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An analytical investigation into the Nusselt number and entropy generation rate of film condensation on a horizontal plate †

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Abstract

This study investigates the heat transfer characteristics and entropy generation rate of a condensate film formed on a horizontal plate with suction at the wall. Applying the minimum mechanical energy principle, the dimensionless liquid film thickness along the plate is found to vary as a function of the Rayleigh number, the Jakob number, the Prandtl number and the suction parameter. The governing differential equation of the condensate thickness is solved numerically by using a finite-difference shooting method. Closed-form analytical expressions are derived for the Nusselt number and the dimensionless overall entropy generation number. When there is no suction at the wall, the results obtained from the analytical expression for the Nusselt number are found to be in good agreement with those presented in the literature.

Keywords: Film condensation; Minimum entropy generation; Wall suction

1. Introduction

Laminar film condensation is of practical importance in a variety of engineering and industrial applications ranging from heat exchangers to drying and cooling processes, packed-bed heat exchangers, distillation processes, chemical vapor deposition processes, and so on. The problem of laminar film condensation on a vertical plate was first addressed by Nusselt [1] in 1916 and has been the subject of intense study ever since. Following the publication of Nusselt's original study, many researchers attempted to improve the accuracy of the analytical results by incorporating more realistic assumptions. The earliest study of film condensation heat transfer on a horizontal surface was performed by Popov [2]. In a later study, Nimmo and Leppert [3, 4] conducted experimental and theoretical investigations into film condensation on a finite-size horizontal plate. In the theoretical study of Nimmo and Leppert [4] , the boundary condition at the edge of the plate was simply assumed by a particular boundary condition. By contrast, in investigating the same problem, Shigechi et al. [5] improved the accuracy of the numerical results by adjusting the inclination angle of the vapor-liquid interface at the plate edge. However, the precise nature of the boundary condition at the plate edge was not fully clarified. Adopting a special transformation method with certain approximations, Yang and Chen [6] conducted an analytical investigation into the problem of condensation on a horizontal plate with suction effects acting at the wall. The results indicated that the presence of a suction force yielded a significant improvement in the condensation heat transfer performance. However, the closed-form analytical expression for the Nusselt number proposed in their study did not consider the

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wall suction parameter.

In practical thermal systems, irreversibilities due to friction and heat transfer inevitably exist. These irreversibilities induce an entropy generation effect and destroy the energy available in the system to perform work. Bejan [7] proposed a method for minimizing the rate of entropy generation in single-phase convection heat transfer systems. Adeyinka and Naterer [8] analyzed the problem of entropy generation in a vertical condensation film subject to phase change. Recently, Dung and Yang [9] applied the entropy generation minimization method proposed by Bejan to optimize the heat transfer in a condensate film on a horizontal tube.

Although many researchers have studied the problem of film condensation heat transfer on a horizontal plate, a comprehensive thermodynamic analysis of the detailed characteristics of this particular heat transfer problem has yet to be presented. Accordingly, the current study performs thermodynamic analyses of laminar film condensation on a horizontal plate with suction effects acting at the wall. A closed-form analytical expression for the Nusselt number is derived as a function of the Jakob number *Ja*, the Rayleigh number *Ra* and the suction parameter *Sw*. In addition, an approach is proposed for minimizing the overall entropy generation induced by the condensation heat transfer and condensate film flow friction irreversibilities.

2. Analysis

The current analysis considers a pure quiescent vapor in a saturated state with a uniform temperature T_{sat} and pressure P_{sat} condensing on a horizontal, clean, permeable flat plate with a width $2L$ and a constant temperature T_w. Fig. 1 presents a schematic diagram of the physical model and the corresponding coordinate system. Under steady-state conditions, a filmwise condensate layer forms on the plate and has a maximum thickness at the plate center. The flow rate of this condensate layer is governed by the variation in the hydrostatic pressure acting upon it. The momentum boundary layer is subject to a uniform suction force which removes the condensate at a constant velocity. In investigating the heat transfer characteristics of the condensate film, the following assumptions are made: (1) the inertia and kinetic terms within the liquid film are negligible, (2) the physical properties of the dry vapor and condensate layer are constant,

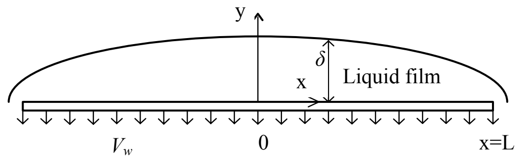

Fig. 1. Condensate film flow on finite-size horizontal plate with suction at wall.

and (3) the wall and the vapor both have a uniform temperature. Given these assumptions, the governing equations for the liquid film with boundary layer simplifications have the form:
continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0\ , \tag{1}$$

x-direction momentum equation:

$$0=-\frac{\partial P}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial y^2}\right), \tag{2}$$

y-direction momentum equation:

$$P=P_{sat}+\rho g\left(\delta-y\right), \tag{3}$$

energy equation:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\frac{k}{\rho Cp}\frac{\partial^2 T}{\partial y^2}\ , \tag{4}$$

where u and v are velocity components in the *x*- and *y*-directions, respectively, P is the liquid pressure, δ is the thickness of the liquid film, and ρ, Cp, μ and k are the density, specific heat, dynamic viscosity and thermal conductivity of the saturated liquid, respectively.

The boundary conditions imposed at the plate surface (y=0)are given by

$$u=0 \tag{5}$$
$$v=v_w, \tag{6}$$
$$T=T_w, \tag{7}$$

while those at the liquid-vapor interface ($y=\delta$)have the form

$$\frac{\partial u}{\partial y}=0\ , \tag{8}$$

$$T = T_{sat} \,. \tag{9}$$

Substituting Eq. (3) into Eq. (2) and applying the boundary conditions given in Eqs. (5) and (8), the velocity profile can be expressed as

$$u = \frac{\rho g}{\mu}\frac{d\delta}{dx}\left(\frac{y^2}{2} - y\delta\right). \tag{10}$$

Since the thickness of the liquid film is very small relative to the width of the plate, the temperature profile can be expressed in the following linear form:

$$T = T_w + \Delta T \frac{y}{\delta}, \tag{11}$$

where $\Delta T = T_{sat} - T_w$.

According to Bejan [7], the local entropy generation rate for two-dimensional convection heat transfer can be written as

$$S_g''' = \frac{k}{T_w^2}\left[\left(\frac{\partial T}{\partial x}\right)^2 + \left(\frac{\partial T}{\partial y}\right)^2\right] + \frac{\mu}{T_w}\left(\frac{\partial u}{\partial y}\right)^2. \tag{12}$$

The overall entropy generation rate in the current problem is derived by the following three-step procedure. Eqs. (10) and (11) are substituted into Eq. (12) to yield the local entropy generation rate:

$$S_g''' = \frac{k}{T_w^2}\left(\frac{\Delta T}{\delta}\right)^2 + \frac{\mu}{T_w}\left[\frac{\rho g}{\mu}\frac{d\delta}{dx}(y-\delta)\right]^2. \tag{13}$$

Eq. (13) is then integrated with respect to y from zero to δ to obtain the entropy generation rate across the film thickness:

$$S_g' = \int_0^\delta S_g''' dy = \frac{k}{T_w^2}\frac{\Delta T^2}{\delta} + \frac{\rho^2 g^2 \delta^3}{3\mu T_w}\left(\frac{d\delta}{dx}\right)^2. \tag{14}$$

Finally, Eq. (14) is integrated from $x= -L$ to $x=L$ to obtain the overall entropy generation rate across the width of the plate.

$$S_g = \int_{-L}^{L} S_g' dx = 2\int_0^L S_g' dx = \frac{2k\Delta T^2}{T_w^2}\int_0^L \frac{1}{\delta}dx + \frac{2\rho^2 g^2}{3\mu T_w}\int_0^L \delta^3\left(\frac{d\delta}{dx}\right)^2 dx \,. \tag{15}$$

In Eq. (15), the thickness of the liquid film, δ, is the only unknown.

From the first law of thermodynamics and the Fourier conduction law, the thermal energy balance between the condensate layer and the plate surface can be given as

$$\frac{d}{dx}\left\{\int_0^\delta \rho u\left(h_{fg} + Cp\left(T_{sat} - T\right)\right)\times dy\right\}dx + \rho v_w\left(h_{fg} + Cp\Delta T\right)dx = \frac{k\Delta T}{\delta} \,. \tag{16}$$

The right hand side of Eq. (16) represents the energy transferred from the liquid film to the solid plate. Meanwhile, the first term on the left hand side expresses the net energy flux across the liquid film (from x to $x+dx$)while the second term describes the net energy sucked out of the condensate layer by the permeable plate. The assumption is made that the suction velocity, v_w, is constant.

Substituting Eqs. (10) and (11) into Eq. (16) and rearranging yields

$$\delta\frac{d}{dr}\left(\delta^3\frac{d\delta}{dx}\right) = -\frac{3k\Delta T\mu}{\rho^2 g\left(h_{fg} + \frac{3}{8}Cp\Delta T\right)} + 3\delta\left(\frac{h_{fg} + Cp\Delta T}{h_{fg} + \frac{3}{8}Cp\Delta T}\right)\frac{v_w \mu}{\rho g} \,. \tag{17}$$

For analytical convenience, let the following parameters be defined:

$$\mathrm{Ja} = \frac{\mathrm{Cp}\Delta T}{\mathrm{h_{fg}} + \frac{3}{8}Cp\Delta T}, \tag{18}$$

$$Ra = \frac{\rho^2 g \Pr L^3}{\mu^2}, \tag{19}$$

$$\Pr = \frac{\mu \mathrm{Cp}}{\mathrm{k}} \tag{20}$$

$$Sw = \frac{\Pr}{Ja}\frac{\rho v_w L}{\mu}\left(1 + \frac{5}{8}Ja\right), \tag{21}$$

where Ja is the Jakob number, Ra is the Rayleigh number, Pr is the Prandtl number and Sw is the suction parameter.

With these parameters, the thermal energy balance given in Eq. (17) can be rewritten as

$$\delta \frac{d}{dx}\left(\delta^3 \frac{d\delta}{dx} \right) = 3\frac{Ja}{Ra}\left(SwL^2\delta - L^3 \right). \tag{22}$$

The liquid film has its maximum thickness at the center of the place (i.e., x=0) and the film thickness gradually reduces in the *x*-direction toward a minimum value at the plate edge. The boundary conditions of the film thickness can therefore be written as

$$\frac{d\delta}{dx}\text{, at } x = 0 \tag{23}$$

$$\delta = \delta_{\min}\text{ . at } x = L \tag{24}$$

However, it is still impossible to solve Eq. (22) since $\delta_{\min}$ is unknown.

In practice, the condensate film cannot have exactly zero thickness at the plate edge. Chang [10, 11] suggested that the critical condensate thickness could be derived by using the minimum mechanical energy principle presented by Bakhmeteff [12]:

$$\left[\frac{\partial}{\partial \delta}\int_0^{\delta}\left(\frac{u^2}{2} + gy + \frac{P}{\rho} \right)\rho u dy \right]\Bigg|_{\dot{m}_c} = 0\,, \tag{25}$$

where $\dot{m}_c$ is the critical value of the mass flow over the plate edge.

Substituting Eqs. (3) and (10) into Eq. (25) yields the new boundary condition as

$$\left.\frac{d\delta}{dx}\right|_{x=L} = -\left(\frac{35}{6}\frac{\Pr}{\text{Ra}}\frac{L^3}{\left(\delta|_{x=L} \right)^3} \right)^{1/2}\,. \tag{26}$$

For convenience, the following normalized variables are introduced:

$$x^* = x/L\,, \tag{27}$$

$$\delta^* = \delta/L\,. \tag{28}$$

Eq. (22) and the corresponding boundary condition equations (Eqs. (23) and (26)) can then be rewritten as

$$\delta^* \frac{d}{dx^*}\left(\delta^{*3} \frac{d\delta^*}{dx^*} \right) = 3\frac{Ja}{Ra}\left(Sw\delta^* - 1 \right), \tag{29}$$

$$\frac{d\delta^*}{dx^*} \text{ at } x^* = 0\,, \tag{30}$$

$$\left.\frac{d\delta^*}{dx^*}\right|_{x^*=1} = -\left(\frac{35}{6}\frac{\Pr}{Ra\left(\delta^*|_{x^*=1} \right)} \right)^{1/2} \text{ at } x^* = 1. \tag{31}$$

The governing differential equation of the condensate thickness (Eq. (29)) is solved numerically subject to the corresponding boundary conditions (Eqs. (30) and (31)) by using a finite-difference shooting method. The overall solution procedure is illustrated schematically in Fig. 2. When the final value of δ^* has been derived at each grid point along the *x*-direction, the mean Nusselt number is computed in accordance with

$$\overline{Nu} = \frac{\overline{h}L}{k} = \int_0^1 \frac{1}{\delta^*}dx^*\,, \tag{32}$$

where

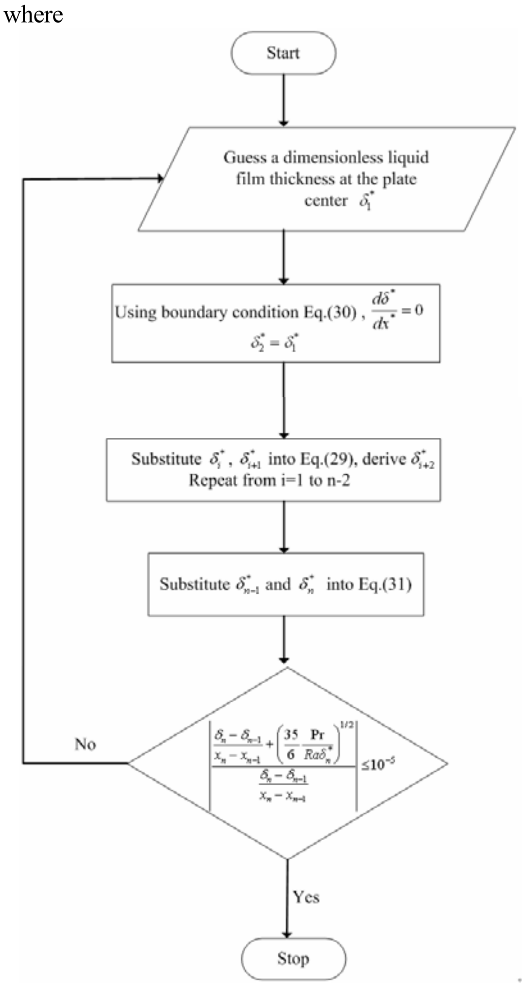

Fig. 2. Numerical solution procedure.

$$\bar{h} = \frac{1}{2L}\int_{-L}^{L} h(x)dx = \frac{1}{L}\int_{0}^{L}\frac{k}{\delta}dx = \frac{k}{L}\int_{0}^{1}\frac{1}{\delta^*}dx^* . \quad (33)$$

Using the normalized variables and dimensionless parameters defined in Eqs. (18)-(21), (27) and (28), the overall entropy generation rate given in Eq. (15) can be rewritten as

$$S_g = \frac{2k\Delta T^2}{T_w^2}\int_0^1 \frac{1}{\delta^*}dx^* + \frac{2(\rho g)^2}{3\mu T_w}\int_0^1 \delta^{*3}\left(\frac{d\delta^*}{dx^*}\right)^2 dx^* . \quad (34)$$

Let the following dimensionless parameters be defined:

$$\psi = \frac{\Delta \mathrm{T}}{\mathrm{T_w}} , \quad (35)$$

$$Br = \frac{gL\mu}{kT_w} , \quad (36)$$

$$Gr = \frac{\rho^2 g L^3}{\mu^2} , \quad (37)$$

where ψ is the dimensionless temperature difference, Br is the modified Brinkman number, and Gr is the modified Grashof number.

Eq. (34) can then be rewritten in dimensionless form as

$$S_g = 2k\psi^2 \int_0^1 \frac{1}{\delta^*}dx^* + \frac{2}{3}kBrGr\int_0^1 \delta^{*3}\left(\frac{d\delta^*}{dx^*}\right)^2 dx^* . \quad (38)$$

The characteristic entropy generation rate can be defined as

$$S_0 = \frac{2k\Delta T^2}{T_w^2} = 2k\psi^2 . \quad (39)$$

The dimensionless overall entropy generation number, $N = \frac{S_g}{S_0}$, can then be written as

$$N = \frac{S_g}{S_0} = \int_0^1 \frac{1}{\delta^*}dx^* + \frac{1}{3}BrGr\psi^{-2}\int_0^1 \delta^{*3}\left(\frac{d\delta^*}{dx^*}\right)^2 dx^* . \quad (40)$$

In Eq. (40), the first and second terms on the right hand side correspond to the entropy induced by the heat transfer irreversibility and the liquid film flow friction irreversibility, respectively. It is observed that the term describing the heat transfer irreversibility is identical to Eq. (32), i.e., the analytical expression for the mean Nusselt number.

3. Results and discussion

Fig. 3 illustrates the variation of the dimensionless liquid film thickness along the *x*-direction for four different combinations of *Ja/Pr* and *Sw*. As expected, the thickness of the liquid film reduces as the value of the suction parameter, *Sw*, increases. Additionally, it can be seen that the film thickness near the plate edge increases as the ratio *Ja/Pr* decreases. This result is reasonable since a larger hydrostatic pressure gradient is required to drive the flow over the plate edge when the fluid has a higher Prandtl number (i.e., a smaller *Ja/Pr* ratio). However, in the non-edge region of the plate, it is apparent that the film thickness is insensitive to *Ja/Pr*.

As shown in Eq. (32), the Nusselt number varies inversely with the film thickness since the thermal resistance of the condensate layer increases as its thickness increases. Furthermore, Eqs. (29) and (32) show that the value of the Nusselt number also varies as a function of Ja, Ra, Pr and *Sw*. Fig. 4 shows that the mean heat transfer coefficient, $\overline{Nu}\left(Ja/Ra\right)^{1/5}$, increases significantly as the suction parameter *Sw* is increased. However, $\overline{Nu}\left(Ja/Ra\right)^{1/5}$ increases only slightly with increasing *Ja/Pr* (as observed also in Fig. 3). From inspection it is found that if the effects of the

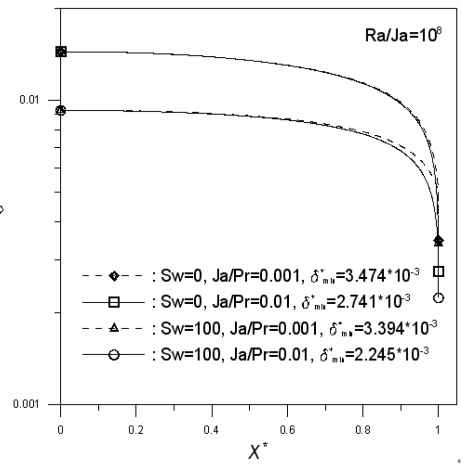

Fig. 3. Variation of dimensionless film thickness in x-direction.

Prandtl number, Pr, are neglected, the corresponding error in the computed value of the mean heat transfer coefficient is less than 2%. Accordingly, the remaining analyses neglect the effects of Pr and simply assign a constant value of Pr=1.76 (calculated from water at 100℃). Fig. 5 plots the variation of the mean heat transfer coefficient as a function of the wall suction parameter. Applying a curve-fitting technique, it can be shown that $\overline{Nu}$ and Sw are related as follows:

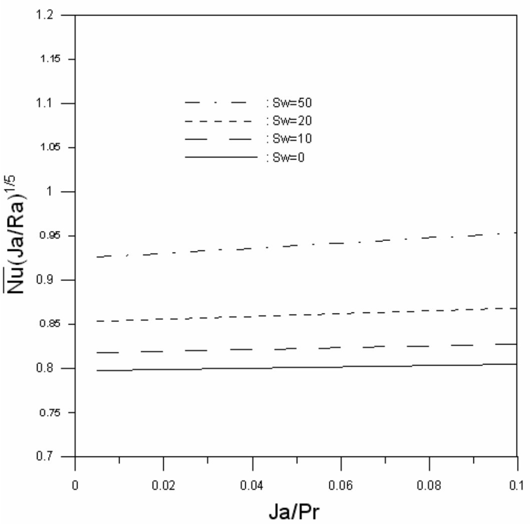

Fig. 4. Variation of heat transfer coefficient with Ja/Pr as function of Sw.

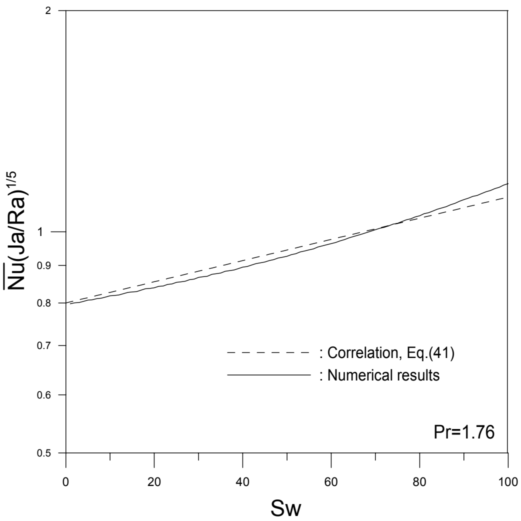

Fig. 5. Comparison of Nusselt number correlation results and numerical results.

$$\overline{Nu} = 0.8\times(1.00378)^{Sw}\left(\frac{Ra}{Ja}\right)^{1/5} . \tag{41}$$

From inspection, the maximum error between the results predicted by Eq. (41) and those determined numerically is found to be less than 4%.

In their study of the heat transfer characteristics of a condensate film on a finite-size horizontal plate, Yang and Chen [6] showed that in the absence of wall suction effects, the Nusselt number is given by

$$\overline{Nu} = 0.81\left(\frac{Ra}{Ja}\right)^{1/5} . \tag{42}$$

Substituting a value of Sw=0 into Eq. (41), it is found that the expression developed in the current study for the mean heat transfer coefficient of a condensate layer on a finite-size horizontal disk is consistent with that presented by Yang and Chen [6].

Fig. 6 shows that the liquid film flow friction irreversibility (as given by the second term on the right hand side of Eq. (40)) reduces as the wall suction parameter increases. Applying a curve-fitting technique, it is found that $\int_0^1 \delta^{*3}\left(\frac{d\delta^*}{dx^*}\right)^2 dx^*$ and Sw are related approximately as follows:

$$\int_0^1 \delta^{*3}\left(\frac{d\delta^*}{dx^*}\right)^2 dx^* = 2.1\times10^{-4}\left(\frac{Ja}{Ra}\right)\times(1.021)^{-Sw} . \tag{43}$$

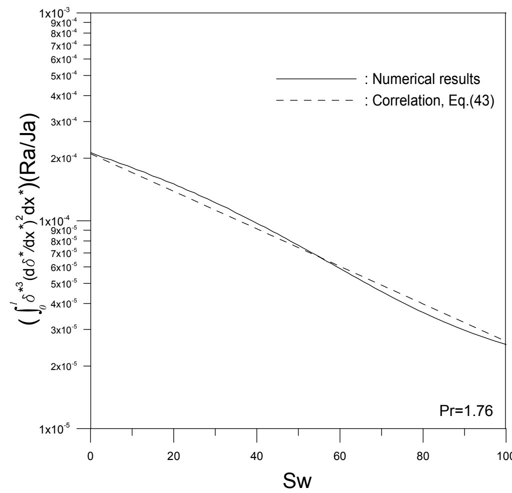

Fig. 6. Comparison of liquid film flow friction irreversibility correlation results and numerical results.

Substituting Eqs. (41) and (43) into Eq. (40) yields the following expression for the dimensionless overall entropy generation number:

$$N = 0.8\times(1.00378)^{Sw}\left(\frac{Ra}{Ja}\right)^{1/5} + 7\times10^{-5}BrGr\psi^{-2}(1.021)^{-Sw}\left(\frac{Ja}{Ra}\right) \quad (44)$$

In general, for a specified value of the parameter group $BrGr\psi^{-2}$, the dimensionless overall entropy generation number, *N*, can be minimized by differentiating *N* with respect to *Ra/Ja* and setting it equal to zero (i.e. $\frac{dN}{d\left(Ra/Ja\right)}=0$). The optimal value of *Ra/Ja* can be obtained as

$$\left(\frac{Ra}{Ja}\right)_{opt} = 1.588\times10^{-3}\left(BrGr\psi^{-2}\right)^{5/6}\times(1.0207)^{-Sw} \quad (45)$$

Fig. 7 shows the variation of the dimensionless total entropy generation number, *N*, with the ratio *Ra/Ja* as a function of *Sw* for constant $BrGr\psi^{-2}=10^8$. From Eq. (45), it is calculated that the $\left(\frac{Ra}{Ja}\right)_{opt}$ =7371, 2646 and 950 for wall suction parameter values of *Sw*= 0, 50 and 100, respectively. Then corresponding minimum values of *N* are also derived as 5.7, 5.61 and 5.52 for wall suction parameter values of *Sw*= 0, 50 and 100, respectively.

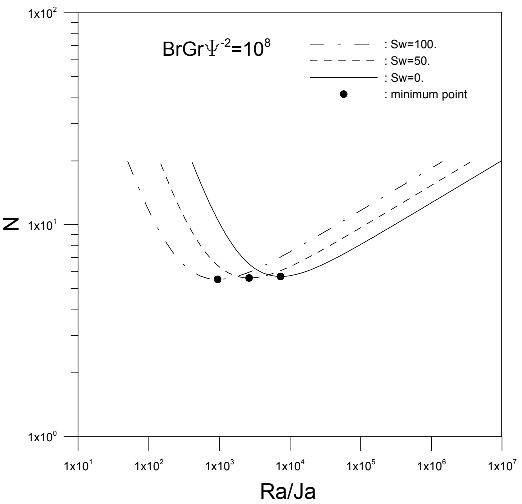

Fig. 7. Variation of entropy generation parameter with *Ra*/*Ja* as function of *Sw*.

4. Conclusion

This study has investigated the heat transfer characteristics and the overall entropy generation rate in a condensate layer on a finite-size horizontal plate with suction effects acting at the wall. The minimum mechanical energy principle has been applied to determine the boundary condition at the plate edge. The results show that the mean Nusselt number varies as a function of the ratio *Ra/Ja* and *Sw*. Moreover, the overall entropy generation rate induced by the heat transfer irreversibility effect is equivalent to the Nusselt number. Applying a curve-fitting method, closed-form analytical expressions have been developed to predict the mean Nusselt number and the entropy generation rate. In both cases, a good agreement has been observed between the predicted results and the numerical solutions. Finally, the value of *Ra/Ja* which minimizes the dimensionless overall entropy generation number has been derived as a function of the parameter group $BrGr\psi^{-2}$.

Acknowledgments

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Nomenclature

Br	:	Modified Brinkman number (defined in Eq. (36))
Cp	:	Specific heat at constant pressure
g	:	Acceleration of gravity
Gr	:	Modified Grashof number (defined in Eq. (37))
h	:	Heat transfer coefficient
h_{fg}	:	Vaporization heat
Ja	:	Jakob number (defined in Eq. (18))
k	:	Thermal conductivity
L	:	Half-width of plate
$\dot{m}$	:	Condensate mass flux
N	:	Dimensionless overall entropy generation number (defined in Eq. (40))
Nu	:	Nusselt number (defined in Eq. (32))
P	:	Pressure
Pr	:	Prandtl number (defined in Eq. (20))

Ra : Rayleigh number (defined in Eq. (19))
S''_g : Local entropy generation rate
S_g : Overall entropy generation rate
S_0 : Characteristic entropy generation rate
Sw : Suction parameter (defined in Eq. (21))
T : Temperature
ΔT : Saturation temperature minus wall temperature
x, y : Horizontal and vertical coordinates
u, v : Horizontal and vertical velocity components

Greek symbols

δ : Condensate film thickness
μ : Liquid viscosity
ρ : Liquid density
ψ : Dimensionless temperature difference (defined in Eq. (35))

Superscripts

— : Average quantity
* : Dimensionless variable

Subscripts

min : Minimum quantity
sat : Saturation property
w : Quantity at wall

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