Condensation on Vertical Porous Metal Fins

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The problem of condensation of vapors on a porous plate fin is formulated and the dimensionless heat transfer coefficients are found to be dependent on two system variables, viz., the fin and the porosity parameters. Numerical simulations indicated that the condensation heat transfer coefficients for a porous fin are larger than those for a non-porous one of similar cross-section. For design purposes, a dimensionless equation for predicting the mean condensation heat transfer coefficient is proposed.

The significant increase in the condensation heat transfer coefficient suggests that the condenser section of a flat plate heat pipe can be made more efficient in operation with the use of porous plate fins instead of impermeable fins of similar cross-section.

Une formulation est proposée pour le problème de la condensation des vapeurs sur une ailette à plaque poreuse; on a trouvé que les coefficients de transfert de chaleur unidimensionnels étaient dépendants des deux variables du système, soit les paramètres de l'ailette et les paramètres de porosité. Des simulations numériques indiquent que les coefficients de transfert de chaleur de condensation pour une ailette poreuse sont plus grands que ceux d'une ailette non poreuse de même section. À des fins de conception, on propose une équation unidimensionnelle pour la prédiction des coefficients moyens de transfert de chaleur de condensation.

L'augmentation considérable du coefficient de transfert de chaleur de condensation laisse supposer que l'on peut améliorer l'erfficacité de la partie condenseur d'un caloduc plat en utilisant des ailettes à plaque poreuse plutôt que des ailettes imperméables de même section.

Keywords: condensation, plate porous fins, heat transfer augmentation, flat plate heat pipe.

After the pioneering studies by Nusselt (1916) on condensation of vapors on isothermal vertical surfaces, many improvements in the analysis have been undertaken in relaxing the assumptions employed by Nusselt, and condensation studies on several geometries have been made. The efforts of various investigators can be found in the review by Merte (1973). The recent trends in the design of thermal equipment are to aim at compactness and achieve better performance for the given system conditions. Hence, augmentation techniques classified under three categories, viz., active and passive or a combination of both, have been increasingly considered in an attempt to achieve a substantial increase in the condensation heat transfer coefficients. Bergles et al. (1982) list references pertaining to augmentation heat transfer, including condensation processes as well. Heat transfer studies on extended surfaces have major practical significance in affecting the compactness and overall performance. Patankar and Sparrow (1979) solved the problem of condensation on an extended surface attached to a vertical plate and cylinder. The results indicated the dependence of temperature distribution and condensate film thickness on both the longitudinal and transverse coordinates of the fin. Further, the velocity and temperature distributions of the condensate were shown to be three-dimensional in nature. The results also deviated from the predictions obtained from Nusselt's (1916) analysis. Although extended surface geometry is often encountered, reports of carefully controlled experimental studies on condensation on fins are not available in literature.

Honda and Fujii (1984) tackled the problem of condensation of flowing vapor on a horizontal tube by using a numerical approach and considering the process to be a conjugate heat transfer problem. Lu and Lii (1985) analytically investigated the condensation of saturated vapor on a vertical plate fin. In their analysis, the stream function related to the condensate flow field was considered to be a function of a single similarity variable, whereas the temperature field in the condensate was assumed to be dependent on two dimensionless variables. Sarma et al. (1987) considered condensation on a vertical non-porous fin of constant thickness for a wide range of system parameters. Subsequently, the case of condensation of vapors on fins of variable thickness was presented by Sarma et al. (1988). The results indicated improved condensation rates due to the non-isothermal nature of the fin.

The design aspects of heat pipes were discussed by Dunn and Reay (1978). According to them, the heat pipe performance can be enhanced by improvising techniques that would raise the condensate rate in the condenser section of the heat pipe. Hence, conceptually it is thought of that if the fin were made of sintered metal, the process of condensation on it could be made more efficient and the condensation heat transfer coefficients could be improved further, consequent to the decrease in condensate film thickness arising from a partial suction effect on the condensate film rolling down on the lateral face of the fin under the action of gravity. The present study can find its application in the development of the condenser section of a heat pipe to raise the limits of its operation. Such a problem has not been solved before.

Thus, the main objective of this investigation is to examine the possibility of achieving augmentation of the condensation heat transfer coefficients by using porous fins

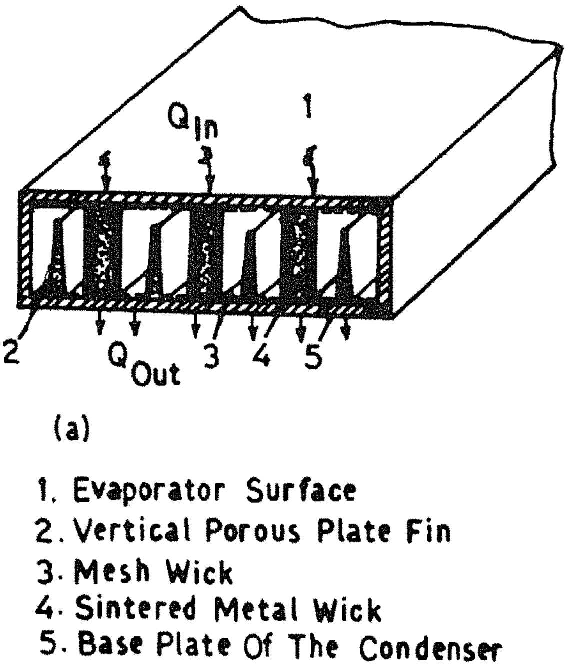

Figure 1(a) — Internal structure of heat pipe.

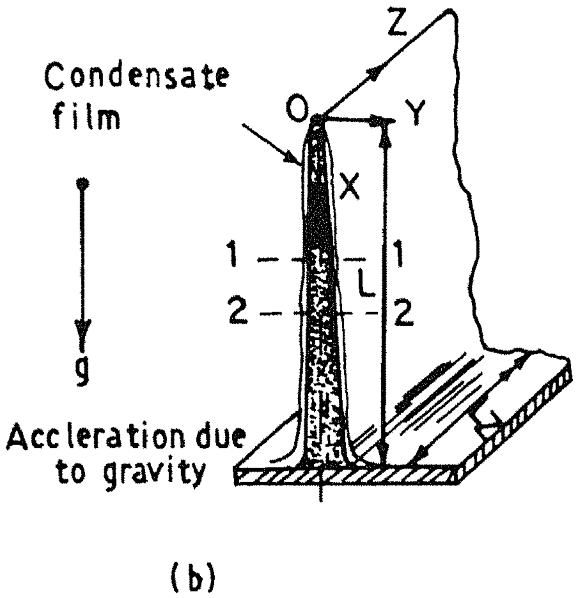

Figure 1(b)— Configuration and coordinate system of the porous fins.

of varying cross-section in the condenser section of a flat plate heat pipe.

Problem statement and formulation

A general view of the flat plate heat pipe is shown in Figure 1a. Porous plate fins of variable cross-section may be used as an integral part of the condenser section of a flat plate heat pipe. By orienting the plate fin as per the coordinate system shown in Figure 1b, the problem can be converted into a one-dimensional analysis in contrast to the earlier work of Patankar and Sparrow (1979). The condensate formed on the lateral surface of the porous fin is partly absorbed into the fin and the rest rolls down due to gravity along its vertical surface towards its base. A sintered metal wick spread at the base of the fin effectively absorbs the condensate. Further, the condensate is driven back to the evaporator section through the wick due to the action of capillary forces working against gravity. In the evaporation section, vaporisation is facilitated by the heat injected into the evaporator. The vapors thus generated get condensed on the porous plate fins situated in the condenser section. The condensate flow is partly through the pores and the rest flows as a thin film on the lateral face of the fin. The flow dynamics of the condensate is gravity controlled. As the pores get saturated with the condensate medium, a net flow of the condensate towards the base of the fin takes place. For the physical configuration and the coordinate system of the fin shown in Figure 1b, the equation of motion of the condensate is derived, assuming idealized potential flow of the condensate through the pores (see Appendix). By considering the equation of motion together with the law of continuity, an expression for the average velocity U can be obtained as

$$U = C\,(2gx)^{0.5} \qquad (1)$$

In Equation (1) the tacit assumption of a slug flow model is used. However, a coefficient C is introduced in Equation (1) as a correction factor to account for the complex flow situation prevailing in the porous fin. Several parameters such as mean pore diameter d, viscosity μ, density ρ_ℓ and surface tension σ are considered to be governing parameters that would fix the magnitude of C. By dimensional reasoning the governing π-parameter can be obtained as $\mu/(\sigma\,\rho_\ell\,d)^{0.5}$.

$$\text{Thus, } C = F\,[\mu/(\sigma\,\rho_\ell\,d)^{0.5}] \qquad (2)$$

where F denotes "function of".

However, an exact functional relationship for a given specimen of sintered metal can be established by a simple experimental procedure. The suction velocity V, normal to the lateral surface of the fin can be linked to the velocity U through the law of continuity

$$V\,P = d(U\,A_1)/dx \qquad (3)$$

where P is the perimeter of the porous fin and A_1 is the effective flow area of the liquid at any section x measured from the tip of the fin. Further, the fin cross section is assumed to be governed by a power law variation of the type.

$$A/A_0 = (x/L)^m = \xi^m \qquad (4)$$

where m is the fin profile index. Thus Equations (1), (3) and (4) yield the variation of the possible suction velocity

$$V/V_\ell = \xi^{m-0.5} \qquad (5)$$

where $V_\ell = C\,(m + 0.5)\,(\epsilon\,A_0/P)(2g/L)^{0.5}$ and ϵ (porosity) $= A_\ell/A$.

When $m \to 0$, the analysis corresponds to the case of a porous fin of non-varying cross-section. The equivalent thermal conduction through the porous fin can be evaluated with the aid of

$$k_{eff} = \epsilon\,k_\ell + (1 - \epsilon)\,k_w \qquad (6)$$

where k_ℓ and k_w are respectively the thermal conductivities of the condensate and the fin material. Equation (6) assumes that for a given temperature potential, thermal diffusion through the fin is the sum of the thermal components conducted by the liquid and the fin material.

Thus, the flow dynamics of the condensate on the lateral face of the fin is to be conjugated to thermal conduction within the fin which comprises two phases viz., solid material and condensate in the pores of the fin having an effective thermal conductivity as given by Equation (6). In addition, the convective component due to the flow of the condensate in the pores must also be included in the conservation of energy equation since it can assume as much importance as the thermal conduction within the fin. The thickness of the porous plate fin is considerably less than its length, so that one-dimensional thermal conduction within the fin is assumed to exist.

Thus, for the coordinate system shown in Figure 1b, the heat conduction in the porous fin of varying cross-sectional area can be described by (refer to Appendix).

$$\mathbf{D}[k_{eff} A\ (\mathbf{D}\ T_w)] - \mathbf{D}[\rho_\ell\ C_p\ T_w\ U\ A_\ell] + \rho_\ell\ C_p\ T_w\ \mathbf{D}[U\ A_\ell] + P\ k_L\ (\partial T/\partial y)|_{y=0} = 0 \quad (7)$$

where $\mathbf{D} \equiv d/dx$

Further, in the problem formulation, the assumptions employed in the classical analysis of Nusselt (1916) are employed. The condensate film is taken as being so thin that the inertial forces can be neglected and the temperature distribution across the condensate film is assumed to be linear. These assumptions are valid since the liquid film becomes thinner due to suction in comparison to that formed on a non-permeable surface. It was established by Roshenow (1973) that subcooling effects are not dominant for thin films. The equation for condensate film flow due to body force with the opposing shear resistance at the wall is

$$\tau_w = g(\rho_\ell - \rho_v)\delta \quad \ldots\ldots (8)$$

For the parabolic velocity profile approximation viz.,

$$u/u_i = [2(y/\delta) - (y/\delta)^2] \quad \ldots\ldots (9)$$

the shear at the wall can be expressed as

$$\tau_w = 2\mu u_i/\delta \quad \ldots\ldots (10)$$

PHASE TRANSFORMATION EQUATION:

The phase transformation is governed by the following mass balance for linear temperature variation across the condensate film.

$$\mathbf{D}[\textstyle\int_0^\delta u dy] + V = k_\ell\ (T_s - T_w)/[\delta\rho_\ell\ h_{fg}] \quad \ldots\ldots (11)$$

where the diffentntial operator $\mathbf{D} \equiv d/dx$. Equations (8) and (11) ignore the effect of the fin profile index m on the gravitational component. Such an assumption is valid for small values of the fin profile index m of Equation (4). Equations (7) and (11) with the aid of Equations (4), (5), (8), (9) and (10) can be rendered dimensionless as

$$\mathbf{D}[\xi^m\ \mathbf{D}T^+] - \mathbf{D}T^+\ [M\ \lambda\xi^{m+0.5}]/[(m + 0.5)\ Ku] - MT^+/\Delta = 0 \quad \ldots\ldots (12)$$

$$\mathbf{D}\Delta^3 + 3\ \lambda\xi^{m-0.5} = 3T^+/\Delta \quad \ldots\ldots (13)$$

where $\mathbf{D} \equiv d/d\xi$

The salient dimensionless groupings encountered in Equations (12) and (13) are listed in the Nomenclature.

The boundary conditions for Equations (12) and (13) are

$$\xi = 0;\ \Delta = dT^+/d\xi = 0 \qquad \xi = 1;\ T^+ = 1 \quad \ldots\ldots (14)$$

Simultaneous solutions of Equations (12) and (13) by employing the Runge-Kutta procedure, subject to the boundary conditions of Equation (14) will yield relevant information for the estimation of local and average heat transfer coefficients, average augmentation and efficiency of the porous fin.

HEAT TRANSFER COEFFICIENTS

The local heat transfer coefficient is given by the relation, $h = k_\ell/\delta$. In dimensionless form,

$$Nu/(Ra \cdot Ku)^{0.25} = 1/\Delta \quad \ldots\ldots (15)$$

The average Nusselt number can be evaluated as

$$Nu_m = \textstyle\int_0^1 Nu\ d\xi \quad \ldots\ldots (16)$$

For design purposes, Equations (15) and (16) can be applied only when the local and average temperature potentials are known *a priori* for the given system conditions. Since these values are unknown, the average heat transfer coefficient is redefined in terms of the known temperature potential at the base of the fin.

$$h_m(T_s - T_{w,1})PL = k_{eff}A_0\ [dT_w/dx]_{x=L} \quad \ldots\ldots (17)$$

Equation (17) in dimensionless form transforms to

$$Nu_m/(Ra \cdot \mathrm{Ku})^{0.25} = (1/M)\ [dT^+/d\xi]_{\xi=1} \quad \ldots\ldots (18)$$

Equation (17) is the lower bound in the estimation of the average heat transfer coefficient. If the fin is maintained under isothermal conditions at its base temperature $T_{w,1}$ all along its length, Nusselt's analysis (1916) holds good. For such a case the local and the average Nusselt numbers can be calculated with the aid of the Equations (19) and (20), respectively.

$$h_{iso}L/k_\ell = (4\ Ra\ Ku)^{0.25}\ \xi^{-0.25} \quad \ldots\ldots (19)$$

and

$$\bar{h}_{iso}L/k_\ell = 0.943\ (Ra\ Ku)^{0.25} \quad \ldots\ldots (20)$$

where $\bar{h}_{iso}$ is the average condensation heat transfer coefficient for isothermal conditions of the fin.

EFFICIENCY OF THE FIN:

Efficiency E, of the fin is defined as the ratio of the actual heat realised by the fin in condensing vapors, to the heat rejected to an isothermal vertical surface maintained at the base temperature $T_{w,\ell}$. Thus, efficiency is

$$E = k_{eff}A_0 \, (dT_w/dx)_{x=L} / [h_{iso} \, PL(T_s - T_w)] \quad \ldots (21)$$

Equation (21) can be transformed using the definitions of the relevant dimensionless groups.

$$E = (dT^+/d\xi)_{\xi=1} / (0.943 \, M) \quad \ldots (22)$$

According to this definition, efficiency would be same as the result obtained from Equation (18) except for a scale factor of multiplication.

AVERAGE AUGMENTATION

The average augmentation ratio ϕ, is defined as

$$\phi = \int_0^L h dx / \int_0^L h_{iso} \, dx \quad \ldots (23)$$

Making use of Equations (15) and (19), Equation (23) in dimensionless form can be rewritten as

$$\phi = 1.06 \int_0^1 d\xi / \Delta \quad \ldots (24)$$

Results and discussion

FIN TEMPERATURE PROFILES

Estimation of the possible ranges of porosity parameter λ and M are carried out for porosities varying between 1% to 5% and for condensation of vapors of ammonia and ethyl alcohol on porous fins made out of copper or stainless steel. For those ranges of M, λ and Ku, temperature profiles are shown plotted in Figure 2. For the case when $\lambda \rightarrow 0$, the temperature profiles of the present analysis agree well with those of Lu and Lii (1985) (see Figure 2). To avoid overcrowding of the lines in Figure 2, comparison is made for three values of the fin parameter. Hence, the validity of the formulation and numerical analysis employed in the present study is indirectly assured. Further, Figure 2 reveals that the fin parameter M and the porosity parameter λ exhibit influences on the temperature profiles. Nevertheless, the relative influence of M on the temperature profile is found to be more than that due to λ, the porosity parameter. High values of M will make the fin highly non-isothermal, whereas the fin tends to an isothermal condition when $M \rightarrow 0$. Besides, for a given m, an increase in the value of the porosity parameter λ will further decrease the dimensionless local temperature in contrast to the case of a solid fin i.e., $\lambda = 0$. Such a situation implies that the local temperature potential decreases with an increase in suction velocity at the lateral face of the fin, indicating efficient thermal conduction and convection along its length. For a given value of the fin parameter M and the porosity parameter λ, the temperature variation along the fin is significantly affected by the shape of the fin. The effect of fin profile index m on the temperature profile is more dominant at the tip than at the base. Since the trends observed are similar to those already depicted by Sarma et al. (1988) the results are not shown.

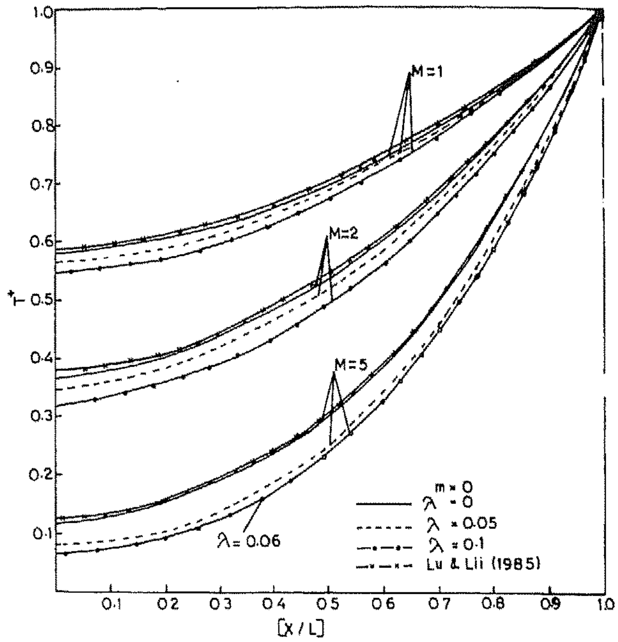

Figure 2 — Effect of porosity parameter λ on the fin temperature profile (for $M = 1, 2, 5$; $m = 0$).

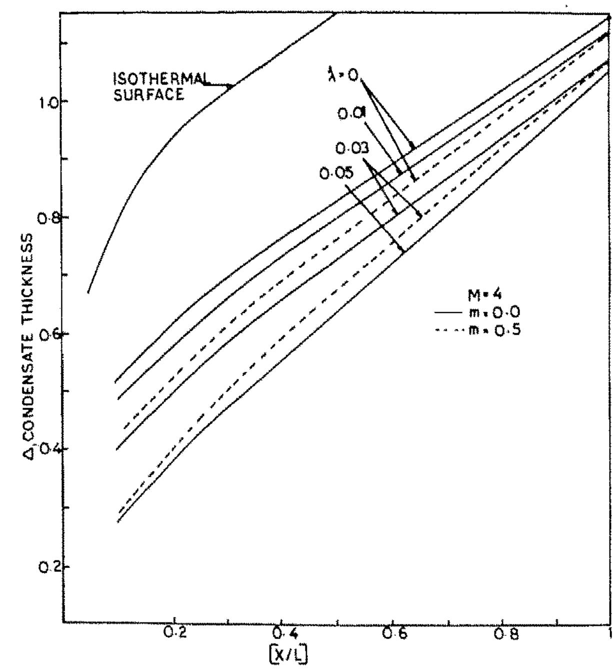

Figure 3 — Effect of porosity parameter λ on condensation film thickness. ($m = 0, 0.5$).

CONDENSATE FILM THICKNESS:

In Figure 3, the variation of condensate film thickness is shown plotted for $0 < \lambda < 0.05$ for $M = 4$ and $m = 0$ and 0.5. Evidently, at a given location the suction effect induced at the wall makes the condensate layer thinner in contrast to the case where $\lambda = 0$. Further, the film thickness for the isothermal condition is also represented in Figure 3. It can be inferred that the profiles of a fin other than a plate type configuration will also exert an influence by decreasing the thickness of the condensate film all along its length.

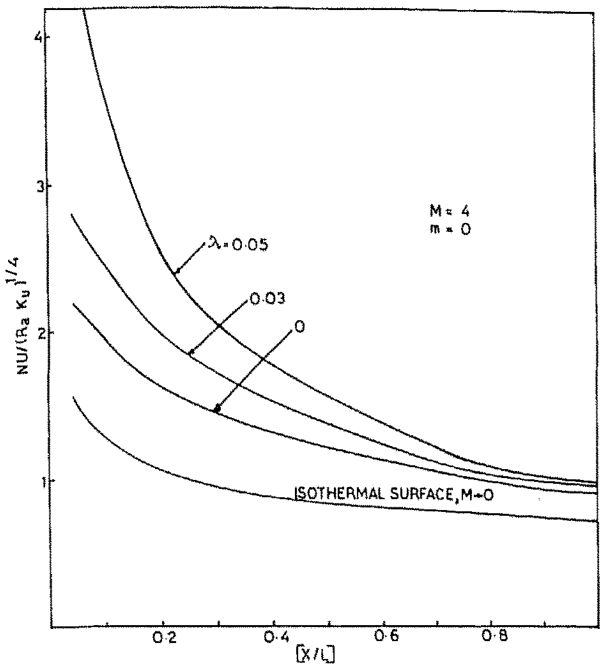

Figure 4 — Effect of porosity parameter λ on local condensation heat transfer coefficients.

HEAT TRANSFER COEFFICIENTS AND AN AUGMENTATION:

The local variation of Nusselt number along the fin is plotted in Figure 4. The salient observation from Figure 4 is that, increasing values of λ give rise to locally enhanced values of the Nusselt number. In addition, fins of either a porous or a non-porous type always indicate higher values than those observed from Nusselt's (1916) analysis. The information from Figure 4 reveals that by increasing the fin profile index m, the local Nusselt number can be increased at any given location on the fin. Hence, porous fins of varying cross-section give higher heat transfer coefficients than in the case of solid fins of non-varying cross-section.

Equations (12) and (13) are also solved with the subcooling parameter varying in the range $5 < Ku < 200$. For this range, the dimensionless heat transfer coefficients are not found to be influenced except for a change in the third decimal place in the numerical values. These results revealed that for thin films of condensate on porous fins under non-isothermal conditions the influence of the subcooling parameter on the condensation heat transfer coefficient is negligible. Such an observation was made earlier by Merte (1973) for the case of condensation on a vertical isothermal surface.

The temperature gradient at the base of the fin is generally a required parameter. Hence, these results are shown in Figure 5. By increasing the fin parameter M, the temperature gradient at the base is monotonically increased for a fin with specified porosity. In addition, by changing the structure of the sintered metal (i.e., affecting the porosity), the temperature gradients can also be altered. By increasing λ, the temperature gradient at the base can be enhanced for any given value of the fin parameter. Equation (18) can be evaluated with the aid of the results shown on Figure 5.

Equation (19) was utilised for the evaluation of fin efficiency. The observed trends (data not shown) from the numerical results are that the fin efficiency decreases with an increase in the fin parameter for a given value of λ. In addition, the influence of the fin profile index m on the efficiency is marginal.

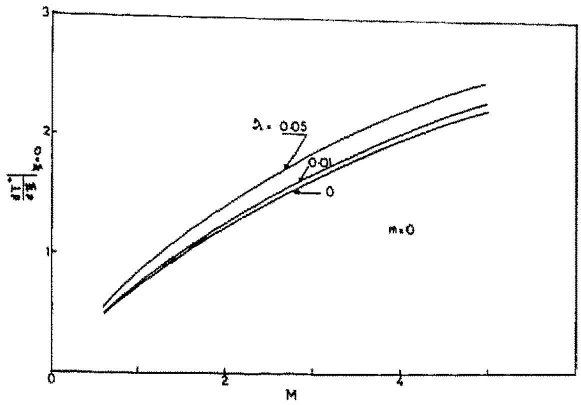

Figure 5 — Variation of temperature derivative at the base of the fin (0 < ? < 0.05).

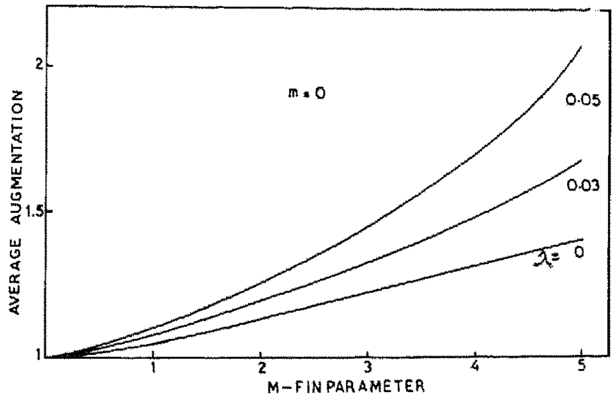

Figure 6 — Effect of porosity on average augmentation condensation heat transfer.

To establish the superiority of the performance of the porous fin, average values of augmentation as defined by Equation (20) are evaluated and further depicted in Figure 6. It is obvious that for a given value of M, an increase in porosity gives a substantial rise in augmentation. For example, from Figure 6 it is evident that at higher values of the fin parameter M, the augmentatin of a porous fin is around 200% for $\lambda = 0.05$. Such a significant gain would be highly advantageous in affecting the compactness of heat pipes for which the analysis is intended. For the sake of design purposes, Equation (18) is expressed by the following relation with the aid of computer results for a plate type of porous fin i.e., $m = 0$.

$$Nu/(Ra\ Ku)^{0.25} = 0.71\ M^{-0.242} \exp(2.66\lambda) \quad \ldots. \quad (25)$$

where $0.017 < M < 5$ and $0 < \lambda < 0.05$.

Equation (25), obtained by regression, estimates the average Nusselt number with a deviation of ±6%.

Conclusions

The following conclusions can be drawn from this theoretical study:

1. Condensation of vapors on a fin, of porous or non-porous type, will make it highly non-isothermal for high values of the fin parameter.
2. The sub-cooling parameter, Ku, has insignificant influence on the local and mean condensation heat transfer coefficients in the range $5 < Ku < 200$ for thin films.

3. A porous fin is superior to a non-porous one in its performance. The condensation heat transfer coefficient can be substantially increased by choosing an appropriate value of the porosity and fin parameters.
4. Porous fins of varying cross-section have indicated marginal gain in influencing both the efficiency and the augmentation, in contrast to plate fins of uniform cross-section. Nevertheless, by making use of fins of varying cross-section, material saving can be accomplished.
5. Equation (25) can be employed for design purposes for the case of $m = 0, 0.0176 < M < 5$ and $0 < \lambda < 0.05$. Nevertheless, this theoretical study should be further validated by conducting an experimental investigation of the condensation of vapors on porous fins.
6. Fins of this type of geometric configuration can be incorporated into the condenser section of a flat plate heat pipe to enhance its performance and to make it more compact.

Appendix

Determination of the average flow velocity of condensate in the porous fin:

The equation of motion of the condensate through porous fin is derived by applying a force balance for the liquid element between sections 11 and 22 (see Figure 1b). For invicid flow conditions of the condensate, the equation can be written as

$$\mathbf{D}[\rho_\ell A_\ell U^2] - \rho_\ell VUP = \rho_\ell A_\ell g \quad \text{(A1)}$$

where $\mathbf{D} = d/dx$.

Further, the pressure gradient is ignored because the porous fin is exposed to an ambient medium of vapor at finite system pressure. The law of continuity is given by

$$\mathbf{D}[U A_\ell] = VP \quad \text{(A2)}$$

Solving Equations (A1) and (A2) subject to the boundary conditions $x = 0$, $U = 0$, the average velocity U is given by

$$U = (2gx)^{0.5} \quad \text{(A3)}$$

Equation (A3) is for an highly idealized situation which assumes potential flow condition. However, the flow of the condensate is highly tortuous because of the porous structure. Equation (A3) is to be modified in the context of the physical situation under study. Hence, a correction factor C is introduced to take care of the effects of several factors such as viscous resistance to flow and the withdrawal rate of the condensate at the base of the fin back into the evaporator section with the aid of a porous wick. Thus, the influencing parameters are pore diameter d, viscosity μ, surface tension σ (ensuring feed back of the condensate into the evaporator section).

Equation (A3) is modified to the form

$$U = C\,(2gx)^{0.5} \quad \text{(A4)}$$

where C, as given by dimensional analysis is governed by the π-parameter, $C = F(\sigma\rho_\ell\, d/\mu^2)$, Equation (2).

Hence, C can be deemed to be a factor similar to the coefficient of discharge which can be experimentally determined for the given specimen on which condensation takes place.

FIN EQUATION:

For the configuration of the control volume represented between sections 11-22 as shown in Figure (1b) the conservation of energy can be written as follows

$$\mathbf{D}[K_{eff} A_\ell \mathbf{D} T_w] - \mathbf{D}[\rho_\ell C_p A_\ell UT_w] + \rho_\ell C_p VPT_w + k_\ell \mathrm{P}[(T_s - T_w)/\delta] = 0 \quad \text{(A5)}$$

Equation (A5) is derived based on the conditions that the temperatures of wall and the liquid are equal at any given location. Making use of the law of continuity, Equation (A5) for the porous fin will be as follows:

$$\mathbf{D}[k_{eff} A\, \mathbf{D}\theta_w] - [UA_\ell\rho_\ell C_p]\, \mathbf{D}\theta_w - [k_\ell \mathrm{P}\theta_w/\delta] = 0 \quad \text{(A6)}$$

where $\theta_w = (T_s - T_w)$

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Nomenclature

A	= cross sectional areas of the fin at any x, m^2
A_0	= cross-section of the fin at its base m^2
A_ℓ	= live flow area of condensate at any x in the fin, m^2
C	= correction factor defined by Equation (2), dimensionless
C_p	= specific heat at constant pressure, J/kg · K
d	= diameter of the pore in the fin, m
E	= efficiency of the fin, defined by Equations (21) or (22)
g	= acceleration due to gravity, m/s^2
Gr	= Grashof number, $gL^3 (\rho_\ell - \rho_v)/(\rho_\ell \mu^2)$
h	= local heat transfer coefficient W/m^2 · K
h_m	= mean heat transfer coefficient, W/m^2 · K
h_{fg}	= latent heat of condensation, J/kg
k	= thermal conductivity, W/m · K
Ku	= sub-cooling parameter, $h_{fg}/[C_p(T_s - T_{w,0})]$
L	= length of the fin, m
m	= fin profile index
M	= fin parameter, $(k_\ell PL/k_w A)(Ra \cdot \mathrm{Ku})^{0.25}$
Nu	= local Nusselt number, hL/k_ℓ
Nu_m	= mean Nusselt number, h_mL/k_ℓ
P	= perimeter of the fin, m
Pr	= Prandtl number of condensate, $\mu C_p/k_\ell$
Ra	= modified Rayleigh number, $Gr \cdot Pr$
T	= temperature, K
T^+	= dimensionless temperature, $(T_s - T_w)/(T_s - T_{w,1})$
u	= velocity of the condensate layer at any y, m/s
U	= average velocity of condensate flow in the pores, m/s
V	= suction velocity normal to the plate fin, m/s
V_ℓ	= suction velocity normal to the plate fin at the base i.e., at $x = L$, m/s
x, y, z	= spatial coordinates

Greek symbols

α	= thermal diffusivity of the condensate, m^2/s
δ	= condensate film thickness, m
Δ	= dimensionless film thickness, $(\delta/L)(Ra \cdot Ku)^{0.25}$
λ	= porosity parameter $(LV_L/\alpha)\, Ku^{0.75}/Ra^{0.25}$
μ	= absolute viscosity of the condensate, N · s/m^2
ν	= kinematic viscosity of the condensate, m^2/s
ϕ	= average augmentation ratio, Equation (24)
ρ	= density, kg/m^3
ξ	= dimensionless space variable, x/L

Subscripts

eff = effective
i = vapor-liquid interface
iso = isothermal condition
ℓ = liquid
s = saturation
v = vapor
w = wall
w, 1 = base of the fin ie., at $x = L$

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