after passing through regions C′D′ and D′E′, the surface settles down in film boiling at E′. As for pool boiling, in forced flow a vapor film separates the heater surface from the liquid core. This flow configuration is called inverse annual flow boiling. If the liquid film dries (dryout heat flux) the surface will be cooled by forced flow of vapor. Some liquid droplets may be entrained in the vapor, and single phase forced convection correlations modified to account for the presence of droplets can be applied to determine the post dryout wall temperature.

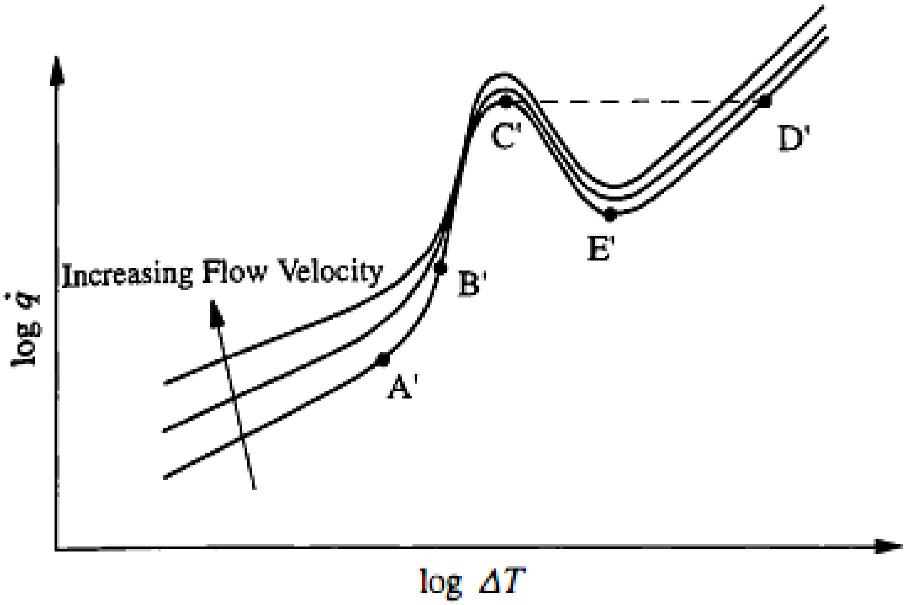
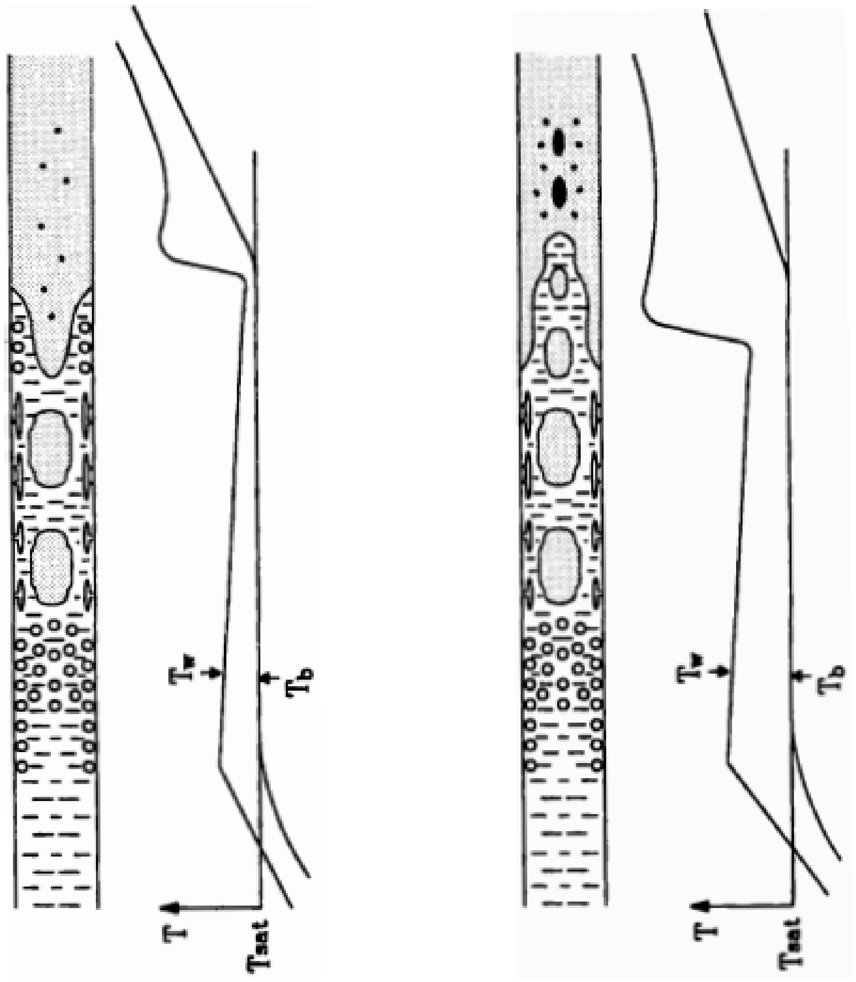


Figure 3.6 Flow boiling curve.

In the transition boiling region (C′D′), partial contact of the liquid with the solid occurs. As in pool boiling, this mode of boiling is unstable, and it is difficult to obtain steady state transition boiling data on a heat flux controlled surface. If the dryout front in a low flow/low heat flux case oscillates, te surface in the vicinity of the dryout front will be alternately wet and dry. As such, large variation in temperature will occur as the surface transitions from totally wet to totally dry conditions. Figure 3.7 shows, qualitatively, the flow regimes and variations in tube wall temperature for low flow/low heat flux and high flow/high heat flux conditions.



Low Heat Flux/Low Flow Rate   High Heat Flux/High Flow Rate

Figure 3.7 Flow regimes during flow boiling in a tube.

3.4 Evaporation within porous medium

Evaporation technology was reported by Paul E.M. (1986). The nucleate boiling heat transfer can be increased by artificially roughening the surface with techniques such as scratching, sand blasting, and etching. Fluted boiling surfaces have also been used. Boiling on fluted tubes is a thin-film process that takes place on each crest. The performance of fluted boiling surfaces is relatively poor when compared to the results achieved when condensing. The reason for this not clear but is probably due to the inability of maintaining thin films on the crests of the flutes. The crests may run dry; they may be flooded by liquid displaced from the troughs by axial flow of the accumulated vapor; or nucleate boiling may occur in the troughs.

An important technique to increase artificial rough surface is Linde surface. In this technique, the artificial rough surface can be created by using porous material. The Linde surface consists of a thin layer of porous metal bonded to the heat transfer substrate (Figure 3.8). Bubble nucleation and growth are promoted within a porous layer that provides a large number of stable nucleation sites of a predesigned shape and size. Microscopic vapor nuclei in the form of bubbles entrapped on the heat transfer surface must exist in order for nucleate boiling to occur. Surface tension at the vapor-liquid interface of the bubbles exerts a pressure above that of the liquid. This excess pressure requires that the liquid be superheated in order for the bubble to exist and grow. The porous surface substantially reduces the superheat required to generate vapor. The entrances to the many nucleation sites are restricted in order to retain part of the vapor in the form of a bubble and to prevent flooding of the site when liquid replaces the escaping bubble. Many individual sites are also interconnected so that fresh liquid is continually supplied.

Boiling heat-transfer coefficients, which are functions of pore size, fluid properties, and heat flux, are 10 to 50 times greater than smooth-surface values at a given temperature difference. Boiling coefficients are stable and relatively independent of convection effects. Nucleate boiling exists at much lower temperature difference than required for smooth surfaces.

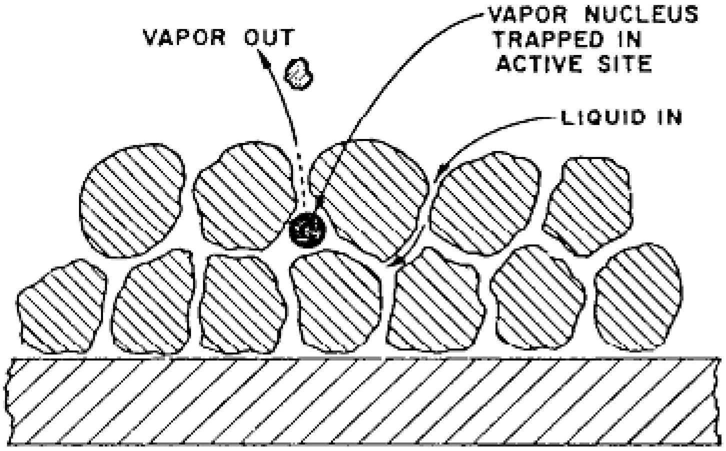


Figure 3.8 Linde porous boiling surface-mechanism of operation.

The Linde surface can be used in conjunction with fluted tubes or other enhanced surfaces to greatly increase heat transfer rates. Therefore, heat transfers with phase change in porous media arise in a number of scientific and engineering disciplines.

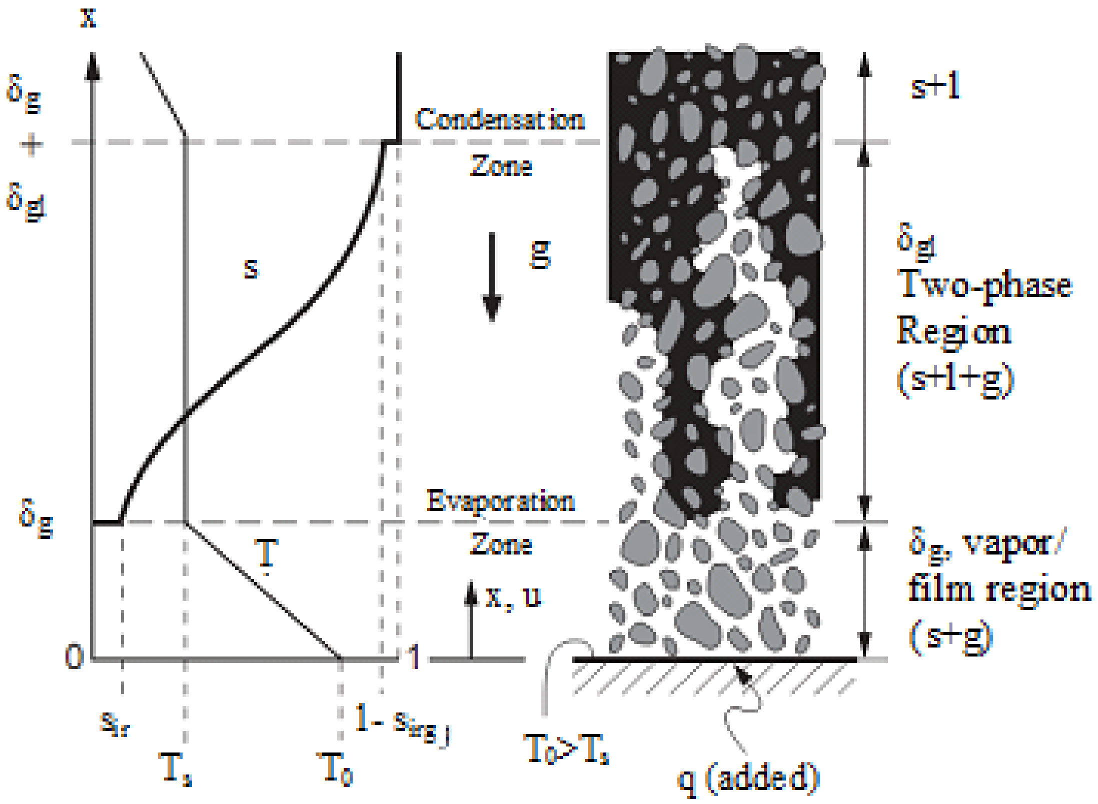


Figure 3.9 Evaporation due to the heat addition from below at temperatures above the saturation. The vapor-film region, the two-phase region, and the liquid region, as well as the evaporation and condensation zones are shown. Also shown are the distributions of temperature and saturation within these regions (M. Kaviany 1995).

M. Kaviany (1995) proposed the one-dimensional model for evaporation in porous media with heat addition q from the impermeable lower bounding surface maintained at $T_0 > T_s$, where T_s is the saturation temperature. The descriptive system and the distributions of temperature are shown in figure 3.9. The vapor-film region has a thickness δ_g , and the two-phase region has a length δ_{gl} . For $\delta_g \le x \le \delta_g + \delta_{gl}$□, the saturation is expected to increase monotonically with x. The vapor generated at the evaporation zone (the thickness of this zone is in practicefinite bu t here taken as zero) at $x = \delta_g$, moves up-ward (buoyancy-driven), condenses (condensation occurs in the condensation zone which is taken to have zero thickness) at x = $\delta_g + \delta_{lg}$, and returns as liquid (buoyancy- and capillary-driven). By allowing for irreducible saturations s_{ir} and s_{irg}, i.e., assuring continuous phase distributions for the two-phase flow, we have to assume an evaporation zone just below x = δ_g in which s undergoes a step change and evaporation occurs. A similar zone is assumed to exist above x = $\delta_g + \delta_{gl}$ over which s undergoes another step change and condensation occurs (condensation zone). Next, a case with $\delta_g > d$ and $\delta_{gl} >> d$ is considered, where the volume-averaged governing equations based on bulk properties can be applied. For $s < 1$, the liquid will be in a superheated state depending on the local radius of curvature of the meniscus. Therefore, the two-phase region is only approximately isothermal. For steady-state conditions, the heat supplied q is removed from the upper single-phase (liquid) region. Since the heat supplied to the liquid region causes an unstable stratification, natural convection can occur that can influence the two-phase region.

In this thesis, the governing equation of evaporation model that was proposed by M. Kaviany (1995) was modified to solve problem. Assumptions using in the model were shown in section 5.1.2 to simplify the problem.