Int. J. Heat Mass Transfer. Vol. 35, No. 3, pp. 683–689, 1992
Printed in Great Britain

0017-9310/92 $5.00+0.00
Pergamon Press plc

Heat transfer and film thickness during condensation of steam flowing at high velocity in a vertical pipe†

R. BELLINGHAUSEN and U. RENZ

Lehrstuhl für Wärmeübertragung und Klimatechnik, RWTH Aachen, 5100 Aachen, Germany

Abstract—In the present study the heat transfer of pure steam flowing downward in a vertical condenser pipe is determined and the thickness of the wavy film is measured using a laser absorption method. The measured data are compared to numerical predictions based on the solution of the conservation equations for mass, momentum and energy for the vapor and liquid phases. The k–ε turbulence model of Jones and Launder is used for both phases. It will be shown that the transition from laminar to turbulent film flow can be predicted reasonably well if the turbulent kinetic energy in the film is conserved to a minimum value.

1. INTRODUCTION

THE HEAT transfer in condenser pipes is normally dominated by the heat transfer resistance of the liquid film. Therefore, the film thickness is one of the most important properties to predict the local heat flux. For laminar films the well known Nusselt theory [1] provides a relation between Nusselt number and a dimensionless film thickness. This relation is valid for smooth films at low vapor velocities:

$$Nu_F = 1/\delta^* \tag{1}$$

where the dimensionless film thickness δ^* is defined by $\delta^* = \delta/L$ with $L = (\nu_F^2/g)^{1/3}$ and the film Nusselt number by $Nu_F = h_F L/\lambda_F$.

With increasing film thickness, which corresponds to an increasing film Reynolds number defined by

$$Re_F = \rho_F u_F \delta/\eta_F = \dot{m}_F/\pi d\eta_F, \tag{2}$$

waves appear. Above a critical Reynolds number, which depends on the Prandtl number and the shear stress at the interface, the film becomes turbulent. In that flow regime the film surface is very rough, resulting in a three-dimensional, highly unsteady film flow for which an exact solution of the conservation equations is impossible. A first numerical approach has been published by Wasden and Dukler [2], who calculated a two-dimensional wavy film surface based on a measured film thickness.

In the present paper measurements of heat transfer and film thickness are described. These results are compared to predictions based on a numerical solution of the two-dimensional turbulent conservation equations of both phases.

† Dedicated to Professor Dr.-Ing. Dr.-Ing.e.h. Ulrich Grigull.

2. EXPERIMENTAL SET-UP

2.1. *Condenser pipe*

The experimental set-up is presented in Fig. 1. The test section consists of two concentric pipes. The steam flows downward inside the inner pipe, whereas in the annulus cooling water is flowing upward. The temperatures of the inner wall and of the cooling water are measured along the pipe by 11 thermocouples, each with an outer diameter of 0.5 mm (see Fig. 1(a)). To check the symmetry of the flows each temperature is measured at two different positions. The mass flow rates and the inlet and outlet temperatures of the vapor, condensate, and the cooling water are measured. From these data the heat fluxes and hence the heat transfer coefficients can be evaluated along the tube.

2.2. *Measurement of film thickness*

At the end of the condenser tube the copper pipe is replaced by a quartz glass section where the film thickness is measured using an optical method (see Fig. 2). The measuring system is an improved version of the system proposed by Sattelmayer *et al.* [3]. An infrared laser beam ($\lambda = 1523$ nm) enters the condenser pipe through a quartz glass cylinder and is focussed onto the surface of the film at the opposite wall. The beam traverses the film and is caught by a set of lenses and sent to the photomultipliers by a fiber to measure the beam intensity. To monitor the intensity of the incoming beam a chopper is used to send the laser beam directly to the detectors periodically. Reflection and diffraction effects are corrected by simultaneously using a second laser beam with a wavelength of $\lambda = 632.8$ nm. This red laser light is not absorbed by the liquid but undergoes reflection and diffraction like the infrared laser beam.

The film thickness can be determined by

NOMENCLATURE

c_η	constant of the turbulence model
d	pipe diameter
f_η	constant of the turbulence model
f_R	correction factor of the red laser beam
f_{IR}	correction factor of the infrared laser beam
g	gravitational acceleration
h_F	heat transfer coefficient
$\hat{h}$	total enthalpy
I_R	intensity of the red laser beam
I_{IR}	intensity of the infrared laser beam
$I_{R,0}$	reference intensity of the red laser beam
$I_{IR,0}$	reference intensity of the infrared laser beam
$I_{R,0,\delta=0}$	reference intensity of the red laser beam without film
$I_{IR,0,\delta=0}$	reference intensity of the infrared laser beam without film
k	kinetic energy of turbulence
k'	absorption coefficient
L	characteristic length
$\dot{m}_F$	mass flow of condensate
$\dot{m}_{V,in}$	mass flow of vapor at the inlet
Nu_F	film Nusselt number
p	pressure
Pr	Prandtl number
Pr_t	turbulent Prandtl number
r	radial coordinate
Re_F	film Reynolds number
$Re_{V,in}$	vapor Reynolds number at pipe inlet
$Re_{V,x}$	local vapor Reynolds number
S_Φ, S_p, S_t	source terms
u	velocity in x-direction
u_F	film velocity in x-direction
$u_{V,in}$	vapor velocity at the inlet
v	velocity in r-direction
x	axial coordinate.

Greek symbols

δ	film thickness
δ^*	dimensionless film thickness
ε	dissipation of kinetic energy of turbulence
η	dynamic viscosity
η_{eff}	effective dynamic viscosity
η_t	turbulent viscosity
λ	heat conductivity
ν	kinematic viscosity
ρ	density
ρ_F	density of the film
σ	general Prandtl number
σ_t	general turbulent Prandtl number
Φ	general variable.

$$\delta = (k')^{-1} \ln [(I_R I_{IR,0})/(I_{R,0} I_{IR}) * (f_{IR}/f_R)]. \quad (3)$$

The correction factors for the infrared and the red beam f_{IR} and f_R are taken from a measurement without film

$$f_{IR} = I_{IR,\delta=0}/I_{IR,0,\delta=0} \quad \text{and} \quad f_R = I_{R,\delta=0}/I_{R,0,\delta=0}. \quad (4)$$

The absorption coefficient of water has been taken as $k' = 1.67\ \text{mm}^{-1}$.

The film thickness at the measuring location can be varied by changing the mass flow rate of the cooling water and keeping the steam velocity constant.

3. NUMERICAL APPROACH

The flow of the steam and the condensate film can be described by a set of conservation equations for mass, momentum and energy. For the turbulent flow regimes the low-Reynolds k,ε-model of Jones and Launder [4, 5] is used for both the vapor and the film flow.

All conservation equations can be written in the form of a general conservation equation

$$\rho u \frac{\partial \Phi}{\partial x} + \rho v \frac{\partial \Phi}{\partial r} = \frac{1}{r}\frac{\partial}{\partial r}\left[\left(\frac{\eta}{\sigma} + \frac{\eta_t}{\sigma_t}\right) r \frac{\partial \Phi}{\partial r}\right] + S_t + S_\Phi \Phi + S_p \frac{\partial p}{\partial x} \quad (5)$$

with $\Phi = u, \hat{h}, k, \varepsilon$ (Table 1).

The measured wall temperatures are used as thermal boundary conditions. At the pipe axis the condition of symmetry is applied.

The numerical solution of the coupled partial differential equations is based on the differentiation scheme of Patankar and Spalding [6]. This method has been used previously in predicting condensation phenomena [7–9].

In the inlet region of the pipe the condensate film is laminar and becomes turbulent when a certain film thickness is exceeded. This critical film thickness expressed by a critical film Reynolds number depends on the interaction between vapor and film flow, especially on the shear stress at the interface. Details are given in refs. [10] and [11]. In the present investigation the k, ε-turbulence model is used in both flow regimes of the film. In the laminar regime the turbulence is damped and the well known theoretical results for laminar films are found. A minimum tur-

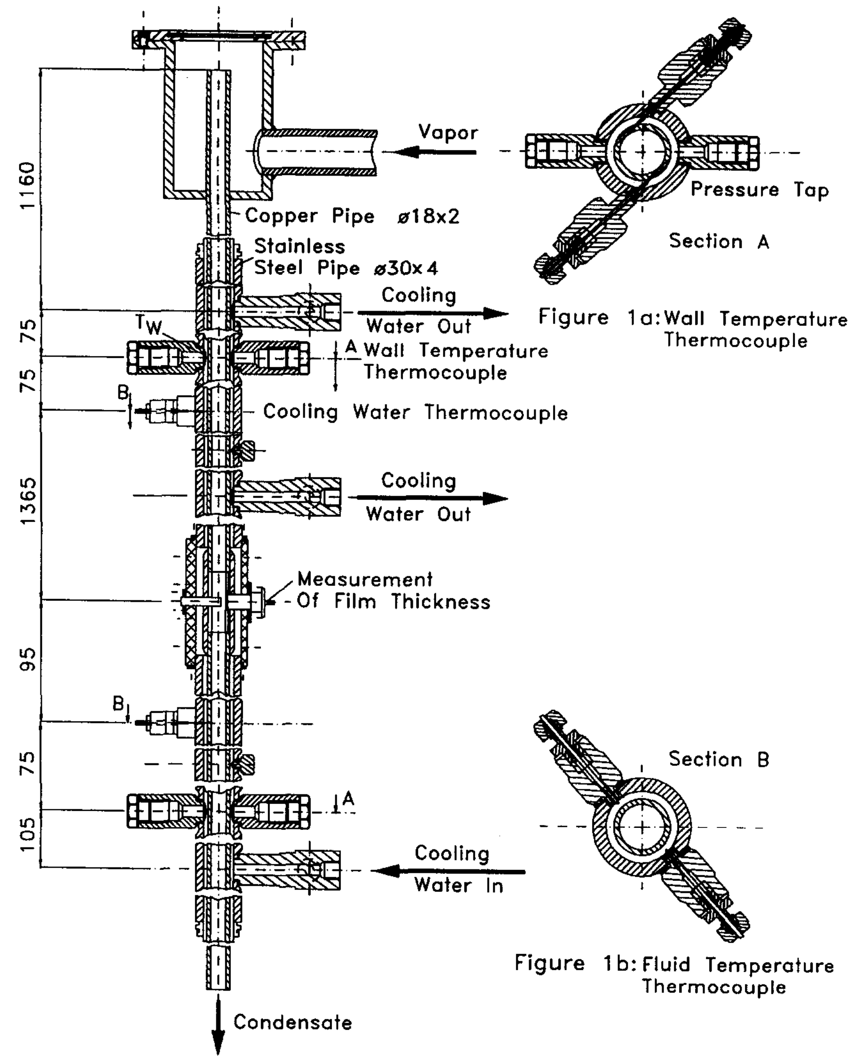

FIG. 1. Test condenser.

bulence level within the film has to be conserved to enable the turbulence model to predict the onset of film turbulence and the transition to the fully turbulent film flow

$$k_{\mathrm{F,min}} = (\eta_{\mathrm{F,t,min}}\varepsilon_{\mathrm{F}}/c_{\eta}f_{\eta}\rho_{\mathrm{F}})^{1/2} \tag{6}$$

with

$$\eta_{\mathrm{F,t,min}} = 0.5\eta_{\mathrm{F}}. \tag{7}$$

4. RESULTS

4.1. *Heat transfer*

The heat transfer coefficients along the tube are presented in Figs. 3 and 4 for vapor velocities at the pipe inlet of 23 and 96 m s^{-1}, respectively. In addition to the measurements and the numerical predictions the empirical results of Butterworth [12] are shown for comparison. It can be seen that in both cases the heat transfer is predicted quite satisfactorily. In Fig. 3 the film is laminar all along the pipe with a maximum film Reynolds number at the tube exit of $Re_{\mathrm{F}} = 160$. The numerical results agree reasonably well with the predictions of Butterworth. Compared to the experimental data the predicted heat transfer coefficients are slightly too low at the beginning of the pipe and too high at its end. This might be caused by inaccurately measured wall temperatures at the beginning of the condensation section where a slightly asymmetric wall temperature distribution was found. There is an excellent agreement between predicted and measured tube lengths for a complete condensation of the vapor.

At high vapor inlet velocities (see Fig. 4), the agreement between experiment and numerical prediction is even better. In this case the average heat transfer coefficients are higher by a factor of 3–4. A sharp

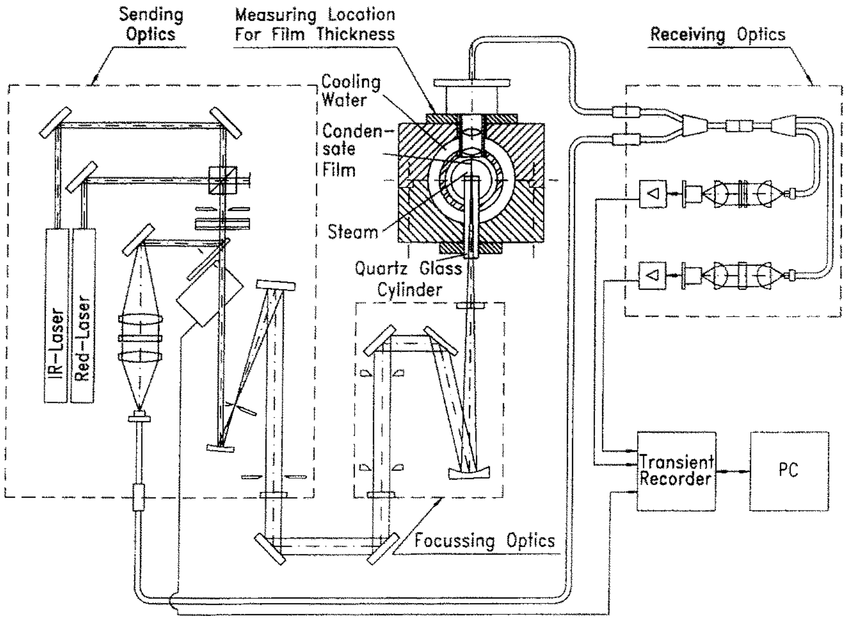

FIG. 2. Measuring system for film thickness.

Table 1. Terms of the transport equation

Φ	σ	σ_t	S_t	S_Φ	S_p
u	1	1	$2\rho g$	$-\frac{\rho g}{u}$	-1
$\hat{h}$	Pr	Pr_t	$\frac{1}{r}\left(\eta_{eff}-\left(\frac{\eta}{Pr}\right)_{eff}\right)\frac{\partial}{\partial r}\left(r\frac{\partial u^2/2}{\partial r}\right)+\rho u g$	0	0
k	1	σ_k	$\eta_t\left(\frac{\partial u}{\partial r}\right)^2$	$-\frac{\rho\varepsilon}{k}-c_4\frac{\eta}{k}\left(\frac{\partial\sqrt{k}}{\partial r}\right)^2$	0
ε	1	σ_ε	$c_1 f_1\frac{\varepsilon}{k}\eta_t\left(\frac{\partial u}{\partial r}\right)^2+c_3\frac{\eta\eta_t}{\rho}\left(\frac{\partial^2 u}{\partial r^2}\right)^2$	$-c_2 f_2\frac{\rho\varepsilon}{k}$	0

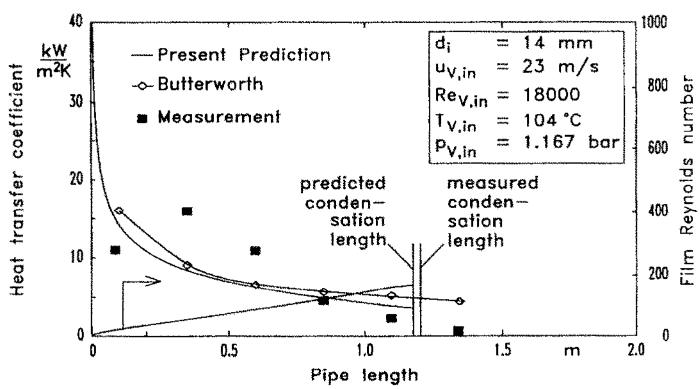

FIG. 3. Heat transfer during condensation of pure steam with low vapor velocity.

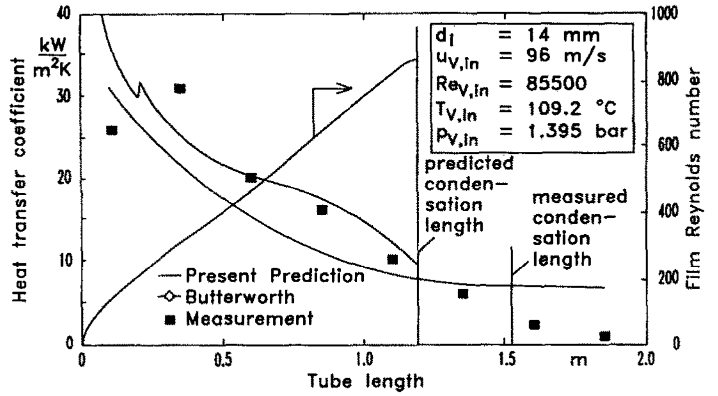

FIG. 4. Heat transfer during condensation of pure stream with high vapor velocity.

increase of the heat transfer coefficient is predicted at a tube length of 0.2 m which is due to the onset of the turbulence at a film Reynolds number of $Re_F = 250$.

4.2. *Film thickness*

4.2.1. *General remarks.* At the end of the pipe an average film thickness of between 0.1 and 0.5 mm was measured (see Figs. 5–8). With an acquisition of 50 000 data per second the time resolution is quite high. However, frequent variations of the laser intensities were observed which had to be compensated by a numerical filtration procedure. Furthermore, slight changes of the correction factors f_{IR} and f_R were found. Because these factors can only be determined without a condensate film in the pipe a check during a measuring period of about two hours was not possible. A number of experiments with large changes of those factors determined before and after a measuring period had to be discarded.

4.2.2. *Surface structure.* Measurements of the film thicknesss are taken for several local vapor and film Reynolds numbers which are determined from the heat transfer measurement.

In Fig. 5 the film thickness is presented for a time period of 0.3 s. These data are measured for an inlet vapor velocity of 66 m s^{-1}. For the given operational conditions, 99% of the vapor mass flow is already condensed at the measuring section of the film thickness, resulting in a very low vapor Reynolds number. Amplitudes of up to 1.6 mm are found with average values of 0.2 mm. With increasing vapor velocities the film becomes thinner and the film waves show smaller amplitudes with higher frequencies (see Fig. 6).

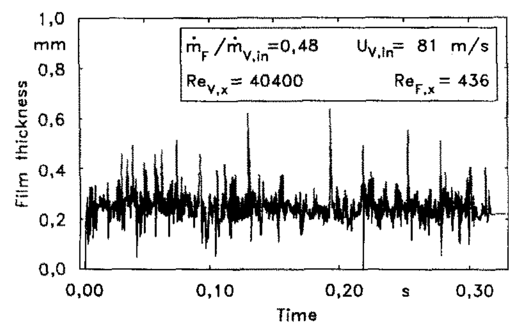

FIG. 6. Film thickness profile during condensation of pure steam.

4.2.3. *Time-averaged film thickness.* In Fig. 7 measured and numerically predicted values of the time-averaged dimensionless film thickness are plotted as a function of the local film Reynolds number. As convection effects have nearly no influence in the presented experiments the film thickness can be described

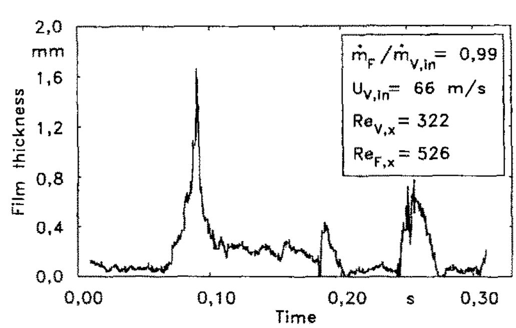

FIG. 5. Film thickness profile during condensation of pure steam.

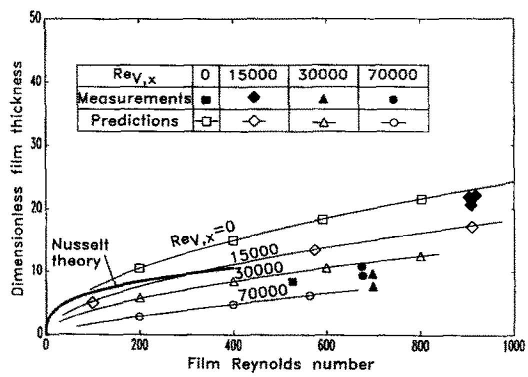

FIG. 7. Dimensionless averaged film thickness during condensation of pure steam for different vapor Reynolds numbers.

Table 2. Numerically predicted constants of equation (9)

$Re_{V,x}$	0	15 000	30 000	70 000
a	0.52	0.55	0.57	0.75
c	0.67	0.42	0.28	0.053

by local properties only. The result of the Nusselt theory for the laminar film with stagnant vapor and a wavy free interface, equation (8), is shown for comparison:

$$\delta^* = 1.44 Re_F^{0.33}. \qquad (8)$$

Due to the strong coupling conditions between vapor and film the predictions based on the solution of the complete system of conservation equations differ greatly from that predicted by Nusselt's theory.

However, the numerical predictions can be approximated by simple power laws of Nusselt type

$$\delta^* = c\, Re_F^a \qquad (9)$$

with the constants a and c depending on the local vapor Reynolds number according to Table 2.

As the experimental data show a large scatter, an enormous number of measurements are needed to evaluate an average value. The agreement between prediction and experiment is rather poor.

Finally, in Fig. 8 measurements of film thickness within a stagnant vapor are shown for the laminar film region. Each experimental value represented in the figure is an average of 3–5 measurements consisting of 4000 up to 16 000 measurement values for the film thickness. The scatter of the results is again quite important. In spite of the large scatter a regression analysis of the measured data leads to an equation

$$\delta^* = 1.55\, Re_F^{0.34} \qquad (10)$$

which is in close agreement with Nusselt's equation, equation (8).

5. SUMMARY

In the present paper the heat transfer and the film thickness in a condenser pipe are examined by experimental and theoretical means.

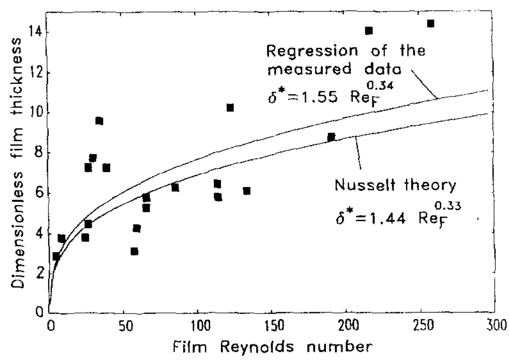

FIG. 8. Dimensionless film thickness inside a pipe with stagnant steam.

The condensation is predicted on the basis of the conservation equations for mass, momentum and energy in combination with the k,ε-turbulence model for low turbulence Reynolds numbers by Jones and Launder for both the vapor and the film flow. To predict the transition from laminar to turbulent film flow a limiting value for the kinetic energy of turbulence is suggested in order to avoid a complete damping of turbulence and to simulate the effects of film waves.

The heat transfer coefficients along the condensation tube are evaluated from heat balances at different test sections. Additionally, the time-dependent film thickness is measured with high local resolution by a new optical absorption method. A comparison between predicted and measured heat transfer coefficients and film thicknesses shows a reasonably good agreement, even for high vapor velocities.

Acknowledgements—The authors thank Prof. S. Wittig and his co-workers at the University of Karlsruhe for supporting the film thickness measurements and the Deutsche Forschungsgemeinschaft (German National Science Foundation) for sponsoring this work.

REFERENCES

1. W. Nusselt, Die Oberflächenkondensation des Wasserdampfes, *Z. VDI* **60**, 541–546, 569–575 (1916).
2. F. K. Wasden and A. E. Dukler, Insights into the hydrodynamics of free falling wavy films, *A.I.Ch.E. J.* **35**, 187–195 (1989).
3. Th. Sattelmayer, K. H. Sill and S. Wittig, Optisches Meßgerät zur Bestimmung der Eigenschaften welliger Flüssigkeitsfilme, *Technisches Messen* **4**, 155–160 (1987).
4. W. P. Jones and B. E. Launder, The prediction of laminarization with a two-equation model of turbulence, *Int. J. Heat Mass Transfer* **15**, 301–314 (1972).
5. W. P. Jones and U. Renz, Condensation from a turbulent stream onto a vertical surface, *Int. J. Heat Mass Transfer* **17**, 1019–1028 (1974).
6. S. V. Patankar and D. B. Spalding, *Heat and Mass Transfer in Boundary Layers*, 2nd Edn. Intertext Books, London (1970).
7. U. Renz and H.-P. Odenthal, Numerical prediction of heat and mass transfer during condensation from a turbulent vapour/gas-stream onto a vertical liquid film, *18th National Heat Transfer Conf.*, San Diego, CA, 6–8 Aug., *Symp. Vol. Condensation Heat Transfer*. ASME, New York (1979).
8. R. Cossmann, H.-P. Odenthal and U. Renz, Heat and mass transfer during partial condensation in a turbulent pipe flow, *Proc. 7th Int. Heat Transfer Conf.*, Munich, Vol. 5, pp. 53–58 (1982).
9. F. Ramadan and U. Renz, Der Wärme- und Stoffaustausch bei der Filmkondensation reiner Dämpfe im turbulent durchströmten Rohr, *Chem. Ing. Tech.* **56**, 482–483 (1984).
10. F. A. Ramadan, Zur Berechnung des Wärme- und Stoffaustausches bei der Filmkondensation reiner Dämpfe im durchströmten Rohr. Dissertation, RWTH Aachen (1983).
11. R. Bellinghausen, Die Filmkondensation reiner Dämpfe. Dissertation, RWTH Aachen (1990).
12. D. Butterworth, *Film Condensation of Pure Vapor*. Hemisphere, New York (1983).

TRANSFERT THERMIQUE ET EPAISSEUR DE FILM PENDANT LA CONDENSATION DE VAPEUR D'EAU S'ECOULANT A GRANDE VITESSE DANS UN TUBE VERTICAL

Résumé—Le transfert thermique pour une vapeur d'eau pure s'écoulant vers le bas dans un tube vertical de condenseur est déterminé et l'épaisseur du film ondulé est mesurée en utilisant une méthode d'absorption laser. Les données mesurées sont comparées aux prédictions numériques basées sur la solution des équations de conservation de la masse, de la quantité de mouvement et de l'énergie pour la vapeur et la phase liquide. On utilise pour les deux phases le modèle k–ε de Jones et Launder. On montre que la transition entre écoulement du film laminaire et turbulent peut être prédite raisonablement bien si l'énergie cinétique turbulente dans le film est conservée à une valeur minimale.

WÄRMEÜBERTRAGUNG UND FILMDICKE BEI DER KONDENSATION EINES SCHNELL STRÖMENDEN DAMPFES IN EINEM SENKRECHTEN ROHR

Zusammenfassung—In einem senkrechten Rohr wurde der Wärmeübergang bei der Kondensation eines schnell strömenden reinen Dampfes untersucht. Die sich einstellenden Kondensatfilmdicken konnten mit einer Laser-Absorptions-Meßtechnik mit zeitlich und örtlich hoher Auflösung erfaßt werden. Die Versuchsergebnisse werden mit Rechenergebnissen auf der Grundlage einer numerischen Lösung der Erhaltungsgleichungen für den Impuls und die Energie der Dampf- und der Flüssigkeitsphase verglichen. Die Übereinstimmung ist sowohl im laminaren als auch im turbulenten Bereich gut. Der Umschlagpunkt laminar/turbulent des Films läßt sich theoretisch voraussagen, wenn ein Mindestwert für die Filmturbulenz festgelegt wird.

ТЕПЛОПЕРЕНОС И ТОЛЩИНА ПЛЕНКИ В УСЛОВИЯХ КОНДЕНСАЦИИ ВОДЯНОГО ПАРА ПРИ ТЕЧЕНИИ С БОЛЬШОЙ СКОРОСТЬЮ В ВЕРТИКАЛЬНОЙ ТРУБЕ

Аннотация—С помощью метода лазерной абсорбции определяется теплоперенос при нисходящем течении чистого водяного пара по вертикальной трубе конденсатора и измеряется толщина волнообразной пленки. Экспериментальные данные сравниваются с численными результатами, полученными на основе решения уравнений сохранения массы, импульса и энергии для парообразной и жидкой фаз. Для обеих фаз используется k–ε модель турбулентности, разработанная Джонсом и Лаундером. Показано, что переход от ламинарного течения к турбулентному пленочному можно достаточно точно определить при условии минимизации турбулентной кинетической энергии в пленке.