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Local and instantaneous distribution of heat transfer rates through wavy films ☆

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Abstract

Heat transfer through a laminar-wavy falling silicon oil film on a vertical plate has been investigated. The film flows down an electrically heated metal foil which delivers a constant heat flux. The temperature field at the backside of the foil is measured by a very sensitive infrared camera with high temporal resolution. Local and instantaneous heat transfer rates through the film are evaluated from the temporal development of the local wall temperature. Investigations in two-dimensional waves show the influence of the Prandtl number on the transport processes even in the laminar region. The experimental data confirm the results of numerical calculations with a spectral element method. Furthermore, the temporal averaged heat transfer has been investigated in thermally developed three-dimensional wavy films over a film Reynolds number range of 10–129 and a Prandtl number range of 10–45. The Prandtl number dependency of the heat transfer in the laminar-wavy region of the film agrees well with a recently published experimental study.

Keywords: Wavy film; Heat transfer; Nusselt number; Prandtl number

1. Introduction

Film flows can be found in a variety of engineering applications, e.g., condensers, evaporators and reactors. Design and operation of these apparatus require a quantitative knowledge of the momentum, energy and mass transfer in the liquid film phase, since its transport resistance frequently restricts the performance of the apparatus.

One of the earliest studies on film flows is that of Nusselt [1] who investigated a vertical laminar condensate film with smooth surface. However, it can be observed that waves propagate even on laminar liquid films at low film Reynolds number. The instability of film flows against slightest perturbations has been shown analytically by stability analyses. Several experimental studies on flow dynamics of falling films reveal that the flow conditions can be classified into five states [28]. At film Reynolds numbers $Re < 0.47 Fi^{1/10}$ the film surface is smooth and the Nusselt assumptions are valid. If the flow exceeds after a first transient regime $0.47 Fi^{1/10} < Re < 2.2 Fi^{1/10}$ the flow becomes wavy but is still laminar, therefore this region $2.2 Fi^{1/10} < Re < 75$ is denoted as stable laminar-wavy. After a second transition regime at about $75 < Re < 400$ where singular turbulent spots are induced increasingly by interacting laminar waves the flow can be regarded as turbulent for $Re > 400$.

Due to the technical relevance of film flows a variety of investigations have been performed to analyze heat and mass transfer [2–16] of falling films. It was observed that both, heat and mass transfer in wavy films are significantly higher than those predicted by the Nusselt solution. In these studies semi-empirical correlations of the heat and mass transfer enhancement have been presented, however, the mechanisms of the transport processes and the flow characteristics of wavy films have not been fully explained.

Apart from analytical approaches numerical simulation techniques have been applied with promising results. Renz and Gromoll [17] showed the influence of the film waves on the turbulence characteristics of the gas/vapor flow. Ramadan [18] published numerical predicted heat and mass transfer rates from turbulent films solving the parabolic con-

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Nomenclature

b	foil width	m
c	specific heat	$\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$
f	frequency	$\mathrm{s^{-1}}$
g	acceleration of gravity	$\mathrm{m \cdot s^{-2}}$
l_h	length of the heating section	m
$\dot{m}$	mass flow rate	$\mathrm{kg \cdot s^{-1}}$
Nu	Nusselt number $= \alpha \cdot \lambda^{-1} \cdot (\nu^2 \cdot g^{-1})^{1/3}$	
Pr	Prandtl number $= \eta \cdot c \cdot \lambda^{-1}$	
$\dot{q}''$	heat flux	$\mathrm{W \cdot m^{-2}}$
Re	Reynolds number $= \dot{m} \cdot \eta^{-1} \cdot b^{-1}$	
s	foil thickness	m
T	temperature	K
t	time	s
u	stream wise velocity	$\mathrm{m \cdot s^{-1}}$
x	stream wise co-ordinate	m

Greek symbols

α	heat transfer coefficient	$\mathrm{W \cdot m^{-2} \cdot K^{-1}}$
δ	film thickness	m
η	dynamic viscosity	$\mathrm{kg \cdot m^{-1} \cdot s^{-1}}$
λ	thermal conductivity	$\mathrm{W \cdot m^{-1} \cdot K^{-1}}$
ν	kinematic viscosity	$\mathrm{m^2 \cdot s^{-1}}$
ρ	density	$\mathrm{kg \cdot m^{-3}}$
σ	surface tension	$\mathrm{N \cdot m^{-1}}$
$\dot{\Phi}'''$	volumetric heat source	$\mathrm{W \cdot m^{-3}}$

Subscripts

0	initial value
Am	ambient
Ex	excitation
M	mean value
S	surface
W	wall
bW	backside of the wall
fW	forefront of the wall

servation equations. Bach and Villadsen [19] solved the Navier–Stokes equations of a film flow for Reynolds numbers $Re \leqslant 25$ employing a finite element technique. In their calculations a gravitational wave with a preceding capillary wave arises from an initial Gaussian perturbation which agrees with experimental observations and is confirmed by further studies, e.g., Kheshgi and Scriven [20], and Ho and Patera [21]. Stuhlträger et al. [22,23] found by finite difference calculations that heat transfer through the waves is affected by convection. Jayanti and Hewitt [24], who assumed the wave form in their calculations, obtained roll waves, i.e., a recirculation zone in a moving coordinate system, if the wave amplitude exceeds a certain value. However, the influence of this recirculating flow on heat transfer is negligible in their results. They ascribe the amplification of heat transfer to the effective film thinning caused by the waves. Miyara [25] solved the Navier–Stokes equations and the energy equation simultaneously for a Reynolds number of $Re = 100$ without restrictive assumptions. He obtained gravitational roll waves with preceding capillary waves. His investigation of heat transfer shows that the heat transfer is enhanced by effective film thinning and by convection. The dominating process depends on the Prandtl number. This was confirmed by Adomeit and Leefken [26] who further separated convection from conductive heat transfer. Hence, they could show that the influence of convection is dominant in the waves particularly in the roll wave, where a bulk of surface-tempered fluid is accumulated.

Due to the lack of experimental data the validity of these models cannot be verified. Accurate measurements with high temporal and spatial resolution are needed to prove the present conclusions and to develop more exact models. In this study the local heat transfer coefficients of two and three-dimensional instantaneous laminar-wavy films are determined experimentally using a highly sensitive infrared thermography.

2. Experimental work

2.1. Experimental apparatus

The closed-loop test facility is shown in Fig. 1. Liquid silicon oil film ($T_0 = T_{\mathrm{Am}}$) flows down vertically on a planar constantan foil (length 700 mm, width 240 mm, thickness 0.025 mm). An adjustable gap in the liquid distributor allows to set the initial film thickness. The foil is heated electrically at 190 W. The effective area of the heated section was determined with the infrared camera which leads to a heat flux from the foil to the film of 6000 $\mathrm{W \cdot m^{-2}}$. At the end of the test section the film flows into a reservoir equipped with heating and cooling devices to control the fluid temperature. The film fluid is circulated by a piston pump of adjustable flow rate which is measured by a positive displacement flow meter. The local mean temperature of the film is measured with two thermocouples (1 mm diameter), which are immersed in the film in a row.

To create two-dimensional waves the film flow can be perturbed in an exactly defined mode by a loudspeaker located above the liquid distributor. Silicon oils (DMS) with different viscosity have been employed as testing fluid. Since the heat conductivity is roughly the same for all silicon oils the Prandtl numbers vary with the viscosity. Three different oils with Prandtl numbers $Pr = 10$, $Pr = 25$ and $Pr = 45$ have been used, see Table 1.

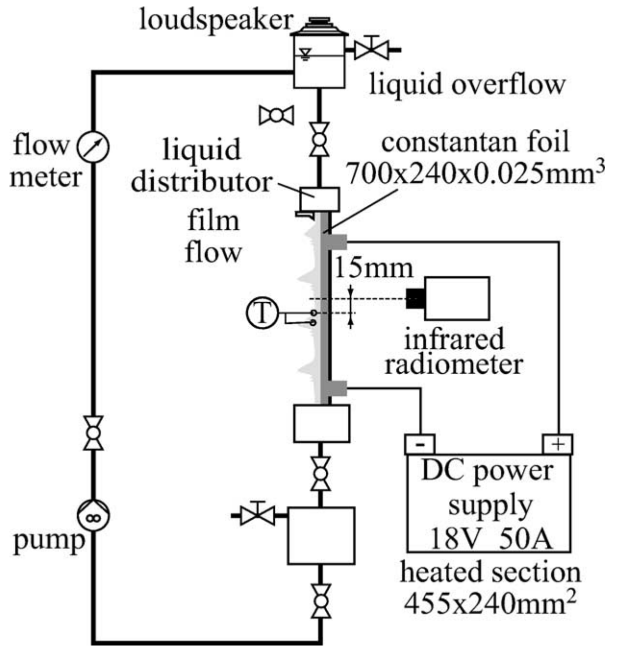

Fig. 1. Schematic diagram of the experimental facility.

Table 1
Silicon oils used, operating conditions and range of key variables

Fluid	Re	Pr	η 10^{-3} Pa·s	ρ kg·m^{-3}	λ W·m^{-1}·K^{-1}	σ N·m^{-1}
DMS-T02	38–129	10	1.88	870	0.26	0.0187
DMS-T05	14–71	25	4.75	915	0.28	0.0197
DMS-T11	10–35	45	10.02	933	0.32	0.0201

2.2. Infrared thermography

In order to measure the local heat transfer the liquid film is heated with a constant wall heat flux provided by the electrically heated constantan foil. Due to film waves the wall temperature will vary spatially and temporally depending on the local flow condition. The local wall temperature is measured on the backside of the foil by an infrared camera. This technique was suggested and performed by Hetsroni and Rozenblit [27]. Fig. 2 shows an exemplary measured time regime of the temperature at the backside which is coated with paint which is black within the camera's spectral range of 9–11 µm wavelength. The temperature oscillations caused by the fluctuating heat transfer through the film is about 100 mK, which is in the order of the camera's resolution of 25 mK. The amplitude of the noise signal could be reduced to approx. 10 mK by averaging the data spatially over an area of 0.25 mm by 0.50 mm (8 pixels). The data are recorded at a frame rate of 2 480 Hz.

The temperature T_{bW} measured on the backside of the foil is not exactly the desired temperature on the forefront T_{fW}. The underlying transient heat conduction problem was solved numerically to estimate the difference between T_{bW} and T_{fW}. The boundary condition at forefront of the foil was

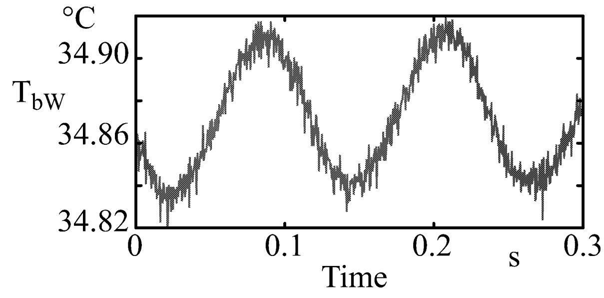

Fig. 2. Exemplary measured wall temperature T_{bW} at the backside of the foil.

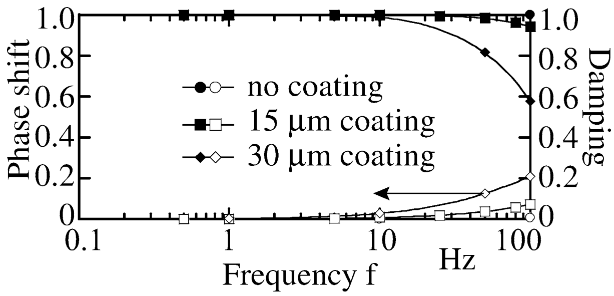

Fig. 3. Phase shift $\Delta t \cdot f$ and damping $T_{\mathrm{bW}}/T_{\mathrm{fW}}$ of the temperature T_{bW} at the backside of the foil referred to the temperature T_{fW} at the forefront if the heat transfer at the forefront oscillates with the frequency f.

assumed to be a sinusoidal oscillating heat transfer caused by the liquid film. The heat transfer at the backside caused by radiation and free convection is assumed to be constant.

The mean thickness of the coated paint, which was determined by a displacement meter is about 15 µm. Fig. 3 shows the dimensionless phase shift $\Delta t \cdot f$ of T_{bW} referred to T_{fW} and the damping $T_{\mathrm{bW}}/T_{\mathrm{fW}}$ at different frequencies f of the heat transfer at the forefront. The response of the 15 µm coating is sufficiently accurate over the whole frequency range. The comparison with 30 µm coating and no coating is given to show the influence of the coating on the response T_{bW}. For a 30 µm coating the measured temperatures T_{bW} will significantly differ from the temperature T_{fW} if the heat transfer oscillation exceeds 10 Hz. Hence, it is strongly recommended to keep the coating as thin as possible. The influence of the foil itself (no coating) is negligible.

The local heat transfer coefficient α is defined with the local wall temperature T_{fW} and the local mean temperature of the film T_{M}:

$$\alpha \equiv \frac{\dot{q}''_{\mathrm{W}}}{T_{\mathrm{fW}} - T_{\mathrm{M}}} \tag{1}$$

and a Nusselt number Nu based on the heat transfer coefficient defined by Eq. (1)

$$Nu = \alpha \left(\frac{\nu^2}{g}\right)^{1/3} \frac{1}{\lambda} = \frac{\dot{q}''_{\mathrm{W}}}{T_{\mathrm{fW}} - T_{\mathrm{M}}} \left(\frac{\nu^2}{g}\right)^{1/3} \frac{1}{\lambda} = 1.43\, Re^{-1/3}$$

$$\dot{q}''_{\mathrm{W}} = \text{const} \tag{2}$$

The local wall heat flux $\dot{q}''_{\mathrm{W}}$ is calculated by the locally supplied heat source less the unsteady heating of the foil:

$$\dot{q}''_{\mathrm{W}} = s\dot{\Phi}''' - \rho_{\mathrm{W}} c_{\mathrm{W}} s \frac{\mathrm{d}T_{\mathrm{W}}}{\mathrm{d}t} \quad (3)$$

where s is the thickness of the foil and $\dot{\Phi}'''$ is the electrically supplied volumetric heat source, ρ_{W} and c_{W} are the density and the heat capacity, respectively of the constantan foil. Eq. (3) is derived from an integral energy balance of the foil, assuming an uniform foil temperature T_{W} which was validated by comparing Eq. (3) with a numerical solution of the governing energy equation. The heat transfer by radiation and free convection on the backside of the foil is neglected since it is less than 0.1% of the heat transfer through the liquid film.

3. Results

In most studies experimental data are compared to the Nusselt solution to show the heat and mass transfer enhancement caused by the waves. Usually, the Nusselt solution for condensation is used where the heat transfer coefficient is defined with a temperature difference between the wall temperature T_{W} and the temperature of the film surface T_{S}. In this work the heat transfer coefficient is defined with the wall temperature and the mean temperature of the liquid film T_{M}.

3.1. Thermal development of the film

The liquid entering the heating section is cold and has to be heated before it reaches its thermally developed state. From Eq. (1) it can be seen that the low mean temperature of the liquid and the higher temperature of the wall at the beginning of the heated region will lead to higher temperature difference and therefore to a lower local heat transfer coefficient as the heat source is constant. Afterwards both, the fluid and the wall temperature increase. At a certain distance the temperature difference between wall and medium and hence the temporally averaged local heat transfer reach constant values. The film flow can be regarded as thermally developed even though both temperatures further rise. It has been shown by experimental investigations of Wilke [3] and Bays and McAdams [2] that the length of the developing region is related to the product of Prandtl and Reynolds number. This has been reviewed by Seban and Faghri [8]. The developing length has been investigated in the present study for a liquid of Prandtl number $Pr = 25$ and Reynolds number of $Re = 40$ and $Re = 71$. A Reynolds number of $Re = 71$ leads to the highest product $Re \cdot Pr$ of the observed cases. The total length of the heating section is about 455 mm; the local heat transfer has been measured at 5 different positions. Fig. 4 shows the evaluated Nusselt number along the heated section for $Pr = 25$ at $Re = 40$ and $Re = 71$.

As expected, the Nusselt number reaches its constant value earlier for lower Reynolds numbers. The developing

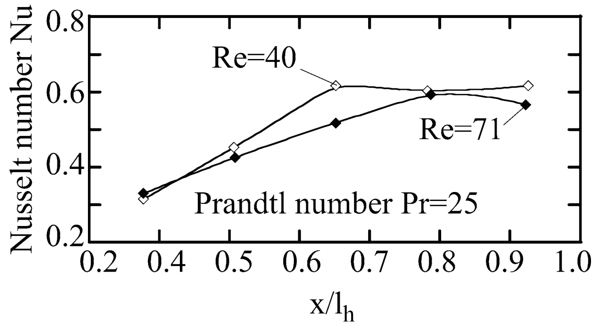

Fig. 4. Nusselt number at different lengths x/l_{h}.

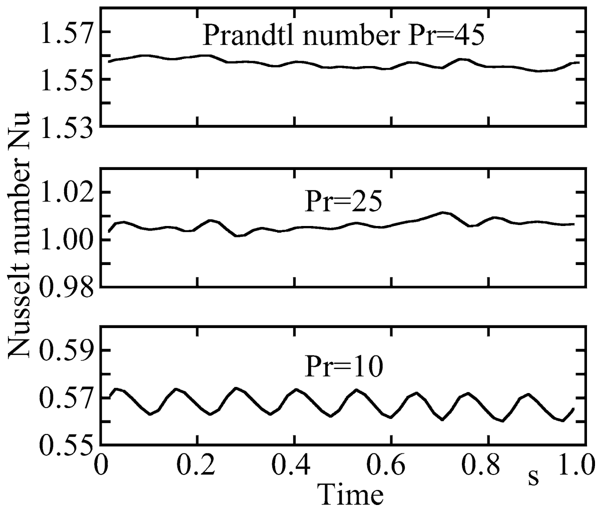

Fig. 5. Temporal evolution of the local Nusselt number of varying Prandtl number in two-dimensional laminar-wavy films of $Re = 26$ at $x/l_{\mathrm{h}} = 0.1$.

region for the wavy film exceed the correlation of Gimbutis [29] for smooth liquid films by a factor of 3 to 4.

3.2. Two-dimensional film flows

Two-dimensional waves can only be found in a small region at the beginning of the film. Investigations of Nakoryakov and Alekseenko [28] show that two-dimensional waves can reach a stationary state if they are excited in a limited range of frequencies which depends on the liquid properties and on the flow rate. In the present investigations the film flow has been excited by a loudspeaker to generate stable two-dimensional waves. However, the waves can only be realized in a region where the flow is not thermally developed. This means that the temporally averaged local Nusselt number cannot be compared with the Nusselt solution. Fig. 5 plots the temporal evolution of the local Nusselt numbers at a Reynolds number of $Re = 26$ for different Prandtl numbers.

Nevertheless, the analysis of the local heat transfer can give insight to the heat transfer processes and to the quality of the infrared thermography technique. The excitation frequency f_{Ex} of 8–10 Hz is exactly reflected by the oscillation of the Nusselt number which is shown in particular by the

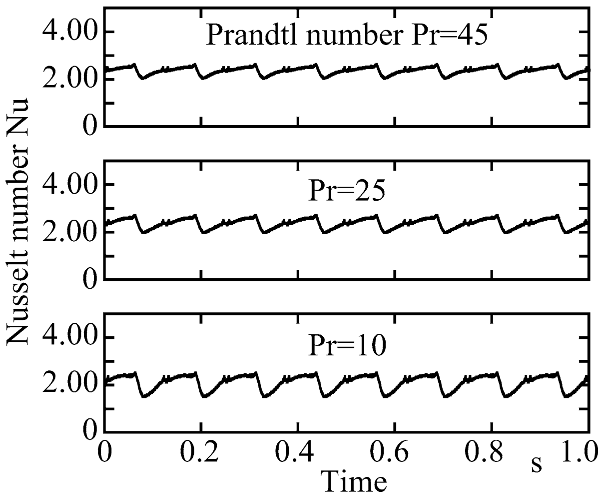

Fig. 6. Numerical calculation of the temporal local Nusselt number for a film with two-dimensional waves at different Prandtl numbers and $Re = 8$.

curve of $Pr = 10$. This indicates the eminent sensitivity and high temporal resolution of the applied infrared thermography. Moreover, the amplitude of the oscillating Nusselt number decreases with increasing Prandtl number. This effect confirms the numerical results of Adomeit and Leefken [26] for film condensation: If the Prandtl number is low heat conduction dominates the heat transfer. The heat transfer by convection which increases with increasing Prandtl number is dominant in the waves and therefore smoothes the overall heat transfer rate. A numerical result is depicted in Fig. 6, which shows the calculated local heat transfer coefficient in a two-dimensional wave for different Prandtl numbers.

3.3. *Three-dimensional film flows*

In laminar-wavy film flows three-dimensional waves are arising due to slightest system immanent disturbances. Therefore, the external excitation with the loudspeaker is not needed.

The local wall temperature is measured in a region where the film flow can be regarded as thermally developed. The heat transfer coefficient is evaluated according to Eq. (2) and time averaged.

Fig. 7 shows the Nusselt number using different silicon oils as a function of the Reynolds number. Due to the limited power of the piston pump the maximum Reynolds number for the most viscous oil ($Pr = 45$) is limited to $Re = 35$. The plots show a strong dependency of the Nusselt number on the Prandtl number. The heat transfer increases with increasing Prandtl number and the Nusselt solution is exceeded by a factor up to 1.3. Alhusseini et al. [10] recently published similar experimental results from thermocouple measurements for condensation. Their correlation implies a strong effect of the Prandtl number. However, due to the different definition of the Nusselt number their correlation cannot be compared quantitatively with the present results.

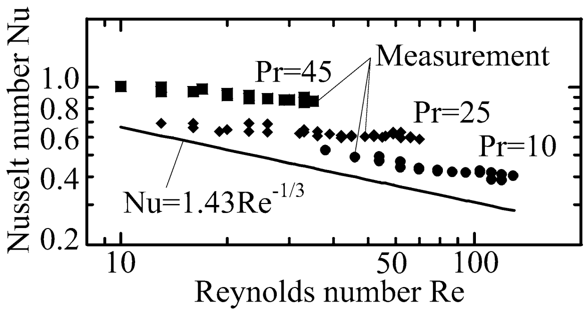

Fig. 7. Measured Nusselt numbers for silicone oils of different Prandtl number compared with the Nusselt solution *Nu*.

4. Conclusion

Heat transfer of laminar-wavy films has been investigated by an infrared thermography technique with very high temporal resolution.

The results for two-dimensional waves confirm own numerical prediction, that the heat transfer by convection increases with increasing Prandtl number. Since convection is high in regions where heat conduction is low the increasing convection leads to a smoothed temporal course of the total heat transfer.

The evaluated heat transfer coefficient in three-dimensional waves is time averaged as a time-dependent local heat transfer can hardly be interpreted without information about instantaneous flow condition. The experimental data show a clear dependence of the Nusselt number on the Prandtl number. Alhusseini et al. [10] published recently a correlation for the Nusselt number which shows a significant effect of the Prandtl number supporting the present findings.

In future work the instantaneous local film thickness and heat transfer will be measured simultaneously to gain a deeper insight to the influence of flow dynamics on heat transfer and to verify the present models.

The temporal resolution of the instantaneous local heat transfer has to be improved by taking the heat transfer through the foil and the coating into account. Efforts will be made to solve the underlying inverse problem.

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References

[1] W. Nusselt, Die Oberflächenkondensation des Wasserdampfes, Z. VDI 60 (1916) 541–546.
[2] G. Bays, W. McAdams, Heat transfer coefficients in falling film heaters: Streamline flow, Ind. Engrg. Chem. 29 (1937) 1240–1246.
[3] W. Wilke, Wärmeübergang an Rieselfilme, VDI Forschungsheft 490 (1962) 1–36.

[4] U. Renz, H. Vollmert, Der Wärme- und Stoffaustausch bei der Verdampfung eines Brennstoff-Films in einem heißen Gasstrom, Forschung im Ingenieurwesen 40 (1974) 96–101.

[5] W.P. Jones, U. Renz, Condensation from a turbulent stream onto a vertical surface, Internat. J. Heat Mass Transfer 17 (1974) 1019–1028.

[6] U. Renz, Die partielle Filmkondensation aus laminaren und turbulenten Grenzschichtströmungen, Lett. Heat Mass Transfer 2 (1975) 9–12.

[7] K. Cun, R. Seban, Heat transfer to evaporating liquid films, J. Heat Transfer 91 (1971) 391–396.

[8] R.A. Seban, A. Faghri, Wave effects on the transport to falling laminar liquid films, J. Heat Transfer 100 (1978) 143–147.

[9] S.S. Kutadeladze, I.I. Gogonin, Heat transfer of slowly moving vapour, Internat. J. Heat Mass Transfer 22 (1979) 1593–1599.

[10] K. Alhusseini, M. Tuzla, J.C. Chen, Falling film evaporation of single component liquids, Internat. J. Heat Mass Transfer 41 (1998) 1623–1632.

[11] S. Banerjee, E. Rhodes, D.S. Scott, Mass transfer to falling wavy liquid films at low Reynolds numbers, Chem. Engrg. Sci. 22 (1967) 43–48.

[12] R. Gregorig, J. Kern, K. Turek, Improved correlation of film condensation data based on a more rigorous application of similarity parameters, Wärme- und Stoffübertragung 7 (1974) 1–13.

[13] W.H. Henstock, T.J. Hanratty, Gas absorption by a liquid layer flowing on the wall of a pipe, AIChE J. 25 (1) (1978) 122–131.

[14] I. Mudawar, R.A. Houpt, Measurement of mass and momentum transport in wavy-laminar falling liquid film, Internat. J. Heat Mass Transfer 36 (1993) 4151–4162.

[15] T.D. Karapantsios, M. Kostoglou, A.J. Karabelas, Local condensation rates of steam-air mixtures in direct contact with a falling film, Internat. J. Heat Mass Transfer 38 (1995) 779–794.

[16] P.N. Yoshimura, T. Nosoko, T. Nagata, Enhancement of mass transfer into a falling laminar liquid film by two-dimensional surface waves—some experimental observations and modeling, Chem. Engrg. Sci. 51 (1996) 1231–1240.

[17] U. Renz, B. Gromoll, Der Einfluß der Welligkeit von Rieselfilmen auf den Wärme- und Stoffaustausch an der Phasengrenze, Chem. Ing. Tech. 48 (1976) 483.

[18] F. Ramadan, Zur Berechnung des Wärme- und Stoffaustausches bei der Filmkondensation reiner Dämpfe im durchströmten Rohr, Dissertation, RWTH Aachen, 1983.

[19] P. Bach, J. Villadsen, Simulation of the vertical flow of a thin, wavy film using a finite-element method, Internat. J. Heat Mass Transfer 27 (6) (1984) 815–827.

[20] H.S. Kheshgi, L.E. Scriven, Disturbed film flow on a vertical plate, Phys. Fluids 30 (1987) 990–997.

[21] L.W. Ho, A.T. Patera, A Legendre spectral element method for simulation of unsteady incompressible viscous free-surface flows, Comp. Methods Appl. Mech. Engrg. 80 (1990) 355–366.

[22] E. Stuhlträger, Y. Naridomi, A. Miyara, H. Uehara, Flow dynamics and heat transfer of a condensate film on a vertical wall—I. Numerical analysis and flow dynamics, Internat. J. Heat Mass Transfer 36 (1993) 1677–1686.

[23] E. Stuhlträger, A. Miyara, H. Uehara, Flow dynamics and heat transfer of a condensate film on a vertical wall—II. Flow dynamics and heat transfer, Internat. J. Heat Mass Transfer 38 (1995) 2715–2722.

[24] S. Jayanti, G.F. Hewitt, Hydrodynamics and heat transfer of wavy thin film flow, Internat. J. Heat Mass Transfer 40 (1) (1997) 179–190.

[25] A. Miyara, Numerical analysis on flow dynamics and heat transfer of falling liquid films with interfacial waves, Heat Mass Transfer 35 (1999) 298–306.

[26] P. Adomeit, A. Leefken, U. Renz, Experimental and numerical investigations on wavy films, in: E.W.P. Hahne, W. Heidemann, K. Spindler (Eds.), Proceedings of the 3rd European Thermal Science Conference, Edizioni ETS Pisa, Heidelberg, 2000, pp. 1003–1009.

[27] G. Hetsroni, R. Rozenblit, Heat transfer to a liquid-solid mixture in a flume, Internat. J. Multiphase Flow 20 (1994) 671–689.

[28] S.V. Alekseenko, V.E. Nakoryakov, B.G. Pokusaev, Wave Flow of Liquid Films, Begell House, New York, 1994.

[29] G. Gimbutis, Heat Transfer to a Falling Fluid Film, Mokslas, Vilnius, 1988.