MODELING AND DESIGN OF MICRO GROOVE FALLING FILM EVAPORATORS

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ABSTRACT

In the present study, heat transfer in a falling film micro groove evaporator has been simulated by an analytical model. The flow and thermal fields were divided in two regions, i.e. macroscopic flow inside the groove and the microscopic flow where intensive evaporation takes place at the thin film interline region. For the micro region model, pressure in the liquid film was expressed as a sum of surface tension and disjoining pressure effects. The film thickness profile was obtained by solving the 4th order differential equation by Runge-Kutta method. Then, this micro region model was combined with the macro region model. Macro region model solves one dimensional bulk flow inside the groove with gravitational effect taken into account. Constant curvature of the liquid vapor surface was assumed for the macro flow. It is shown that the gravitational force is essential for providing the liquid to wide range of heat transfer area. Thus, diverging branch evaporator is investigated. It is demonstrated that this concept has large potentiality for improving the performance of the micro groove falling film evaporator.

INTRODUCTION

Since thin liquid films can promote very rapid and intensive heat and mass transfer, falling film systems have been attracting large attention in the area of extraction, absorption, phase change and exothermic processes. With a micro-grooved surface, falling film systems can generate thin liquid films with thickness less than 100 µm on a solid wall. Thus, falling film system is expected to be one of the key technologies for developing micro or mini heat exchangers and reactors. Figure 1 shows the schematic view of the falling film evaporator with triangular grooves. In case of micro groove evaporator, it is known that intensive evaporation can be achieved at the interline region where very thin liquid film is formed adjacent to the adsorbed film (see Wayner, 1997, 1999). However, this strong evaporation occurs at a very limited area. For example, the length scale of this strong evaporation region is in a submicron range. Therefore, the main challenge in
developing a compact micro groove evaporator is to spotlight and emerge such microscopic strong evaporation in a macroscopic heat transfer surface. It is expected that this can be achieved by optimizing the width, length, path patterns and directions of the micro grooves on the surface.

In the present study, evaporation in a grooved surface is simulated analytically extending the method proposed by previous researchers, e.g. Wayner, et al. (1976, 1982), Kamotani (1978) and Stephan and Busse (1992) etc. In the following sections, the model equations for the micro and macro regions are presented. Finally, the model is applied to micro groove falling film evaporators. A concept that can improve the performance of the falling film evaporator, i.e. diverging branch evaporator, is investigated.

MODELING OF FALLING FILM EVAPORATOR

Micro Region Model

In the present study, the analytical model described by Stephan and Busse (1992) is adopted for the micro region. Since Reynolds number is very small and the surface tension becomes dominant in this scale, gravity is ignored in the micro model. Figure 2 shows the meniscus of the liquid captured in a triangular groove, and Fig. 3 shows the enlarged view of the micro region. In this study, right triangular shape is adopted for the cross section of the groove because of the simplicity of analytical treatment. Intensive evaporation takes place at the so-called interline region where liquid film gets very thin. The heat flux \( q \) in the liquid film is assumed to be one dimensional in the wall normal direction and can be written as

\[
\dot{q} = (T_w - T_{iv}) \left[ \frac{\delta}{\lambda_i} + \frac{T_{sat} \sqrt{2\pi R T_{sat}}}{h_f \sigma} \right] (2 - f) \]

where the second term in the denominator corresponds to the interfacial heat resistance. The evaporation coefficient \( f \) was set as \( f = 1 \). Constant wall temperature \( T_w \) is assumed in this study. The temperature at the vapor side of the interface \( T_{iv} \) is related to the saturation temperature \( T_{sat} \) by the following relation;

\[
T_{iv} = T_{sat} \left( 1 + \frac{p_v}{h_f \rho_l} \right)
\]

where \( p_v \) is the pressure difference between vapor and liquid phases;

\[
p_v = p_v - p_l = K \sigma + \frac{A}{\delta}.
\]

The first term describes the surface tension effect and the second term corresponds to the disjoining pressure effect. The curvature \( K \) can be expressed using film thickness \( \delta \) as

\[
K = \frac{d^2 \delta}{d \xi^2} \left( 1 + \left( \frac{d \delta}{d \xi} \right)^2 \right)^{3/2}.
\]

This pressure distribution due to the film thickness profile pulls the liquid to the strong evaporating interline region until it balances with the viscous drag of the flow.

The local and mean velocity profiles \( u \) and \( \bar{u} \) in a laminar boundary layer are given as

\[
u = \frac{1}{\mu} \frac{dp_v}{d \xi} \left( \frac{\delta^2}{2} - \delta \frac{d \delta}{d \xi} \right),
\]

\[
\bar{u} = \frac{1}{\delta} \int_0^\delta u d \xi - \frac{\delta^2}{3 \nu \rho_l} \frac{dp_v}{d \xi}.
\]

Then the mass flow rate \( \dot{M} \) can be obtained using Eq. (6);

\[
\dot{M} = -\frac{\delta^3}{3 \nu \rho_l} \frac{dp_v}{d \xi}.
\]

Thus, evaporation rate \( m \) is obtained as follows:


\[ \dot{\delta} = \frac{d\delta}{d\xi} = \frac{1}{3\nu_l} \frac{d}{d\xi} \left( \delta^3 \frac{dp_l}{d\xi} \right) \]  

(8)

Assuming that vapor pressure \( p_v \) is constant, the liquid phase pressure gradient is related to the capillary pressure gradient as

\[ \frac{dp_l}{d\xi} = \frac{dp_v}{d\xi}. \]  

(9)

Thus, one obtains the expression for the heat flux as

\[ \dot{\lambda} = \frac{h_f}{3\nu_l} \frac{d}{d\xi} \left( \delta \frac{dp_l}{d\xi} \right). \]  

(10)

Combining Eqs. (1), (2), (3), (4) and (10), the fourth order differential equation for the liquid thickness \( \delta \) is obtained:

\[
\frac{\sigma}{3\mu_l} \frac{d}{d\xi} \left( \delta^3 \frac{d}{d\xi} \left( \frac{d^2 \delta}{d\xi^2} \right) \right) = \frac{\frac{d^2 \delta}{d\xi^2} \left( \frac{d^2 \delta}{d\xi^2} \right)}{1 + \left( \frac{d\delta}{d\xi} \right)^2} + \frac{A}{\delta \sigma \delta^3}.
\]

\[ T_{sat} - T_w + \frac{T_w \sigma}{h_f \rho_i} \left[ \frac{\frac{d^2 \delta}{d\xi^2} \left( \frac{d^2 \delta}{d\xi^2} \right)}{1 + \left( \frac{d\delta}{d\xi} \right)^2} + \frac{A}{\sigma \delta^3} \right] = \frac{\rho h_f \left[ \frac{\delta}{\lambda_l} + \frac{\frac{d\delta}{d\xi}}{\lambda} \right] + \frac{\frac{2nR_l T_{sat}}{h_f \rho_i} \frac{2 - f}{2f}}}{\frac{2nR_l T_{sat}}{h_f \rho_i} \frac{2 - f}{2f}}. \]  

(11)

In order to solve Eq. (11), following four variables are introduced:

\[ z_1 = \delta, \]

(12)

\[ z_2 = \frac{d\delta}{d\xi}, \]

(13)

\[ z_3 = \frac{d^2 \delta}{d\xi^2} \left( \frac{d^2 \delta}{d\xi^2} \right) + \frac{A}{\sigma \delta^3}, \]

(14)

\[ z_4 = \frac{\sigma}{3\mu_l} \frac{d}{d\xi} \left( \delta^3 \frac{d}{d\xi} \left( \frac{d^2 \delta}{d\xi^2} \right) \right) \left( \frac{d^2 \delta}{d\xi^2} \right) + \frac{A}{\sigma \delta^3}. \]

(15)

where variables \( z_1, z_2, z_3 \) and \( z_4 \) correspond to film thickness, film slope, capillary pressure and flow rate, respectively. Then Eq. (11) can be rewritten as a system of four first order differential equations as follows:

\[ \frac{dz_1}{d\xi} = z_2, \]

(16)

\[ \frac{dz_2}{d\xi} = \left( z_3 - \frac{A}{\sigma \delta^3} \right)^{1/2}, \]

(17)

\[ \frac{dz_3}{d\xi} = \frac{3\mu_l}{\sigma} z_4, \]

(18)

\[ \frac{dz_4}{d\xi} = \frac{T_{sat} - T_w + \frac{T_w \sigma}{h_f \rho_i} z_3}{\rho h_f \left[ \delta + \frac{\frac{2nR_l T_{sat}}{h_f \rho_i} \frac{2 - f}{2f}}{2} \right]} \text{.} \]

(19)

This set of equations is integrated by fourth order Runge-Kutta method. The boundary conditions are shown in Fig. 4. At the macro and micro boundary, the film thickness gradient \( z_{2in} \), and \( z_{3in} (=p_v/\sigma) \) are set as

\[ z_{2in} = \tan \theta, \]

(20)

\[ z_{3in} = K = 1/R_m. \]  

(21)

In the present study, the slope of the vapor liquid interface at the macro-micro boundary is chosen as \( \theta = 30 \) degrees. It is confirmed from the preliminary calculations for \( \theta = 25 \text{–} 45 \) degrees that this condition doesn’t affect the film thickness and the heat flux prediction at the interline region near the adsorbed thin film boundary. The film thickness \( z_{2in} \) and flow rate \( z_{4in} \) at the macro-micro boundary are chosen by iteration so that both film slope \( z_{3end} \) and flow rate \( z_{4end} \) become zero at the micro and adsorbed film boundary, i.e.

![Fig. 4 Boundary conditions](image-url)
Fluid Flow

\[
\Delta p_{\text{surface tension}} = \Delta p_{\text{viscous}} + \Delta p_{\text{gravity}},
\]

where viscous drag \( \Delta p_{\text{viscous}} \) is given by the predictive method for stratified two phase pipe flows described by Taitel and Duckler (1976);
\[
\Delta p_{\text{viscous}} = 4f \frac{\Delta x}{D} \frac{\rho}{2} u_n^2, 
\]
where \( u_n \) is the mean liquid velocity inside the groove. For the friction factor,

\[
f = \frac{24}{Re_D} 
\]
is recommended for low Reynolds number laminar flows by Spedding and Hand (1997). The hydraulic diameter \( D \) is defined as

\[
D = \frac{4(A_{\text{macro}} + A_{\text{micro}})}{S_{\text{macro}} + S_{\text{micro}}}, \tag{27}
\]
where \( S \) is the perimeter of the wet wall and \( A \) is the cross section of the liquid flow, as shown in Fig. 8.

Thus, pressure difference due to the surface tension force can be written as

\[
\Delta p_{\text{surface}} = \frac{96}{Re_D} \frac{\Delta x}{D} \frac{\rho}{2} u_n^2 - \rho g \Delta x \sin \phi, \tag{28}
\]
where \( \phi \) is the angle of inclination of the surface respect to the horizontal plane (see Fig. 1).

Surface tension pressure difference \( \Delta p_{\text{surface}} \) can be also expressed by the curvature radius \( R \) as

\[
\Delta p_{\text{surface}} = \frac{\sigma}{R(x)} - \frac{\sigma}{R(x + \Delta x)}. \tag{29}
\]

Thus, the curvature radius at the neighboring cross section \( R(x + \Delta x) \) can be obtained from Eqs. (27) and (28) as

\[
R(x + \Delta x) = \left( \frac{1}{R(x)} - \frac{96}{Re_D} \frac{\Delta x}{D} \frac{\rho}{2} u_n^2 - \rho g \Delta x \sin \phi}{\sigma} \right)^{-1}. \tag{30}
\]

If the initial value of meniscus radius \( R(0) \) is given at \( x=0 \), evaporation rate \( \dot{g}(0) \), film thickness \( \delta_f(0) \) at the macro-micro boundary, cross sectional area and perimeter length of the micro region, \( A_{\text{micro}} \) and \( S_{\text{micro}} \) can be obtained from the micro region calculation. Then, one obtains the curvature radius and the flow rate at the next control volume. This procedure is continued in sequence until the curvature and the flow rate reach to the inlet condition at the top of the evaporator. In the present study, curvature radius and bulk flow rate are set as \( R(0) = 4 \times 10^{-8} \) (m) and \( \dot{G}(0) = 0 \) (kg/m²s) at the origin \( x=0 \).

**PREDICTION OF FALLING FILM EVAPORATOR**

First, the gravitational effect is investigated by inclining the evaporator wall. Again R123 is adopted with \( T_w=303.15K \) and \( T_{sat}=298.15K \). Figure 9 shows the liquid length \( l \) inside the groove plotted against groove width \( w \). Liquid length \( l \) is defined as the length between the inlet where the groove is filled with inscribed meniscus and the point where the liquid dries out (see Fig. 10). When the surface is placed horizontally (\( \phi=0\deg \)), the grooves easily dry out. Surface tension pulls the liquid along the groove to some extent, but it is unsatisfactory for supplying enough liquid to wide area of heat transfer surface. This means that it is necessary to introduce a mechanism other than surface...
tension to pull the liquid sufficiently into the micro groove. As the heat transfer surface is further inclined and the groove gets upright, gravity effectively pulls down the flow and the liquid length is increased. Thus, it is confirmed that the gravitational force is essential for providing the liquid to large heat transfer area. As the groove width gets smaller \((w<200 \text{ } \mu\text{m})\), surface tension effect becomes dominant again and the liquid is consumed by strong evaporation. This large consumption of the liquid makes the liquid length very short even for large inclination angle cases.

Figure 11 shows the total heat evaporated from unit width of the falling film evaporator \((=\text{averaged heat flux } q \times \text{liquid length } L, \text{ W/m})\) plotted against groove width. The curves show monotone increase against groove width, which indicates that large grooves that can supply liquid longer is better than small grooves that easily dry out. Figure 12 is a sketch of the heat transfer surface with large grooves and small grooves. Since smaller grooves have many interline regions per area, they can produce higher local heat flux than larger grooves at wetted area. However, the present result reveals that it is better to supply liquid to wider area by large grooves to eliminate the dry out area, even though the local heat flux is deteriorated at the wetted area.

Figures 11 and 12 imply that maximum heat exchange can be achieved when the liquid just dries out at the bottom of the falling film evaporator. Figure 13 shows the averaged heat flux of the whole evaporator surface of length \(L\).

Groove width is optimized for each case so that the evaporator surface becomes all wet and the liquid just dries out at the bottom. The values of optimal groove width are also plotted in the figure. Optimal width gets smaller as the surface is inclined. Smaller length \(L\) and larger inclination angle \(\phi\) give larger heat flux. Furthermore, average heat flux becomes more insensitive to the inclination angle as length \(L\) gets smaller.

**DIVERGING BRANCH EVAPORATOR**

Previous results come up with the idea that two important functions, i.e. liquid supply by gravity and the liquid consumption by strong evaporation, should be individually realized in the micro groove falling film evaporator design. In relatively large grooves, gravitational force can effectively flush large amount of liquid, while surface tension force can strongly pull the liquid into the narrow grooves to the strong evaporating region. Thus, diverging branch evaporator is investigated. Bifurcated branching is adopted for simplicity in the present study. The liquid is first supplied into the wide primary groove in the gravitational direction, and then the fluid is pulled horizontally into the small second grooves and consumed by effective evaporation. The schematic of diverging branch system is shown in Fig. 14. At the connection point of the branches, same curvature radius was simply adopted in the calculation.

The groove width and length of the horizontal branch are fixed as \(w_2=50\mu\text{m}\) and \(L_2=314.8\mu\text{m}\), respectively. The primary groove width \(w_1\) is optimized so that the whole evaporator surface of length \(L_1\) gets wet. Figure 15 shows the averaged heat flux for diverging branch heat exchanger. It can be seen that diverging branch evaporator shows nearly twice as large heat flux than the standard groove evaporator of same length (see Fig. 13).
It must be noted that this result was obtained without any optimization for the small horizontal branch width $w_2$. Moreover, it is known that trapezoidal grooves will give better performance than triangular grooves. Different refrigerant with larger thermal conductivity will also drastically increase the heat flux. Thus there is certainly room for further improvement. It can be said that diverging branch evaporator has large potentiality for improving micro groove falling film evaporator performance.

CONCLUSIONS

1. Heat transfer in a falling film micro groove evaporator is simulated by an analytical model. The model described by Stephan and Busse (1992) is employed for the micro region, and it is combined with the macro region model that takes into account the gravitational, surface tension and viscous forces.

2. Gravity can effectively pull down the fluid inside the groove, and can drastically enhance the wetting of the surface. As the groove width gets smaller, surface tension gets dominant, and effective evaporation can be achieved.

3. Maximum heat exchange can be obtained when the liquid just dries out at the bottom of the falling film evaporator.

4. Diverging branch evaporator showed nearly twice as large heat flux than the standard groove evaporator of same length. It is concluded that diverging branch evaporator has large potentiality for improving the performance of the micro groove falling film evaporator.

NOMENCLATURE

$A$ dispersion constant, J, or area, m$^2$
$D$ hydraulic diameter, m
$f$ evaporation coefficient, dimensionless, or friction factor, dimensionless
$G$ macro region mass flow rate, kg/m$^2$s
$g$ macro region evaporation rate, kg/m$^2$s
$hg$ specific heat of evaporation, J/kg
$K$ curvature of the meniscus, 1/m
$L$ evaporator length, m
$l$ liquid length inside the groove, m
$M$ micro region mass flow rate, kg/m$^2$s
$m$ micro region evaporation rate, kg/m$^2$s
$p$ pressure, Pa
$q$ heat flux, W/m$^2$
$R$ curvature radius, m
$S$ perimeter, m
$T$ temperature, K
$w$ groove width, m
$x$ direction of the groove, m
$\delta$ film thickness, m
$\eta$ viscosity, Pa s,
$\phi$ inclination angle, rad
$\lambda$ thermal conductivity, W/mK
$\nu$ kinematic viscosity, m$^2$/s
$\theta$ film thickness gradient
at macro-micro boundary, rad
$\rho$ density, kg/m$^3$
$\sigma$ surface tension, N/m
$\xi$ flow direction in the micro region, m
$\zeta$ distance from the wall, m

Subscript

c capillary
$iv$ vapor side of the liquid-vapor interface
$l$ liquid
$m$ mean
$s$ solid
$sat$ saturation
$v$ vapor
1 primary branch
2 horizontal branch

REFERENCES


Wayner, P. C., Jr., Kao, Y. K. and LaCroix, L. V., 1976, The interline heat-transfer coefficient of an evaporating

