EVALUATION OF CORRELATIONS FOR THE PREDICTION OF HEAT TRANSFER IN A HEAT PIPE FOR SOLAR COOLING APPLICATIONS

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ABSTRACT

This paper describes the theoretical-experimental study on the heat transfer inside a wickless heat pipe to be used in solar collectors for solar cooling systems. When opportunely developed, solar cooling, that is the production of cold out of solar heat, allows to obtain considerable energy conservation compared to conventional plants. The solar energy activates a thermodynamic cycle for the production of cooled water or for air treatment intended to room conditioning or refrigeration processes. In this regard, for the construction of solar panels, an empty heat pipe has been considered because of the high efficiency of heat transfer inside it. Such pipe is able to transfer a large quantity of heat over relatively large distances with small temperature differences between the heat input and output zones.

Further to a series of tests on tubes inclined at 45°, with different filling levels of distilled water (FR = 8.57%÷57.1%), for different levels of input heat into the evaporator section (between 50 and 155 W) and mass flow rate of cooling fluid between 3.0 and 8.5 kg/h, a study has been performed for searching correlations in the open literature that would allow to predict the values of heat transfer coefficients at the evaporator and at the condenser. A positive match was founded between the values predicted by some of the numerous correlations available in literature and the data experimentally recorded. This paper consequently suggests an empirical correlation for the condenser section.

1. INTRODUCTION

Heat pipe, meant as thermal duct, is a heat transfer device, characterized by high thermal conductivity which is based on the principle of evaporation and condensation of a working fluid introduced into a metal container. In its simplest configuration, the so-called wickless, the heat pipe consists of a closed tube of thermally conductive metal, such as copper or aluminum, in which the vacuum has been made and then filled with a small amount of working fluid. The rest of the tube is filled by the vapour of liquid so that there are no other gases.

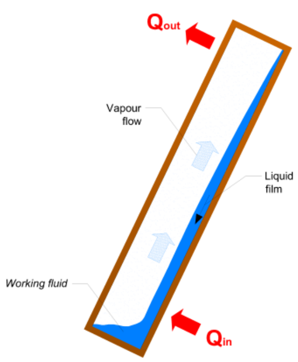

Fig. 1 – Operating principle of an inclined wickless heat pipe.

The hot end, when heated by a generic heat source, releases heat to the liquid contained in the device, which vaporizes and thus provokes an increase in the steam pressure inside the pipe. In addition, the latent heat of vaporization absorbed by the liquid decreases the temperature of the hot end of duct. Since the vapour pressure near the hot end of tube is higher than the condensation one at the cold end, such difference in pressure causes a very quick transfer of vapour toward the cold end, where the vapor exceeding the equilibrium condenses and releases heat to the cold end. The pipe is vacuum because if it contained other non-condensable gas, the latter may slow down the above mentioned vapour movement and thus make the heat transfer less efficient. Being high the latent heat of vaporization, it is possible to transfer large amounts of heat with small temperature differences between the two ends of the pipe. A limit of the wickless heat pipe, also known in literature as *two-phase closed thermosyphon*, is that, in order that the condense returns to the evaporator due to gravity, the evaporator must necessarily be placed at the lowest point. Anyway the simple construction scheme, the low thermal resistance, the high efficiency and low manufacturing costs make the heat pipe a device that is widely used in several applications fields: heat exchangers, cooling of electronic components, solar energy conversion systems, cooling of gas turbine rotor blades, etc.

2. REVIEW OF PREVIOUS WORKS

A considerable amount of both theoretical and experimental studies has been conducted on the heat transfer performance of a wickless heat pipe in several applications. Such studies attempted to determine the effects that the variation of a series of quantities which are peculiar of a heat pipe has on the heat transfer performance.

Nguyen-Chi et al. [1] experimentally investigated the performance of a vertical wickless heat pipe using water as working fluid. Their study focused on the influence of operating parameters on the maximum performance either by dry-out or burn-out limits. Li et al. [2] experimentally investigated the steady-state heat transfer characteristics of a vertical two-phase closed thermosyphon at low temperature differences with water, R11 and R22 as working fluids. Shiraishi et al. [3] made an experimental study on the critical heat transfer rate in thermosyphons by taking into account the aspect ratio, filling ratio, working fluid property and operating pressure. Noie [4] experimentally analyzed, among the various parameters that can affect the performance of a heat pipe, the effects linked to variations in input heat rate ($100 < Q < 900$ W), in the working fluid filling ratio ($30\% \leq FR \leq 90\%$) and in the evaporator length. Amornkitbumrung et al. [5] studied the effect of the inclination angle on the heat transfer rate of a copper thermosyphon filled with water. They concluded that the highest heat transfer rate occurs at 22.5◦ with a filling ratio of 30%. Wang et al. [6] studied condensation heat transfer inside vertical and inclined thermosyphons. They found that the inclination angle of a thermosyphon has an important influence on the condensation coefficient, and the optimum inclination angle varies with liquid filling from 20◦ to 50◦ .Negishi and Sawada [7] made an experimental study on the heat transfer performance of an inclined two-phase closed thermosyphon using water and ethanol as working fluids. Lee and Mital [8] performed an experimental study on the heat transfer performance of a two-phase closed thermosyphon while also developing a relatively simple theoretical analysis for its maximum heat transfer capacity. The study investigated the effects of parameters such as the amount of working fluid, the ratio of heated length to cooled length, the operating pressure, the input heat flux and working fluid were investigated.

The wickless heat pipes reported in literature have larger diameters than those examined in the present study. In this regard, the study conducted by Jouhara et al. [9] – which used a copper thermosyphon with 200 mm length, 6 mm inner diameter and 12 mm outer diameter - proves to be interesting.

3. EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental campaign has been conducted through the testing system named T.O.S.C.A. (Thermal-fluid dynamics Of Solar Cooling Apparatus), which has been designed and constructed in the "Thermal-Fluid Dynamics Applied to Energy Systems" laboratory of ENEA-Casaccia (Rome). A schematic diagram of the system is shown in Fig. 2. The pipes which were used for the tests had been made of copper with inner diameter of 6mm and outer diameter of 8 mm. The total length is 1424 mm. The degree of vacuum created inside the pipe before the filling is 10^{-5} bar. The inclination angle is 45°. The heater is constituted by an electric resistance with wire wound round a copper jacket (length 1240 mm, inner diameter 9 mm, thickness 1.5 mm) in which the heat pipe is inserted.

The heater supply is conveyed through a VARIAC that allows to modulate the supplied power and to simulate the variation of the solar radiation during the entire span of the solar cycle. The heat removal occurs by means of a cooling circuit with demineralized water lap against the outer surface at the top of the heat pipe.

Then, for its entire length, the heat pipe is wound by an insulating layer of 30 mm-thick fiberglass. A further insulating layer is made of 15 mm-thick expanded polyurethane. The insulating system is completed by an aluminum foil reinforced by fiberglass; such foil has proven useful for reducing the losses due to irradiation toward the environment.

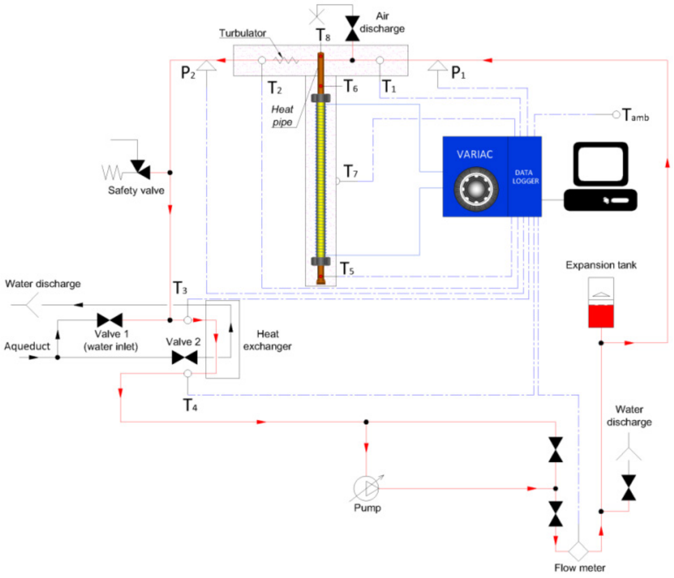

Fig. 2 – Schematic diagram of the experimental apparatus.

In all, 9 Cr/Al thermocouples (K type with isolated hot joint) have been used for measuring the water temperature at the interesting points of the cooling circuit (T_{01}~T_{04}), the ambient temperature (T_{amb}) the temperature of the heat pipe walls (T_{05}~T_{08}) and two pressure transducers (p_1~p_2) for measuring respectively the collector input and output pressure.

A series of tests has been executed with filling ratio varying between 8.57% and 57.1% for two different levels of supplied thermal power: 70 and 155 W. Later on, for the pipe with FR=22.8%, tests were carried out with power variable from 50 to 155 W and afterwards with mass flow rate of the coolant variable from 3.0 to 8.5 kg/h while keeping the other working parameters constant (Q_{in}=70 W).

4. CORRELATIONS

The heat transfer inside the heat pipe depends on a complex phase change process that takes place within the regions of the evaporator and of the condenser; such process can be further complicated by the counter-flow of the liquid and vapour phases.

4.1 Evaporator

The evaporator section turns out as the most complex and least understood part of the heat pipe, because within such section the effects of pool boiling in the lower region add up to those of laminar convection or boiling within the continuous liquid film (the latter shows significantly higher heat transfer coefficients). These phenomena heavily depend on a series of parameters such as filling ratio, inclination angle, thermal fluxes, etc. For this reason, it is not easy to predict the value of the heat transfer coefficient through simple formulas. The general procedure involves the application of empirical or semi-empirical correlations. The data that were recorded during the experiments brought to the hypothesis that the process of heat transfer in the film is negligible in comparison to the pool boiling mechanism. Tests have been carried out on the following correlations, which are available in literature, for the prediction of the heat transfer coefficient of the evaporator when the liquid pool boiling is the main mechanism.

Correlation by Imura et al. [10].

$$h_e = 0.32\left(\frac{\rho_l^{0.65}k_l^{0.3}c_{p,l}^{0.7}g^{0.2}}{\rho_v^{0.25}h_{lv}^{0.4}\mu_l^{0.1}}\right)\left(\frac{p_{sat}}{p_{atm}}\right)^{0.3}q^{0.4} \quad (1)$$

Correlation by Rohsenow [11].

$$h_e = \left(\frac{q}{h_{lv}}\right)^{0.67}\left[\mu_l/\sqrt{\frac{\sigma}{g(\rho_l-\rho_v)}}\right]^{0.33}\frac{c_{p,l}}{C_{sf}}\frac{1}{Pr^{1.7}} \quad (2)$$

where the value of the experimental constant C_{sf} is not univocally defined in literature and it is variable within a fairly wide range according to the references in literature [12]. It was determined $C_{sf} = 0.0063$ with the will of minimizing the RMS error of all the performed tests.

Correlation by Shiraishi et al. [13].

$$h_e = 0.32\left(\frac{\rho_l^{0.65}k_l^{0.3}c_{p,l}^{0.7}g^{0.2}}{\rho_v^{0.25}h_{lv}^{0.4}\mu_l^{0.1}}\right)\left(\frac{p_{sat}}{p_{atm}}\right)^{0.23}q^{0.4} \quad (3)$$

Correlation by Kutateladze [14].

$$h_e = 0.44\left(\frac{k_l}{L_b}\right)\left(\frac{1\times10^{-4}qp}{gh_{lv}\rho_v\mu_l}\frac{\rho_l}{\rho_l-\rho_v}\right)^{0.7}Pr_l^{0.35} \quad (4)$$

Correlation by Labuntsov [15].

$$h_e = 0.075\left[1+10\left(\frac{\rho_v}{\rho_l-\rho_v}\right)^{0.67}\right]\left(\frac{k_l^2}{\nu_l\sigma(T_{sat}+273.15)}\right)^{0.33}q^{0.67} \quad (5)$$

4.2 Condenser

Depending on operating conditions and the pipe inclination angle, it is possible to identify different flow regimes of liquid film. In this regard, it is useful to begin by defining the Reynolds number of liquid film:

$$Re_f = \frac{w_l\delta_l}{\nu_l} = \frac{Q}{(\pi D_i\mu_l h_{lv})} \quad (6)$$

For approximate values of $Re_f < 5$ the motion of film is laminar with a smooth vapour-side interface such that heat is transferred by pure heat conduction across the film. For values of $Re_f > 5$, the effects of viscosity and smoothing effects by surface tension are no longer able to suppress wavy structures. In this case, the heat transfer across the film is improved by the wave formation due to a reduction of the effective thickness of the film between the wave crests.

If thickness and velocity are further increased, or if viscosity is decreased, the film flow becomes turbulent and the heat transfer mechanism is now due to turbulent exchange processes which are much higher than the molecular heat conduction.

Here are illustrated the most used correlations for condensation heat transfer inside a thermosyphon available in open literature.

Correlation by Nusselt [16]. For very low velocity of the vapour, the motion of liquid film is laminar and it is plausible to apply the Nusselt's correlation developed to predict the heat transfer coefficient of the laminar filmwise condensation on a vertical plate.

$$h_c = 0.943\left\{\frac{\rho_l g k_l^3(\rho_l-\rho_v)[h_{lv}+0.68c_{p,l}(T_v-T_c)]}{\mu_l L_c(T_v-T_c)}\right\}^{1/4} \quad (6)$$

otherwise expressed in the dimensionless form in which appears the so-called modified Nusselt number.

$$Nu_c^* = \frac{h_c}{k_l}\left[\frac{\nu_l^2}{g}\left(\frac{\rho_l}{\rho_l-\rho_v}\right)\right]^{1/3} = 0.925\,Re_f^{-1/3} \quad (7)$$

Correlation by Hassan and Jakob [17]. Applying Nusselt's assumptions, Hassan and Jakob developed the following expression of Nu* for an infinitely long inclined tube:

$$Nu_c^* = 0.651Re^{-1/3}\left[\frac{D_i}{(L\sin\varphi)}\right]^{-1/3} \quad (8)$$

which, applied to finite tube length, presents an error less than 10% for values of $(\tan\varphi\, L/D_i) > 2.5$.

Correlations by Uehara et al. [18]. Taking into account the inclination effect, the modified Reynolds number [19] is defined as:

$$Re_\varphi = Re_f\, f_\varphi \quad (9)$$

with $f_\varphi = 2.87\frac{D_i}{(L\sin\varphi)}$ and $f_\varphi = 1$ for vertical tubes ($\varphi = 0°$).

In the laminar-wavy range, in which the presence of waves on the film is no longer negligible ($2 < Re_\varphi < 1333 \cdot Pr_l^{-0.96}$), the literature offers the correlation provided by Uehara et al. for the modified Nusselt number:

$$Nu_c^* = 0.884Re_\varphi^{-1/4}\ . \quad (10)$$

They also suggested a different expression for the turbulent range ($Re_\varphi > 1333 \cdot Pr_l^{-0.96}$):

$$Nu_c^* = 0.044Pr_l^{2/5}Re_\varphi^{1/6}\ . \quad (11)$$

For the experimental tests performed in this study, the values of Re_φ are such that the flow regime was never turbulent, so that only Eq. 10, valid in the laminar-wavy range, has been tested.

Correlation by Gross [19]. Gross developed a correlation valid for different flow regimes aimed at predicting the modified Nusselt number Nu^* through a combination of Eq. 8 valid for laminar flow regime and Eq. 11 valid for turbulent regime:

$$Nu_c^* = \left[\left(f_p Nu_{HJ}^*\right)^2 + (Nu_{Ue}^*)^2\right]^{1/2} \quad (12)$$

with $f_p = 1/\left(1-0.63\,p^{*3.3}\right)$ and $p^* = p/p_{cr}$.

An empirical correlation has been developed to predict the experimental data.

Present correlation. Starting from an equation with a form similar to Nusselt's dimensionless correlation($Nu_c^* = A\,Re_f^{\,B}$), a value of the constants A and B was found that would minimize the RMS error between experimental data and predicted ones. The suggested equation is the following:

$$Nu_c^* = 0.058\,Re_f^{\,0.54} \quad (13)$$

Clearing the heat transfer coefficient of the condenser, the empirical correlation obtained is:

$$h_c = 0.058\, Re_f^{\,0.54} k_l \left[\frac{\nu_l^2}{g}\left(\frac{\rho_l}{\rho_l-\rho_v}\right)\right]^{-1/3} \tag{14}$$

5. COMPARISON OF RESULTS

The values predicted by the correlations were compared with the corresponding experimental values of heat transfer coefficients of both the evaporator and the condenser, respectively:

$$h_e = \frac{Q}{S_e(T_e-T_v)} = \frac{Q}{\pi D_i L_e (T_e-T_v)} \tag{15}$$

$$h_c = \frac{Q}{S_c(T_v-T_c)} = \frac{Q}{\pi D_i L_c (T_v-T_c)} \tag{16}$$

5.1 Evaporator

During the tests performed by varying the FR, as shown in Figs. 3-4, experimental data showed good matches with Eq. 2 and partially with Eq. 1.

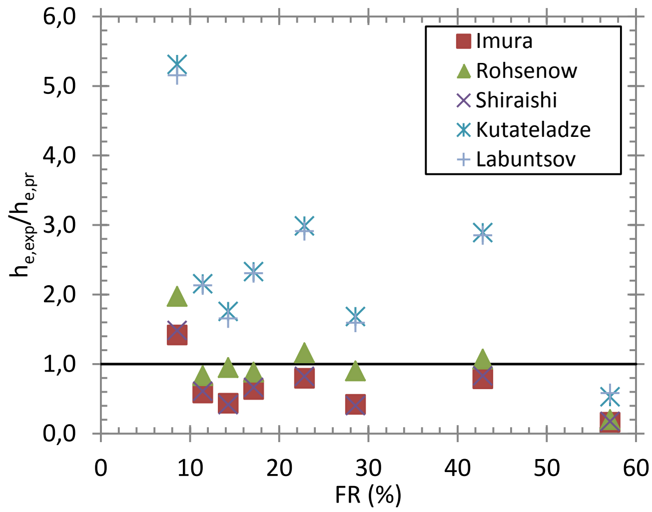

Fig. 3 – Ratio of experimental values of h_e to predicted values versus FR. Q_{in}= 70 W. G≅7.37 kg/h.

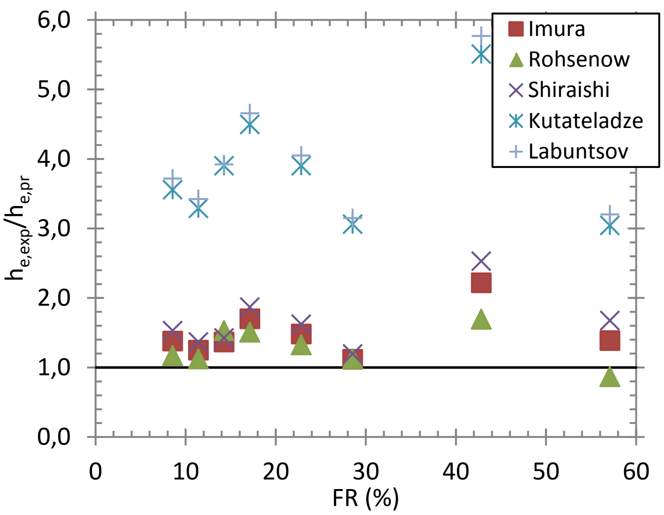

Fig. 4 – Ratio of experimental values of h_e to predicted values versus FR. Q_{in}= 155 W. G≅7.37 kg/h.

The values predicted by Rohsenow's correlation show a mean error ranging from a minimum of 5.1% to a maximum of 49.3% (an exception was found for the test with FR= 57.1% and Q_{in}=70 W where the mean error was 397.2 %).

A good approximation was also obtained for series of tests performed both by varying the input heat rate for the value of FR=22.8%, whose data are plotted in Fig. 5, and by varying the mass flow rate of coolant (Fig. 6).

Imura's correlation (Eq. 1) tends to overestimate the experimental data of h_e for the input heat rate of 70 W and to underestimate them with input heat rate of 155 W (Figs. 3-4).

Shiraishi's correlation (Eq. 3) shows values very close to those predicted by Imura's correlation and therefore the same considerations are valid.

The values predicted by Kutateladze (Eq. 4) and Kruzhilin (Eq. 5) significantly underestimate those experimentally observed. However, these deviations can be predictable if one considers that these correlations were not developed to predict heat transfer in a heat pipe but on a horizontal plate.

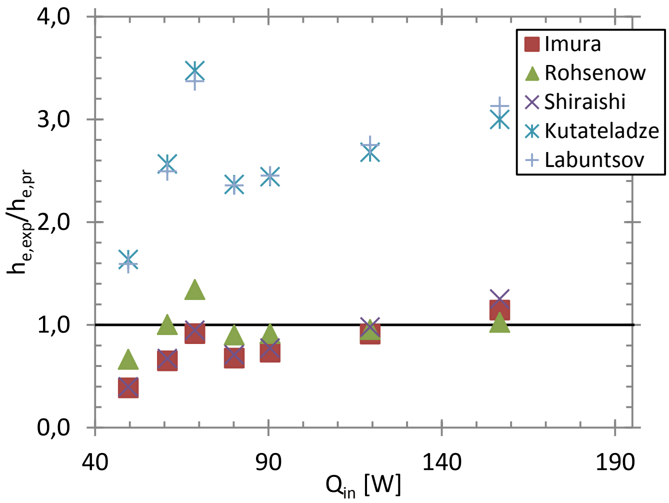

Fig. 5 – Ratio of experimental values of h_e to predicted values versus Q_{in}. FR= 22.8%. G≅7.37 kg/h.

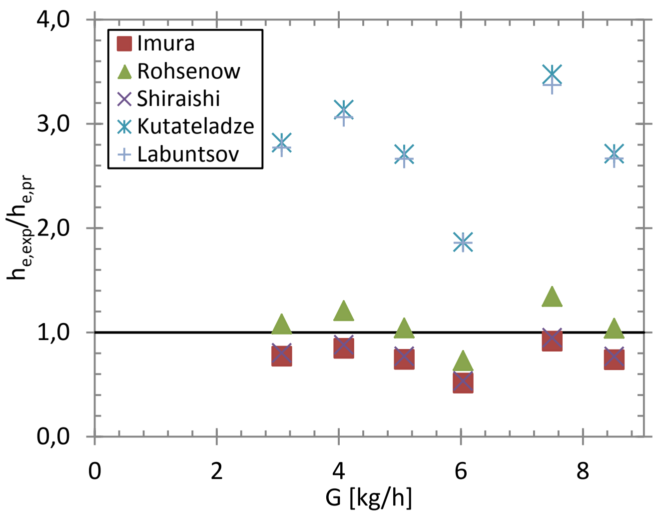

Fig. 6 – Ratio of experimental values of h_e to predicted values versus G. FR= 22.8%. Q_{in}=70 W.

5.2 Condenser

The ratio of experimental values to corresponding predicted values of heat transfer coefficient of the condenser has been plotted as a function of the following parameters changed during the series of tests: the filling ratio (Figs. 7-8), input heat rate (Fig. 9) and mass flow rate of coolant (Fig. 10)

The correlations in literature for predicting the heat transfer coefficient of a heat pipe condenser did not result reasonably valid. As shown in Fig. 7, for the tests in which the filling ratio has been changed, while the input heat rate was set at 70 W, the predicted values were significantly higher than the corresponding experimental values. Only in the case of filling ratios of 14.3% and 28.6% a good match was obtained between experimental values of h_c and predicted ones, even resulting almost coincident if using Eq. 8 or Eq. 12.

The proposed correlation well approximates the experimental data for this series of tests, however a high predictive error occurs at filling ratios of 14.3% and 28.6%.

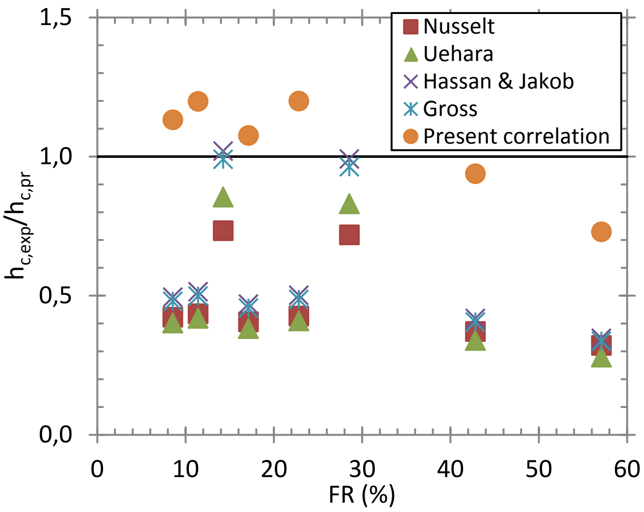

Fig. 7 – Ratio of experimental values of h_c to predicted values versus FR. Q_{in}= 70 W. G≅7.37 kg/h.

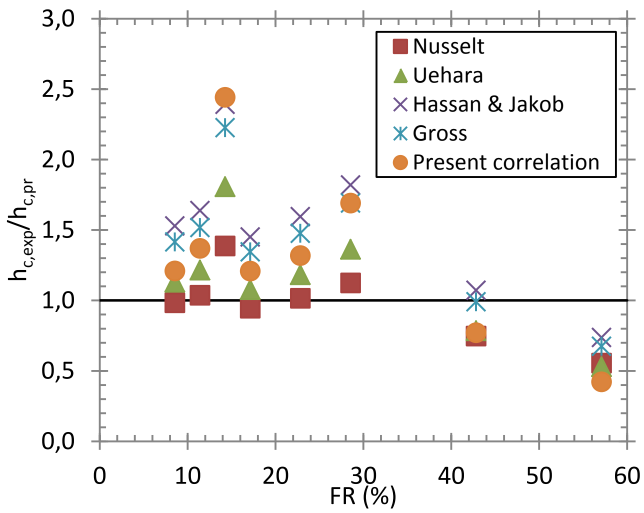

Fig. 8 – Ratio of experimental values of h_c to predicted values versus FR. Q_{in}= 155 W. G≅7.37 kg/h.

In Fig. 8 the experimental tests for studying the effect of filling ratio for an input heat rate of 155 W show that the most accurate predictions of heat transfer coefficient of the condenser are obtained when using the Nusselt's correlation (Eq. 6). The predicted values are in reasonable agreement with experimental ones except for the filling ratio of 57.1% where the error is no longer negligible.

The proposed correlation (Eq. 14), although minimizing the RMS error of all experimental tests, shows for this series of tests predicted values less accurate than those obtained by Eq. 6. However, Eq. 14 allows to obtain predicted values of h_c with a mean error less than 30% except for the filling ratios of 14.3%, 28.6% and 57.1%.

By varying the input heat rate, the correlations in literature agree with experimental values only for heat rate in the range of 120-150 W (Fig. 9). Equation 14 well approximates the values of the heat transfer coefficient of the condenser, obtained by varying the input heat rate, and shows a mean error that ranges from a minimum of 4.4% for Q_{in}=50 W to a maximum of 29.8 % for Q_{in}=155 W.

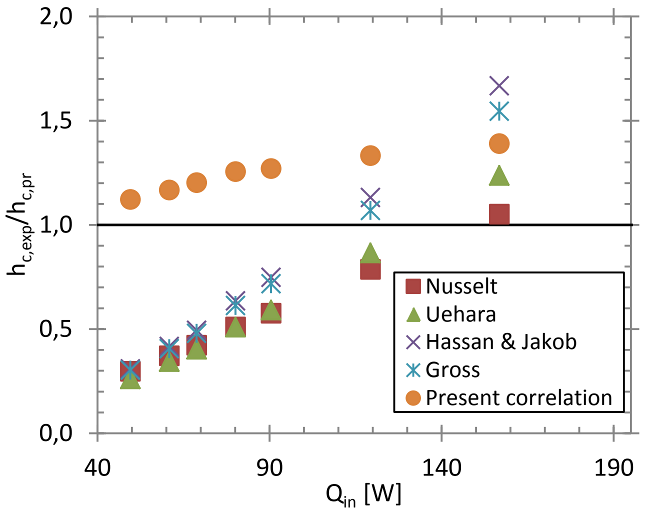

Fig. 9 – Ratio of experimental values of h_c to predicted values versus Q_{in}. FR= 22.8%. G≅7.37 kg/h.

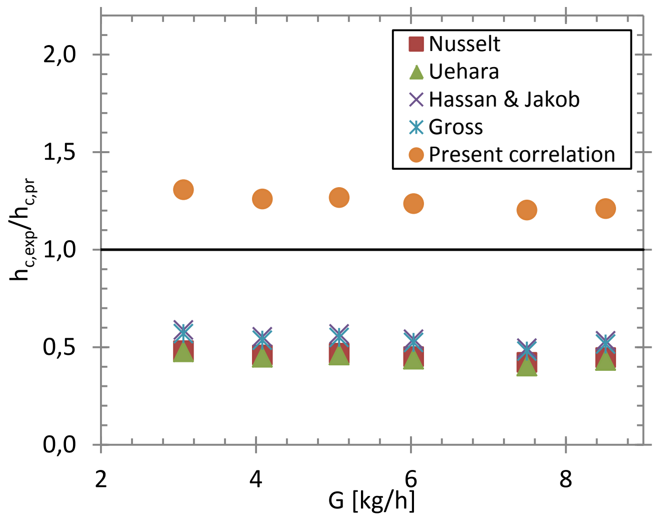

Fig. 10 – Ratio of experimental values of h_c to predicted values versus G. FR= 22.8%. Q_{in}=70 W.

Finally, analyzing the data obtained by varying only the mass flow rate of coolant (Fig. 10), there is a trend of approximately constant values of h_c both experimental and predicted. The correlations developed by Nusselt, Uehara, Hassan & Jakob and Gross tend to overestimate the experimental values with mean errors approximately of 50%. Instead, Eq. 14 predicts values of the heat transfer coefficient slightly lower but still in a range of mean error of 13.1-20.5 %.

6. CONCLUSIONS

The present study attempted to provide guidance to the ability of the main predictive models available in literature to predict the values and trends of the experimental heat transfer coefficients of the evaporator and the condenser of a heat pipe.

The most accurate predictions of the heat transfer coefficient of the evaporator were obtained through the Rohsenow's correlation, which, after a careful evaluation of the experimental coefficient C_{sf}, allowed to obtain an RMS error equal to 22.8% for all the tests performed.

With regard to the condenser section, the proposed correlation (Eq. 14) allowed to estimate the values of the heat transfer coefficient of the condenser with a RMS error of 36.0% for all the tests performed.

NOMENCLATURE

Symbol	Quantity	SI Unit
AR	aspect ratio, L_{ev}/D_i	dimensionless
c_p	specific heat	J/(kg K)
D	diameter	m
FR	filling ratio, V_l/V_{ev}	dimensionless
g	gravitational acceleration	m/s^2
G	mass flow rate of water	kg/h
h	heat transfer coefficient	W/(m^{2}K)
h_{lv}	latent heat of vaporization	J/kg
k	thermal conductivity	W/(m K)
L	length	m
L_b	Laplace constant, $[\sigma/g(\rho_l-\rho_v)]^{1/2}$	m
Nu	Nusselt number, hD/k	dimensionless
p	pressure	Pa
Pr	Prandtl number, $\mu c_p/k$	dimensionless
q	heat flux	W/m^2
Q	heat transfer rate	W
Re	Reynolds number	dimensionless
S	surface	m^2
T	temperature	°C
w	velocity	m/s
μ	dynamic viscosity	N s/m^2
ν	cinematic viscosity	m^2/s
ρ	density	kg/m^3
σ	surface tension	N/m
φ	inclination angle	°

Subscripts

c	condenser section
cr	critic
e	evaporator section
exp	experimental
f	film
in	input into the evaporation section
l	liquid
out	transmitted from the condenser section
pr	predicted
sat	saturation
v	vapour

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