An analytic, slug–deposited liquid film thermal resistance for OHP

Abstract- A simple, analytic model for the thermal resistance of the oscillating (pulsating) heat pipe (OHP or PHP) is proposed and verified. It is based on the Taylor liquid film formed by moving liquid slugs, film heat conduction, and interfacial evaporation/condensation. It uses a unidirectional flow (characteristic of use of check valves) in a horizontal (no gravity effect) single-loop OHP to derive a relationship between the slug velocity and heat flux. The model predictions are verified with extended, prior direct 1-D, simultaneous heat, mass, and momentum simulation and with experiments. In addition, the CFD results for the slug-deposited liquid film thickness δ_l support that the transition from laminar flow (at Reynolds number, Re_D = 2000, D is the diameter) tends toward a constant film thickness regime, previously observed (experimentally) and correlated. The CFD and direct 1-D simulations also show that at yet higher ReD (3700), a constant slug velocity regime is reached. The evaporator specific thermal resistance, $A_eR_e = A_e/G_e = \delta_l/k_l$ (k_l is liquid thermal conductivity), and the total OHP thermal resistance R is predicted and compared with the current and available related experimental results (for R134a and butane) with good agreement. The suggested, high heat flux (constant film thickness $\delta_{l,c}$ limit for Re_D > 2000) dimensional total resistance/conductance is $R = \frac{1}{G} = \frac{\delta_{l,c}}{k_l}\left(\frac{1}{A_e}+\frac{1}{A_c}\right)$, with $\delta_{l,c} = 120D\left(\frac{\mu_l^2}{\rho_l\sigma D}\right)^{2/3}$, providing a lower limit on R (upper limit on G). This analytical model can be extended to non-circular channels by using the hydraulic diameter.

Motivation - The oscillating (pulsating) heat pipe (OHP or PHP) allows for passive and effective transport of heat through a closed pipe running between the heat source (evaporator section of OHP) and heat sink (condenser) with multiple turns with plugs of vapor and liquid traveling through them. While, generally, the wick heat pipe has a larger effective conductivity, the OHP is preferred over the wick heat pipe for spreading heat over large areas (and in low gravity applications). However, due to the complex (even chaotic) fluid dynamics present in the OHP compared to wick heat pipes, an OHP thermal resistance prediction method is not yet established. This will be undertaken here with assumptions supported by CFD.

By considering the dominant conduction mechanism through the wet wick in the heat pipe and the liquid film in OHP, a direct comparison of the specific internal thermal resistance (a performance measure) of the conventional heat pipe and OHP can be made. In a wick heat pipe, the internal thermal resistance is dominated by the heat conduction across the thin wick δ_w. The specific thermal resistance can be defined as AR = $\delta_w/\langle k\rangle$, where $\langle k\rangle$ is the effective thermal conductivity and is proportional to k_w/δ_w with k_w ~10 W/m-K and δ_w ~ 1 mm. In contrast, the internal heat transfer in the OHP has been recently associated with the moving liquid slug thin liquid film deposit, δ_l (Fig. 1) allowing a specific thermal resistance defined as AR = $\delta_l/\langle k\rangle$, where $\langle k\rangle$ is proportional to k_l/δ_l with k_l ~ 0.1 - 0.6 W/m-K and $\delta_w \sim 30-40\ \mu$m.

Model - The evaporator section of a single branch in a single-loop OHP is covered by a film of constant thickness. If this film is not replenished through the passing of a slug, it will completely evaporate due to heat load. To ensure perfect replenishment of the evaporating film, a slug of length $L_{s,o}$ moving at a constant speed must pass such that the entire mass contained by the slug balances the mass removed from the film through evaporation (Fig. 2). The energy balance for this threshold scenario is,

$$Q = \frac{\rho_l A_c L_{s,o}\Delta h_{lg}}{L_e}u_l \quad \text{or} \quad q = \frac{\rho_l D L_{s,o}\Delta h_{lg}}{4L_e^2}u_l$$

where Q is the heat load, ρ_l is the liquid slug density, A_c is the pipe cross-sectional area, $L_{s,o}$ is the minimum slug length, u_l is the slug velocity, Δh_{lg} is the heat of evaporation of the liquid, and L_e is the evaporator length. An approximate minimum slug length for evaporation was determined to be $L_{s,o} \approx 1.75D$ using the spatial-temporal slug velocity with respect to the heat flux relationship established by the 1-D OHP simulation of [1]. At high-heat flux, the slug velocity becomes constant due to the transition to turbulence giving a constant friction factor.
The film thickness – slug velocity relationship has be previously correlated by [2]. At high-Re, a constant film thickness is observed and approximated by,

$$\delta_{l,c} \approx 120D\left(\frac{\mu_l^2}{\rho_l\sigma D}\right)^{\frac{2}{3}}.$$

Combining the film correlation with the simple model relationship between slug velocity and heat flux, a prediction of resistance can be made.

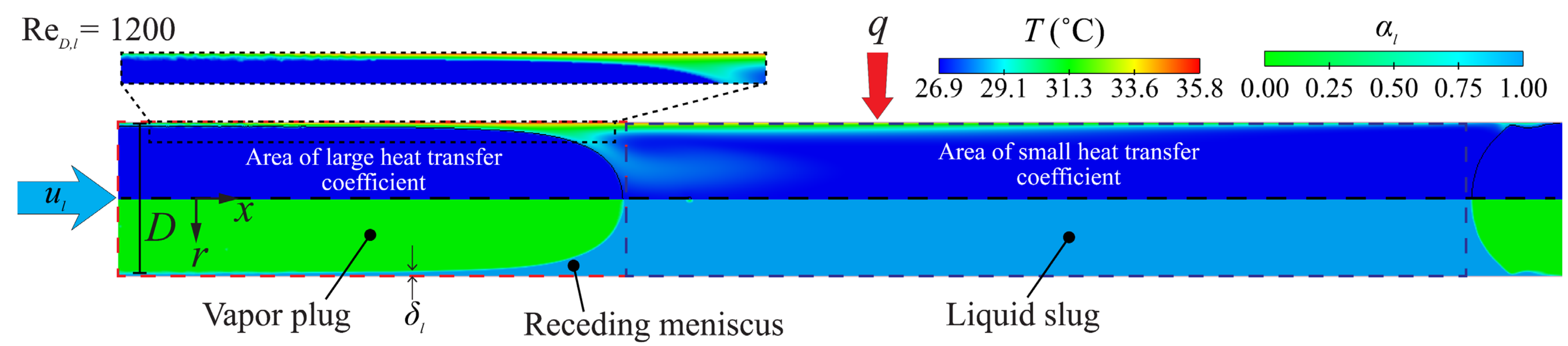

Figure 1. Liquid film formed from liquid slug moving in a circular capillary pipe with strong heat transfer by conduction occurring across this liquid film.

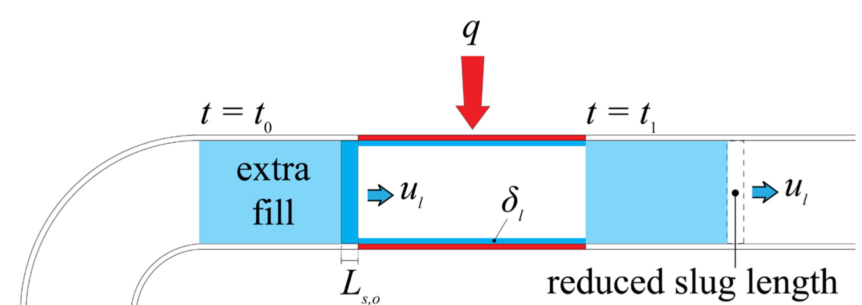

Figure 2. Physical motivation for the simple analytical model. The slug must lose mass (length) to perfectly replenish the film as it evaporates due to the applied heat load.

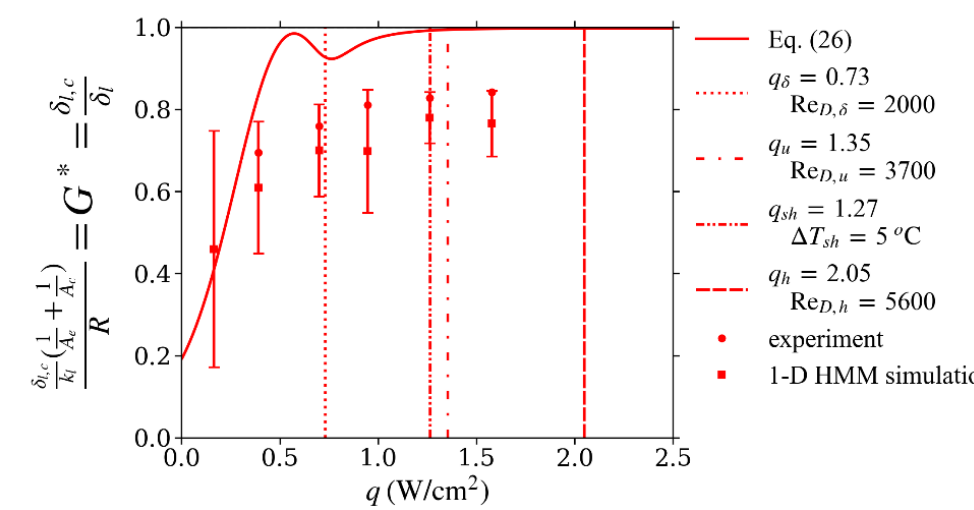

Figure 3: The variations of the dimensionless OHP thermal conductance with respect to heat flux, showing the measured and 1-D simulation predicted results [1], and liquid-film-based prediction of this study. The results are for R134a in a circular pipe of D = 0.8 mm, using check valves. Flooding effect is considered through an exponential decay.

Results - The variation of the predicted dimensionless OHP thermal conductance G^* is compared to the experimental and numerical simulation results of [1] (Fig 3). Important heat fluxes are illustrated as vertical broken lines including the heat flux corresponding to flooding (q_f), onset of constant film thickness (q_δ), onset of constant slug velocity (q_u), superheat (q_{sh}), and onset of film instability (q_h). The high heat flux experimentally determined conductance is G = 2.1 W/K (R = 0.48 K/W) while the model predicts a dimensional resistance of G = 2.5 W/K (R = 0.4 K/W) (a 17% overprediction). This underperformance of the experiment can be due to the thermal resistance through the materials, partial dryout in the evaporator, and liquid blockage in the condenser.

[1] Daimaru, T. et al., Comparison between Numerical Simulation and On-orbit Experiment of Oscillating Heat Pipes. Paper no. ICES-2016-185. Vienna, Austria, July 11, 2016.
[2] Youn, Y. J., Muramatsu, K., Han, Y., & Shikazono, N., The effect of bubble deceleration on the liquid film thickness in microtubes. International Journal of Heat and Fluid Flow, 58:84–92, 2016.