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# Noncondensable gas effect on condensation in a separate type two-phase closed thermosyphon

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**Abstract.** An analytical study is presented of the noncondensable gas effect on vapor condensation in a separate type two-phase closed thermosyphon. The appreciable detrimental effect is observed from the results obtained by investigating condensation of steam-air and steam-hydrogen mixtures. It is shown that the effect of hydrogen on condensation is more remarkable than that of air for the same operating conditions of the thermosyphon. By examining four working temperature levels, the noncondensable gas effect is found to be accentuated at high system pressures. The computed results are all corresponding to the parameter range of engineering interest, and intended to have practical applications to designing separate type heat pipe heat exchangers.

**Der Effekt von nicht kondensierbaren Gasen auf die Kondensation in einem separaten, geschlossenen Zwei-Phasen-Thermosyphon**

**Zusammenfassung.** Eine analytische Studie des Einflusses eines nicht kondensierbaren Gases auf die Dampfkondensation in einem separaten, geschlossenen Zwei-Phasen-Thermosyphon wird vorgestellt. Der merklich schädliche Einfluß wird beim Untersuchen der Kondensation von Dampf-Luft- und Wasserstoff-Dampf-Gemischen beobachtet. Es wird gezeigt, daß für die gleichen Arbeitsbedingungen des Thermosyphons der Einfluß von Wasserdampf auf die Kondensation beachtlicher ist, als der von Luft. Bei der Untersuchung von vier Arbeitstemperatur-Niveaus wurde herausgefunden, daß der Einfluß des nichtkondensierbaren Gases bei hohen Systemdrücken herausragend ist. Die errechneten Ergebnisse sollen bei der Auslegung von einem separaten geschlossenen Zwei-Phasen-Thermosyphon Verwendung finden.

**Nomenclature**

| | |
|---|---|
| $D$ | inner diameter of condenser section |
| $D_H$ | hydraulic diameter of vapor flow |
| $h_c$ | condensation heat transfer coefficient |
| $h_{fg}$ | latent heat of vapor |
| $f$ | friction factor |
| $f_0$ | friction factor in single-phase flow |
| $g$ | gravity |
| $k$ | thermal conductivity |
| $L_c$ | length of condenser |
| $p$ | pressure |
| $p_v$ | partial pressure of vapor |
| $q_c$ | heat flux |
| $Q$ | flowrate |
| $R$ | radius of condenser |
| $Re$ | Reynolds number of vapor-gas mixture |
| $Re_w$ | Reynolds number of wall suction, $v_w D/v_m$ |
| $Sc$ | Schmidt number |
| $Sh$ | Sherwood number |
| $Sh_0$ | Sherwood number without wall suction |
| $T$ | temperature |
| $u$ | velocity of liquid in $z$-direction |
| $v$ | velocity normal to wall surface |
| $w$ | mass fraction of noncondensable gas |
| $y$ | coordinate, see Fig. 2 |
| $z$ | axial distance |

*Greek symbols*

| | |
|---|---|
| $\delta$ | condensate film thickness |
| $\varepsilon$ | the factor defined in Eq. (24) |
| $\mu$ | viscosity |
| $v$ | kinetic viscosity |
| $\varrho$ | density |
| $\tau$ | shear stress |
| $\varphi_F$ | correction coefficient for shear stress |
| $\varphi_a$ | correction factor for mass transfer |
| $\omega$ | see Eq. (16) |

*Subscripts*

| | |
|---|---|
| $c$ | condensate; condenser |
| $i$ | interface of liquid-gas |
| $l$ | liquid |
| $m$ | vapor-gas mixture |
| $v$ | vapor |
| $w$ | wall |

## 1 Introduction

A separate type two-phase closed thermosyphon is a new kind of gravity-assisted wickless heat pipe. Different from the conventional heat pipe, the separate type heat pipe divides the evaporator section from the condenser section, so as to meet the special needs such as applications suitable for large scale plant, avoiding leaks between cold and hot fluids, heating and cooling several fluids at the same time, and facilitating long-distance heat transfer between cold and hot fluids etc. Figure 1 is a schematic illustration of this kind of heat pipe, in which vapor generated by heating the working

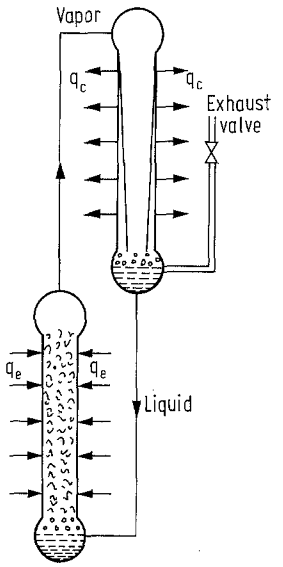


**Fig. 1.** Separate type two-phase closed thermosyphon

fluid in evaporator rises up into the top of condenser tube through auxiliary tubes, and condenses in the condenser section; the condensate returns to the evaporator section by gravity because the evaporator is always positioned below the condenser [1, 2].

One common troublesome problem in the performance of a conventional heat pipe is the presence of noncondensable gas which may come from system leaks, dissolved gas in the working fluid, and/or absorbed gas in the pipe structure. A gas plug may be formed in the condenser by the accumulation of noncondensable gas as vapor condensation proceeds [3], and this plug represents a diffusion barrier to the flowing vapor and almost completely shuts off that portion of the condenser. Because of this important detrimental feature, the effect of noncondensable gas on conventional heat pipe performance has been studied analytically [4] and experimentally [5].

Similarly, the presence of noncondensable gas in a separate type two-phase closed thermosyphon is detrimental to condensation. Unlike the condensation process in the conventional heat pipe, the same plug of noncondensable gas is in the exhaust valve of the separate type heat pipe as shown in Fig. 1, and doesn't affect condensation heat transfer much. But the noncondensable gas sweeping the condenser along with the vapor flow is still a diffusion resistance to condensing the flowing vapor onto the wall. Therefore, the detrimental effect is existing. Because of the needs for designs and operations of separate type heat pipe heat exchangers, and also because two-phase flow pattern is different and nocondensable gas effect behaves in the different manner in a separate type heat pipe as compared with those in a conventional heat pipe, the noncondensable gas effect on condensation in the separate type thermosyphon should be clarified. To the authors' knowledge, no work has been reported on this topic in the literature.

The noncondensable gas with major content in a separate type thermosyphon put in uses may vary from one to another on different operation stages. At the initial working stage of the thermosyphon, the noncondensable gas with main content is air because the thermosyphon is usually poorly or not vacuumed before the operation. While after the long-time operation of the thermosyphon, hydrogen produced by noncompatibility between water and steel pipe material becomes the main noncondensable gas existing in the thermosyphon. Thus, the present work establishes a theoretical model to analyze the effect of any noncondensable gas on condensation in a separate type two-phase closed thermosyphon, the results obtained and the cases examined, however, are only for water as the working fluid and air or hydrogen as the noncondensable gas. It is demonstrated that noncondensable gas has an appreciable effect on condensation heat transfer in the thermosyphon and on the wall temperature distribution along the condenser, and that particularly detrimental effect is due to hydrogen. In this regard, not considering the noncondensable gas effect is inadequate to describe the experimental characteritics of a separate type thermosyphon with small amounts of gas loaded, especially at low heat fluxes.

## 2 Physical model and formulations

The physical model consisted of a condenser with saturated vapor-gas mixture flowing vertically downward as shown in Fig. 2. The coordinates $(y, z)$ have been chosen for mixture gas flow and condensate film flow. The partial pressure of vapor in the system is assumed to be uniform axially and corresponds to the working temperature $T_v$ of fluid. The temperature at the interface of the liquid and the vapor-gas mixture is denoted by $T_i$ and the wall temperature by $T_w$. The concentration of noncondensable gas at the liquid-gas interface is $w_i$ and in the bulk of the mixture is $w$. Constant heat flux is extracted from the outside wall of the condenser.

On account of the different flow patterns of vapor and condensate liquid in the separate type heat pipe as compared with the conventional heat pipe, the noncondensable gas affects condensation heat transfer also in a different manner. Here in the condenser section of the separate type heat pipe, the interfacial temperature $T_i$ is made to be lower than $T_v$ by the presence of noncondensable gas when the vapor-gas mixture is sweeping through the entire condenser, and in such a way condensation heat transfer is reduced because of the decrease in thermal driving forces. In the end portion of the condenser, the noncondensable gas is concentrated more and more due to continuous vapor condensation, which causes the even much smaller interfacial temperature $T_i$ and corresponding reductions in heat transfer rates.

On the basis of the arguments presented in [6], in the region of engineering interest, the flow of liquid film is laminar and the flow of the uniform vapor-gas mixture is turbulent.

Following the Nusselt-type solution and considering the shear stress in the motion of condensate layer, we have con-

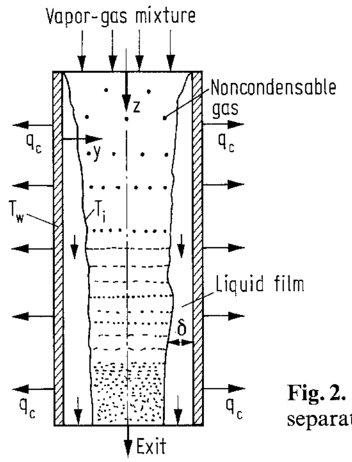


**Fig. 2.** Schematic of the condenser in a separate type closed thermosyphon

densate velocity expressed by:

$$u_l = \frac{1}{\nu_l}\left(-\frac{dp}{dz} - \varrho_l g\right)\left(\tfrac{1}{2}y^2 - \delta y\right) + \frac{\tau_i}{\nu_l}\, y, \tag{1}$$

where $\tau_i$ is interfacial shear stress in the flow direction at $y = \delta$, and is related to the axial pressure gradient by

$$-\frac{dp}{dz} = \frac{4\tau_i}{D_H} + \frac{d(\varrho_m u_m^2)}{dz} \tag{2}$$

if the body force on the vapor-gas mixture is neglected ($\varrho_m \ll \varrho_l$). $u_m$ is bulk velocity of the mixture and hydraulic diameter $D_H$ for the vapor-gas mixture flow is given by

$$D_H = D - 2\delta. \tag{3}$$

Equation (2) was derived from the cross-section averaged momentum equation on mixture flow, and the second term on RHS represents the pressure recovery from the vapor deceleration as condensation proceeds along the tube wall.

The interfacial shear stress $\tau_i$ is calculated locally based on the local mixture velocity,

$$\tau_i = \tfrac{1}{2}\varrho_m u_m^2 f, \tag{4}$$

where $f$ is the gas phase frictional factor at the liquid-gas interface, and should account for the effect of surface condensation mass flux similar to wall suction. Mickley [7] proposed the following expression

$$f = \frac{\varphi_F \exp(\varphi_F)}{\exp(\varphi_F) - 1} f_0, \tag{5}$$

where

$$\varphi_F = \frac{v_w/u_m}{f_0/2.0} \tag{6}$$

and $f_0$ is the gas phase friction factor for an impermeable tube wall, and given by the familiar correlation

$$f_0 = 0.079\, Re^{-1/4} \tag{7}$$

due to the turbulent flow of the mixture.

The liquid flow rate is then obtained by integrating the liquid velocity field over the liquid cross-sectional area, thus

$$Q_l = 2\pi \int_0^\delta \varrho_l (R - y)\, u_l\, dy = 2\pi \left(\frac{\tau_i}{\nu_l}\left(\frac{1}{2}R\delta^2 - \frac{1}{3}\delta^3\right) + \frac{1}{\nu_l}\left(-\frac{dp}{dz} + \varrho_l g\right)\left(\frac{1}{3}R\delta^3 - \frac{5}{25}\delta^4\right)\right). \tag{8}$$

The local film condensate rate on the wall is obtained by energy balance at the interface,

$$Q_c = 2\pi R\, q_c / h_{fg}, \tag{9}$$

where the sensible heat transferred from the mixture core is neglected.

Equation (9), which gives the condensate film mass flow rate on the wall per unit length of the tube, is now combined with Eq. (8), which gives the locally accumulated condensate flow rate. A mass balance in the axial direction reads

$$\frac{dQ_l}{dz} = Q_c \tag{10}$$

or

$$\frac{\tau_i}{\nu_l}\left(\frac{1}{2}R\delta^2 - \frac{1}{3}\delta^3\right) + \frac{1}{\nu_l}\left(-\frac{dp}{dz} + \varrho_l g\right)\left(\frac{1}{3}R\delta^3 - \frac{5}{24}\delta^4\right) = \frac{1}{2\pi} Q_l\Big|_z + \frac{R q_c}{h_{fg}}\Delta z. \tag{11}$$

This equation can be solved by the difference method with the boundary condition $Q_l = 0.0$ at $z = 0$ (mixture inlet). The calculation proceeds in a stepwise manner by first evaluating the film thickness $\delta$ for $z = \Delta z$ which, in turn is used to calculate the local liquid rate by Eq. (8). The local vapor flow rate is then corrected by accounting for the condensation rate. The new local values of $Q_v$, $D_H$, $dp/dz$ and $\tau_i$ are then evaluated to prepare the next calculation.

$T_i$ is equal to $T_v$ only where pure vapor condenses, while in the presence of noncondensable gas in vapor core, the interface temperature $T_i$ is always lower than the saturated temperature $T_v$ of the mixture corresponding to the local pressure, due to the resistance of vapor diffusion to the liquid-gas interface. Thus the determination of interfacial temperature $T_i$ relies on the diffusion process of noncondensable gas in the mixture bulk towards the interface, and mass transfer coefficient on the interface should be known before the calculation for condensation heat transfer of vapor-gas mixture. Dittus-Bolter equation [8]

$$Sh_0 = 0.023\, Re^{0.8}\, Sc^{0.4} \tag{12}$$

is a widely-used correlation to determine mass transfer coefficient in turbulent tube flow without wall suction. For the present problem, this equation should be corrected to account for the effect of condensation mass flux as follows:

$$Sh = \frac{\varphi_a \exp(\varphi_a)}{\exp(\varphi_a) - 1} Sh_0, \tag{13}$$

where the quantity $\varphi_a$ is defined as

$$\varphi_a = Re_w \, Sc / Sh_0 . \tag{14}$$

Such a form of correction was first derived by Ackermann [9], and recently has been successfully used in the studies on condensation of vapor-gas mixture in tubes and flat ducts [10–12].

Another relationship between $Sh$ and $Re_w$ can be found by the fact that the liquid-gas interface is impermeable for noncondensable gas, and thus given by the following,

$$Sh = Re_w \, Sc / (1 - \omega) \tag{15}$$

where

$$\omega = w / w_i \tag{16}$$

is the ratio of noncondensable gas content in the bulk of the vapor-gas mixture to that at the interface.

Combining Eqs. (13) and (15), one obtains

$$Sh = Sh_0 (-\ln \omega) / (1 - \omega) , \tag{17}$$

$$Re_w = Sh_0 (-\ln \omega) / Sc . \tag{18}$$

Naturally, the condensation film rate $Q_c$ is equal to the diffusion rate of vapors from the bulk of the mixture, so

$$Q_c = \pi \mu_m Re_w = \pi \mu_m Sh_0 (-\ln \omega) / Sc . \tag{19}$$

In the Eqs. (15)–(19), either $w_i$ or $w$ is calculated by the Gibbs-Dalton relation for perfect mixture,

$$w = (1 - p_v/p) / (1 - p_v/p (1 - M_v/M_g)) , \tag{20}$$

where $p_v$ represents the partial pressure of vapor, and is available from the tabulated properties of the vapor corresponding to local temperature, and $p$ is total pressure of system. $M_v$ and $M_g$ are the molecular weights of vapor and noncondensable gas, respectively. In each step of axial distance $z$, the interfacial temperature $T_i$ can be determined by the following procedure:

1. from the known $Q_c$ and Eq. (19), calculate $\omega$,
2. with $w$ evaluated from the last step calculation, combining the computed value of $\omega$, determine $p_i$ by Eq. (20), and then the interfacial temperature $T_i$ from the vapor table. The Nusselt solution for the liquid film gives

$$q_c = \frac{k_l (T_i - T_w)}{\delta} . \tag{21}$$

Solving $T_w$ from this equation yields

$$T_w = T_i - \frac{q_c \delta}{k_l} . \tag{22}$$

Having the wall temperature $T_w$, we can express the condensation heat transfer coefficient $h_c$ as follows:

$$h_c = q_c / (T_v - T_w) . \tag{23}$$

The factor $\varepsilon$ characterizing the noncondensable gas effect on condensation heat transfer is defined as

$$\varepsilon = \bar{h}_c / \bar{h}_{c,0} , \tag{24}$$

where the bar represents the mean value over the entire condenser and subscript 0 means absence of noncondensable gas in the heat pipe.

## 3 Results and discussion

In order to demonstrate the noncondensable gas effect on condensation heat transfer in separate type two-phase closed thermosyphons, two kinds of noncondensable gas, namely air and hydrogen, were examined using steam as the working fluid. The findings from these mixtures can be applied to other vapor-gas mixtures and similar systems. For steam-air mixture, $Sc = 0.55$, $M_v/M_g = 0.622$; for steam-$H_2$ mixture, $Sc = 0.32$, $M_v/M_g = 9.0$.

The condenser under study is 20 mm of inner diameter and 800 mm of length.

The wall temperature distribution along the condenser, predicted by using the current model, is shown in Figs. 3 and 4 for different gas loads, $w = 0.0$, 0.001, 0.01 and 0.05. It is seen that the wall temperature is very uniform axially if noncondensable gas is absent. This isothermality is one of the advantages heat pipes possess. However, by the presence of noncondensable gas the axial uniformity in the wall temperature no longer remains, and is greatly destroyed only with small amounts of noncondensable gas, say $w = 0.001$. The more amounts of noncondensable gas, the more nonuniform the wall temperature. The distinction between steam-air system and steam-$H_2$ system is clearly illustrated in both Figs. 3 and 4. For $Sc$ number of $H_2$ in steams is smaller than that of air, the concentration of $H_2$ on the liquid-gas interface is certainly lower than that of air with the same gas load in the mixture core. While on the other hand, the partial pressure of steam in steam-$H_2$ mixture on the interface would be much more reduced by the same gas concentration here than that in steam-air mixture because of the

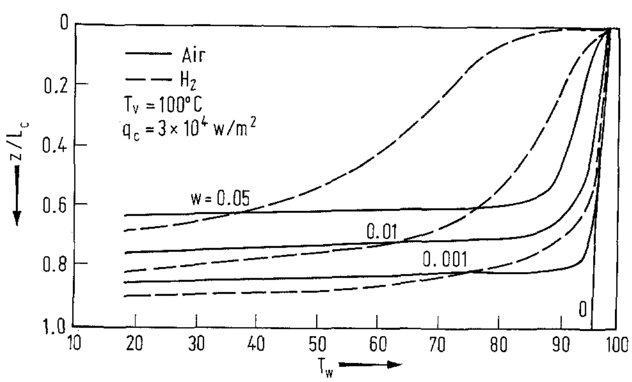


**Fig. 3.** Effect of noncondensable gas on wall temperature profile for small $q_c$

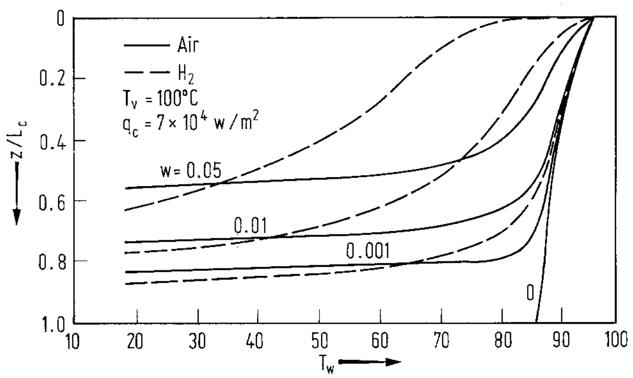


**Fig. 4.** Effect of noncondensable gas on wall temperature profile for large $q_c$

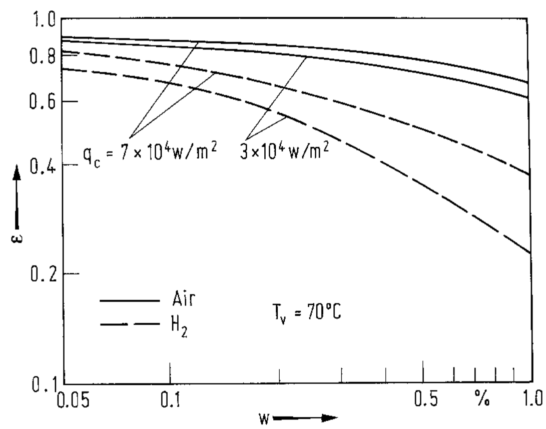


**Fig. 5.** Effect of noncondensable gas on condensation heat transfer in the separate type two-phase closed thermosyphon, $T_v = 70$ °C

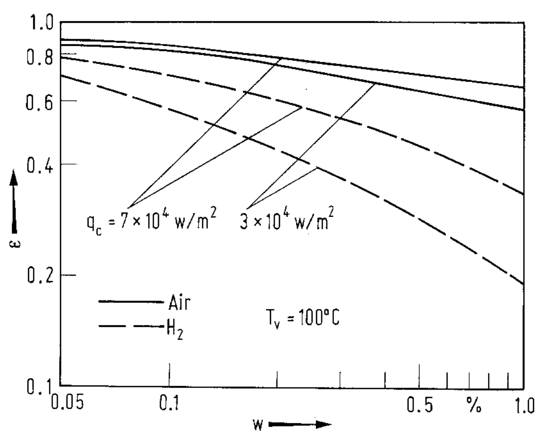


**Fig. 6.** Effect of noncondensable gas on condensation heat transfer in the separate type two-phase closed thermosyphon, $T_v = 100$ °C

larger ratio of molecular weights $M_v/M_g$ the steam-$H_2$ mixture has. Due to these, the effect of $H_2$ on the wall temperature is greater than that of air in the region of small axial distance, but is smaller in the region of large axial distance, as noted in Figs. 3 and 4. Comparing Fig. 3 with Fig. 4, we observe that the extent of the departure in the wall temperature due to the presence of noncondensable gas is greater for

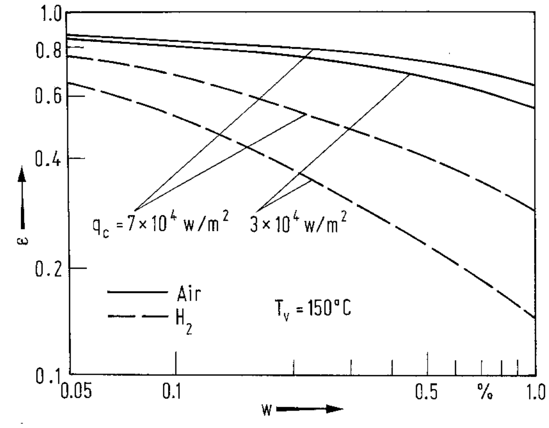


**Fig. 7.** Effect of noncondensable gas on condensation heat transfer in the separate type two-phase closed thermosyphon, $T_v = 150$ °C

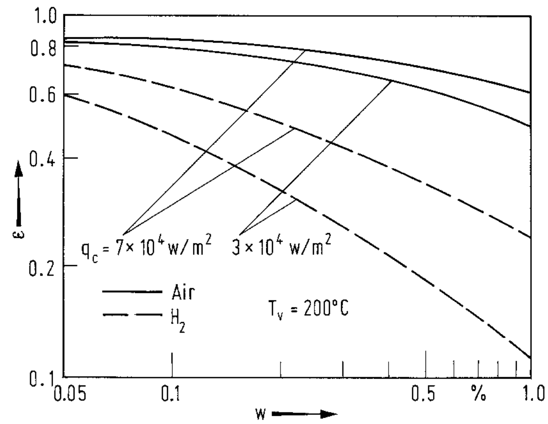


**Fig. 8.** Effect of noncondensable gas on condensation heat transfer in the separate type two-phase closed thermosyphon, $T_v = 200$ °C

larger heat fluxes, but this doesn't mean the noncondensable gas effect on condensation heat transfer is more appreciable at high heat loads, because the factor ε depends on the relative values of temperature differences. On the contrary, Figs. 5–8 showed that ε increases as heat flux is raised.

Figures 5–8 plot the factor ε defined in Eq. (24) vs. the gas load $w$ for different operating parameters and different noncondensable gases. The departure of these curves in Figs. 5–8 from unity indicates that the noncondensable gas has a detrimental effect on condensation heat transfer. The first finding from these figures is that the noncondensable gas effect is slighter at higher heat load. This is because high inlet velocity of vapor resulted from high heat load decreases the diffusion resistance to condensing vapors onto the wall. Secondly, we discovered that the effect of hydrogen on steam condensation is much more appreciable than that of air, especially for high gas loads, which we can anticipate from Figs. 3 and 4. The least and most striking finding is that the noncondensable gas effect is accentuated with the increase of working temperature $T_v$ of vapor, namely with the higher system pressure. This conclusion which is valid for both hydrogen and air is out of our expectation, and is contrary to the results obtained by previous studies [11–14] involving noncondensable gas effect on condensation heat transfer. It is believed that the reason for this contradiction is due to the different boundary conditions specified in the present study and in the previous papers, that is to say, under the constant heat flux boundary condition, condensation heat transfer is more insensitive to a noncondensable gas at low system pressures, while is more sensitive under the isothermal boundary condition. This finding, which is perhaps of great importance for engineering applications, will wait for the further fundamental theoretical studies and experimental verifications.

Maybe as interesting as the aforementioned general trends are the actual magnitudes of the reductions in condensation heat transfer. For a very small amount of $H_2$, say $w = 0.05\%$, the heat transfer is decreased by more than 20% at the moderate temperature levels and heat loads. With the same amount of air in the separate type thermosyphon the reduction in heat transfer would be more than 13% for the operating parameters of engineering interest. These figures remind us that the noncondensable gas effect on condensation heat transfer shouldn't be neglected in designs and operations of separate type heat pipe heat exchangers, especially when noncondensable gas is likely present in the heat pipes.

## 4 Conclusions

The noncondensable gas effect on vapor condensation in a separate type two-phase closed thermosyphon has been investigated analytically. The results obtained by using steam-air and steam-$H_2$ mixtures lead to the following conclusions:

1. The axial uniformity of wall temperature was greatly destroyed by the presence of noncondensable gas. The more the gas amount, the more non-uniform the wall temperature profile. This nonuniformity of temperature is slighter with air as the noncondensable gas than that with hydrogen.

2. The detrimental effect of noncondensable gas on condensation heat transfer is appreciable and cannot be neglected. Under common operating conditions and with only 0.05% amount, the reduction in heat transfer exceeds 20% and 13% by hydrogen and air, respectively.

3. The noncondensable gas effect is more remarkable for low heat fluxes with the constant gas load.

4. Condensation of steams is much more sensitive to hydrogen than to air with the same mass concentration of gases in the separate type thermosyphons.

5. For both hydrogen and air, the noncondensable gas effect is accentuated at high working temperatures, i.e. high system pressures.

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