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Surface evaporation of turbulent falling films

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Abstract—In refrigeration and heat pump applications, falling film evaporators should be very attractive due to high heat transfer coefficients with negligible pressure drop. In this paper results from an experimental study with refrigerant R12 are reported. The results are compared with existing correlations in the literature and the need for a new correlation is recognized. The following equation is derived with data from three different sources (including the present study and the study of Chun and Seban):

$$Nu = 0.012\, Re^{0.28}\, Pr^{0.53}.$$

The equation has been validated with data from three additional studies.

INTRODUCTION

FALLING film evaporators are today used particularly for concentrating heat sensitive liquids, such as fruit juices. They have inherent characteristics such as high heat transfer coefficients and, except in vacuum applications, negligible pressure drop. This means that the temperature difference between the heating medium and the evaporating liquid can be small. In refrigeration and heat pump applications, these features should be very attractive, since the evaporation temperature directly influences the cycle performance. This is the reason for the research activities at the Department of Heat and Power Technology, Chalmers, Sweden, on falling film evaporation of refrigerants in vertical tubes [1, 2].

When using falling film evaporators as concentrators, the flow regime is normally laminar or near laminar. This is due to the concentration specification on the liquid at the evaporator outlet. A certain fraction of the incoming liquid must be evaporated, which leads to low inlet flow rates since the allowed heat flux is limited. The heat transfer regime is normally surface evaporation, since the temperature differences are kept too low for initiating nucleate boiling. With pure refrigerants the above limitations are not present. Turbulent flow can be used throughout the whole evaporator if non-evaporated liquid is recirculated. This means increased heat transfer coefficients as well as better wetting of the evaporator tubes. Nucleate boiling could also occur, since nucleate boiling with refrigerants normally is initiated at a lower temperature difference compared to water at normal pressure. This paper will, however, be restricted to surface evaporation in the turbulent regime.

There are numerous studies on heat transfer to falling films in surface evaporation, but there are not many which include Reynolds numbers above, say, 10 000. Both empirical and theoretical models exist, but the most often cited experimental study is the one of Chun and Seban [3] from 1971. Handbooks, such as *VDI-Värmeatlas* [4] and *Handbook of Heat Transfer Fundamentals* [5], recommend Chun and Seban's empirical correlation for the turbulent regime. The latter also recommends Chun and Seban's correlation for the laminar regime. It is also quite common to evaluate theoretical models by comparison with the results of Chun and Seban, for example Hubbard *et al.* [6], Seban and Faghri [7] and Yih and Liu [8]. The findings of the studies [1, 2], however, show that it is doubtful if Chun and Seban's results for the turbulent regime can be extrapolated outside their experimental conditions. This is of course not a surprising result for an empirical correlation, but still it is seldom pointed out.

Since the main interest for the present study is refrigerants in the turbulent flow regime, those studies with refrigerants as the test fluid known to the authors are summarized below (in chronological order) together with the study of Chun and Seban.

PREVIOUS STUDIES

Struve [9] made measurements at atmospheric pressure with refrigerant R11. The tube was heated by condensing steam and the measurement section was located at the tube bottom. Tubes of different surface roughness were used but the surface did not influence the heat transfer in surface evaporation. The thermal developing length was studied by using several tubes of different lengths (varying between 0.15 and 1.25 m) and the thermal entry length was given as 0.4–0.8 m. The equation describing surface evaporation of R11 at atmospheric pressure for developed flow was given as†

† There exist two common ways to express the Reynolds number. The difference is a factor of 4. In the following all equations are rewritten, if necessary, to the relevant definition in the present paper.

NOMENCLATURE

c_p	specific heat at constant pressure [$J\,kg^{-1}\,K^{-1}$]
d	tube outside diameter [m]
g	gravitational acceleration [$m\,s^{-2}$]
h	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
k	thermal conductivity [$W\,m^{-1}\,K^{-1}$]
L	heated length [m]
L_E	thermal entry length [m]
$\mathscr{L}^*$	dimensionless distance, L/L_E
Nu	Nusselt number, $(h/k)\,(v^2/g)^{1/3}$
Pr	Prandtl number, $c_p\mu/k$
Re	liquid Reynolds number, $4\Gamma/\mu$
Re_{VAP}	vapour Reynolds number
W	mass flow rate [$kg\,s^{-1}$].

Greek symbols

Γ	liquid film flow per perimeter (for a tube $\Gamma = W/(\pi d)$) [$kg\,m^{-1}\,s^{-1}$]
μ	dynamic viscosity [$kg\,m^{-1}\,s^{-1}$]
v	kinematic viscosity [$m^2\,s^{-1}$].

Miscellaneous

$SE_{ESTIMATE}$	standard error of estimate.

$$Nu = 0.0079\,Re^{0.41}. \quad (1)$$

The Prandtl number was 4.1 and the maximum Reynolds number was around 9000. Struve compared his results with the theoretical analysis from Dukler [10], which for the turbulent regime was approximated by Struve with

$$Nu = (0.110-\{0.565/(Pr+5.47)\})\,Re^{0.231}. \quad (2)$$

With $Pr = 4.1$ the above equation overestimates the heat transfer by 65% at $Re = 2000$ and by 26% at $Re = 9000$ compared to equation (1). Even so, Struve recommended equation (2) for surface evaporation with Prandtl numbers between 1 and 10.

Chun and Seban [3] used an electrically heated tube. The tube was 0.6 m long, but only the bottom half was heated. Test fluid was water. The thermal entry length was given as 0.2 m. The pressure range was 0.004–0.1 MPa and the liquid Prandtl number varied between 5.7 and 1.8. The maximum mass flow rate was kept approximately constant, which resulted in a systematic variation of the maximum Reynolds number with the evaporation temperature and thus with the Prandtl number. The maximum Reynolds number at a Prandtl number of 5.7 was 8000 while at a Prandtl number of 1.8 the maximum Reynolds number was 21 000. This resulted in a correlation between the Prandtl and Reynolds numbers, which makes the resulting equation rather unsuitable for extrapolation. The equation given by Chun and Seban for the turbulent regime was

$$Nu = 0.0038\,Re^{0.4}\,Pr^{0.65}. \quad (3)$$

The start of the turbulent regime was given as

$$Re = 5800\,Pr^{-1.06}. \quad (4)$$

Chun and Seban compared their results with the analysis from Dukler (that is with Struve's equation (2)) but found the agreement to be poor. If equations (2) and (3) are compared at $Pr = 4.1$ (corresponding to Struve's experimental conditions), equation (2) gives 48% higher Nusselt numbers at $Re = 2000$ while at $Re = 10\,000$ the difference has decreased to 13%. A comparison between equations (1) and (3) shows better agreement. Equation (1) gives approximately 10% lower values than equation (3) for the same Reynolds number at $Pr = 4.1$.

Elle [11] made experiments with pure R11 and mixtures of R11 and mineral oil near atmospheric pressure. The tube was 0.975 m long and electrically heated. The thermal entry length was given as 0.25–0.4 m. For the heat transfer of pure R11 at 0.12 MPa ($Pr = 4.2$), the following equation was given for the turbulent regime (maximum Reynolds number in the study was 6000):

$$Nu = 0.0092\,Re^{0.4}. \quad (5)$$

This gives results which are close to the ones of Chun and Seban. Values from equation (5) lie 5% below equation (3) with $Pr = 4.2$. Elle varied the Prandtl number in a wide range by adding mineral oil. He concluded that Struve's approximation of the results from Dukler gave a correct Prandtl number dependence, although the values were approximately 30% too high. The resulting equation given by Elle, including a Prandtl number dependence, is therefore equation (2) multiplied by 0.71 or

$$Nu = (0.078-\{0.401/(Pr+5.47)\})\,Re^{0.231}. \quad (6)$$

A comparison between equations (6) and (5) shows good agreement in the Reynolds number range studied. At $Re = 2000$ equation (6) overestimates the heat transfer by 10% while at $Re = 6000$ it underestimates the heat transfer by 9% compared to equation (5).

Fagerholm *et al.* [12] used refrigerant R114 in their study. The tube was 2 m long and the test fluid was distributed on the outside of the tube. Hot water was flowing on the tube inside and used as heating medium. The size of the thermal entry region is not given, since only average values of the whole tube were measured. Pressures between 0.2 and 0.37 MPa and Reynolds numbers up to 11 000 were studied. The measured values at the lowest pressure 0.2 MPa (with $Pr = 5.4$), was represented by the following equation:

$$Nu = 0.026\,Re^{0.31}. \quad (7)$$

This gives values in good agreement with the correlation of Chun and Seban (equation (3)) at Reynolds numbers above 3000. Equation (7) overestimates the heat transfer by 10% at $Re = 3000$ while the difference is negligible at $Re = 11\,000$. Based on this good agreement, the authors recommend equation (3) for other Prandtl numbers.

Munch Berntsson [1] used a similar test rig as Fagerholm, though the falling film was distributed on the tube inside. Liquid R11 was used as heating medium. The apparatus of Munch Berntsson was primarily designed for heat transfer measurements of non-azeotropic mixtures, but measurements with pure refrigerants were also made. Surface evaporation was studied with refrigerants R12 and R114 at 288 K and R11 at 296 K. The measured heat transfer was compared to equations (3), (2) and (for R11 only) equation (1). The results showed that equation (2) overestimated the heat transfer for all fluids, while equation (3) gave reasonable agreement for R12 although the observed heat transfer was less dependent of flow rate than predicted by equation (3). For the other two refrigerants equation (3) gave too high values. Since the Prandtl numbers for R11 and R114 were higher than the Prandtl number for R12, this indicates a lower Prandtl number dependence than given in equation (3). The measured heat transfer for R11 was well described by Struve's equation for R11 (equation (1)), especially for Reynolds numbers below 10 000.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

The experimental equipment used in the present study is schematically shown in Fig. 1. The test tube, 25 mm i.d., was made of a Ni–Mo–Cr alloy (Hastelloy-276). It was part of an electric d.c. circuit and thus directly heated. The heated length used was 2.16 m and the whole plant was thermally insulated from the surroundings.

This project started before the ongoing debate about the depletion of the atmospheric ozone layer. This explains the choice of test fluid, CFC refrigerant R12. (The use of R12 is now restricted by the Montreal protocol.) Its physical and thermodynamic properties are well known. The liquid was distributed on the inside of the tube through a sintered metallic filter. The vapour formed was condensed in a condenser. Non-evaporated liquid was mixed with the condensate and pumped through a subcooler to the distributor again. The evaporation pressure was kept constant by a presssure regulator which controlled the brine flow rate to the condenser.

A total of 27 steel sheathed copper–constantan

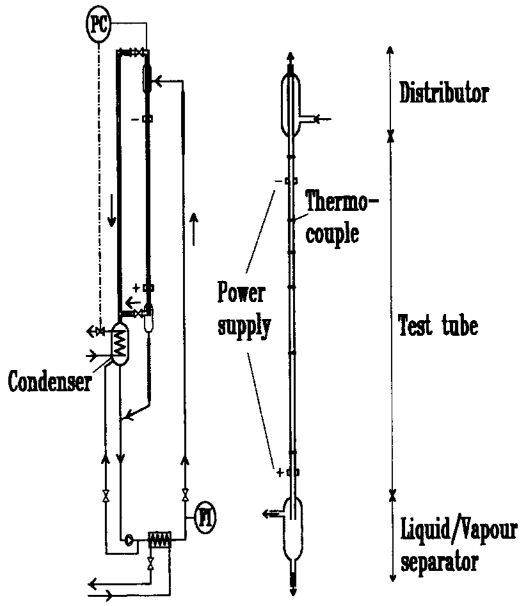

FIG. 1.

thermocouples were installed in the plant (the sheath and thermocouple wires were electrically insulated). The temperature differences between wall and saturated vapour were measured at 18 different locations. In order not to disturb the liquid film or the heat flux generated in the tube, the 18 wall thermocouples were soldered to the outer wall. The inside wall temperature could be calculated from the heat conduction equation with internal heat production since the thermally insulated outside surface could be considered to be adiabatic. From Fig. 1 it is seen that these 18 thermocouples were located at six different levels, three at each level, equally spaced around the perimeter. One level was located above the heated section. This 'extra' measurement made it possible to verify that the axial thermal conduction was negligible. The rest of the thermocouples were mainly used for supervising purposes. Temperature measurement errors were estimated to be around 0.05 K.

The experiments were used to get heat transfer data for both surface evaporation and nucleate boiling, although only surface evaporation will be discussed here. Therefore the following parameters were varied:

- refrigerant flow;
- evaporation pressure;
- heat flux.

In each run, the inlet flow rate and evaporation pressure were kept constant while the heat flux was increased step by step and the resulting temperature differences at steady-state were recorded together with measured heat flux. The maximum heat flux was adjusted to evaporate approximately 80% of incoming liquid. The flow rate was varied between 0.5 and 4 $dm^3\ min^{-1}$ while the pressure was varied between 0.25 and 0.85 MPa. This resulted in Reynolds numbers between 2000 and 18 000. The liquid Prandtl number was around 3 and practically constant in the pressure range studied. (The corresponding temperature interval was 267–308 K.)

The basic element in the evaluation procedure was the establishing of the boiling curve. That is a plot of heat flux vs wall superheat for constant mass flow and pressure. For each run five boiling curves were constructed, one for each level. The wall superheat was taken as the average value around the perimeter. The assumption that the position around the circumference was insignificant was validated by variance analysis. Points lying on a straight line through the origin of the boiling curve represents surface evaporation. The departure from the straight line was taken as the onset of boiling. In this way the measured points representing surface evaporation were identified.

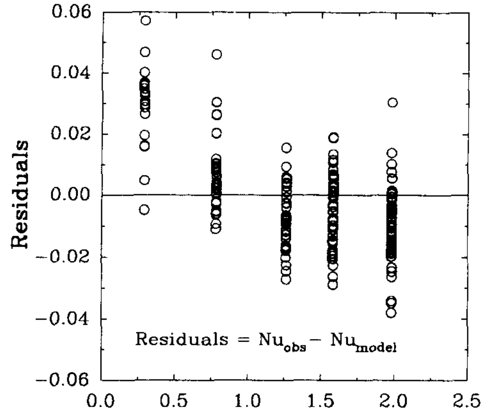

FIG. 2.

RESULTS

Local Reynolds numbers and mass vapour quality were calculated from inlet flow rate, inlet subcooling, heat flux, and level. Due to the small inlet subcooling a fraction of the tube was used to bring the liquid to saturation. Since we were interested in evaporation (and not in heating) in the turbulent flow regime, observations with a quality of less than 1% or with a Reynolds number below 1800 were excluded. The critical Reynolds number 1800 for the start of the turbulent flow regime was taken from equation (4) (from ref. [3]) using $Pr = 3$. Based on regression the following equation for surface evaporation was derived:

$$Nu = 0.027\, Re^{0.25}$$
$$SE_{\text{ESTIMATE}} = 0.017 \tag{8}$$

where SE_{ESTIMATE} is the standard error of estimate at the mean Reynolds number. A residual plot of $(Nu_{\text{OBSERVED}} - Nu_{\text{MODEL}})$ vs the Prandtl number did not show any Prandtl number dependence. This is explained by the fact that the Prandtl number for R12 in the temperature range studied is nearly constant ($Pr \approx 3$) as mentioned above. This means that the Prandtl number dependence could not be determined in this study. If the residuals instead were plotted vs heated length, a clear dependence could be seen (see Fig. 2). However, as could be seen from the figure, the length dependence seems to cease around 1.3 m. The most simple and likely explanation is that there exists a long thermal developing region. In order to keep the equation dimensionless, a dimensionless distance $\mathscr{L}^*$ was introduced

$$\mathscr{L}^* = L/L_E \quad \text{for } L < L_E$$
$$\mathscr{L}^* = 1 \quad \text{for } L \geqslant L_E \tag{9}$$

where L_E is the thermal entry length and L the heated length. In the regression, L_E was an adjustable parameter. Best fit was obtained by setting $L_E = 1.36$ m. The resulting equation including length dependence was

$$Nu = 0.026\, Re^{0.25}\, \mathscr{L}^{*\,-0.10}$$
$$SE_{\text{ESTIMATE}} = 0.012. \tag{10}$$

The pressure drop along the tube was so small that it could not be detected by the manometers used (resolution 5 kPa). This indicates that the generated vapour did not exert any significant vapour shear on the liquid film. Heated length and vapour Reynolds number (and consequently also vapour shear) are, however, correlated since vapour is generated along the tube and vapour Reynolds number thus increases with heated length. As a check Re_{VAP} was used instead of $\mathscr{L}^*$ in the regression. This lead to higher residuals (larger SE_{ESTIMATE}) and it was concluded that the length dependence was indeed the result of thermal development. Plots of residuals vs heat flux and pressure did not reveal any further dependence. Equation (10) is thus the final equation chosen to represent surface evaporation of R12 in the pressure and Reynolds number range studied.

DISCUSSION

How well does the general equations for surface evaporation, that is the equations that include both Reynolds and Prandtl number dependence, describe the observations from the present study? In Fig. 3, the observed Nusselt numbers are plotted vs Reynolds numbers. Included in the figure are the correlations from Chun and Seban (equation (3)) with $Pr = 3$ and equation (10) with $\mathscr{L}^* = 1$. At Reynolds numbers in the lower range, equation (3) predicts Nusselt numbers of the correct size but the discrepancy increases with increasing Reynolds numbers. The observed values at a thermal length of 0.29 m (and a hydrodynamic flow length of 0.71 m) fit reasonably well to equation (3) in the whole range of Reynolds numbers. Note that equation (3) was derived with measured

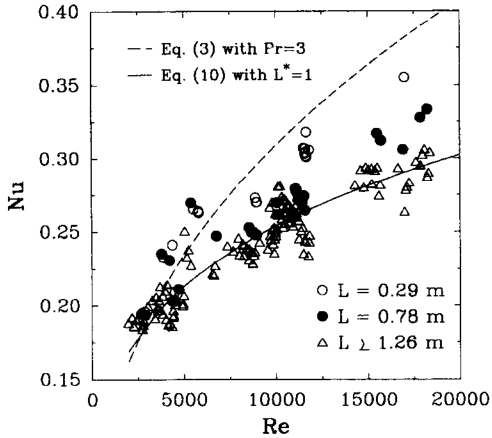

FIG. 3.

values at a thermal length of 0.1–0.2 m and a hydrodynamic flow length of 0.5–0.6 m. Figure 4 shows equations (2), (3) and (5) for $Pr = 3$ and equation (10) with $\mathscr{L}^* = 1$. Equations (5) and (10) are very similar for this Prandtl number, while equation (2) gives 40% higher values. As noted above, the correlation from Chun and Seban gives Nusselt numbers of the same magnitude as equations (5) and (10) at Reynolds numbers in the lower range ($\pm 10\%$ for Reynolds numbers between 2000 and 8000), but at $Re = 20\,000$ the estimated Nusselt numbers are approximately 30% higher. The Reynolds number dependence is apparently too high in equation (3) to describe the observed heat transfer in this study over the whole Reynolds number range. A lower Reynolds number dependence is also noted in the study of Munch Berntsson [1]. Munch Berntsson also found a too high Prandtl number dependence in equation (3). One possible explanation of the inadequacy of equation (3) at high Reynolds numbers is the inherent correlation between the independent variables in Chun and Seban's data mentioned earlier. At Prandtl numbers between 3 and 6 (common values for refrigerants) the maximum Reynolds numbers in their study were 14 000 and 8000, respectively. In most studies with refrigerants, the maximum Reynolds number has been around or below these values. The exceptions are the present study and the one of Munch Berntsson. This could explain why equation (3) has been widely recommended in the past.

In order to overcome the limitations in equation (3) and to make more general use of the results in the present study, regression analysis has been applied on the material from the present study in addition to the data from Munch Berntsson and Chun and Seban. The data from Chun and Seban have been extracted from plots in ref. [3], while the data from Munch Berntsson were available from the author. In the combined material from these three sources, the correlation between the Reynolds and the Prandtl numbers was insignificant. The maximum Reynolds number in the data was 27 000, while the lower limit was given by equation (4). Minimum and maximum Prandtl numbers were 1.8 and 5.7, respectively.

In the regression a weight variable was used to hinder the data from the present study to dominate due to its larger number of observations. With the weight function, each study became equally important. (Weight was simply the inverse of the number of observations for each study.) The resulting equation was

$$Nu = 0.012\, Re^{0.28}\, Pr^{0.53}$$

$$SE_{\text{ESTIMATE}} = 0.028. \tag{11}$$

The exponents in the Reynolds and Prandtl numbers are considerably lower than in equation (3). In Fig. 5, the observed Nusselt numbers are plotted vs predicted values. The maximum error is 20%.

The standard error of estimate from equation (11)

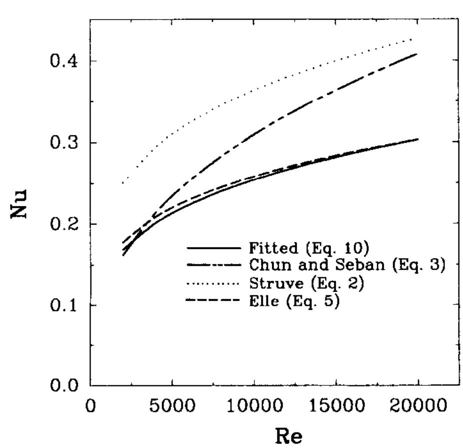

FIG. 4.

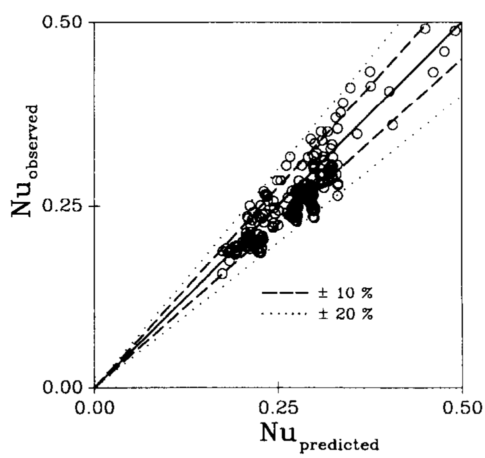

FIG. 5.

could be compared to the standard errors given by equations (2), (3), and (6) used on this material. They are:

$$\text{equation (2)}: SE_{\text{ESTIMATE}} = 0.066$$

$$\text{equation (3)}: SE_{\text{ESTIMATE}} = 0.052$$

$$\text{equation (6)}: SE_{\text{ESTIMATE}} = 0.029.$$

The correlation from Elle, equation (6), gives errors of the same size as equation (11) ($SE_{\text{ESTIMATE}} = 0.028$). That equation could therefore also be recommended, although it is written in a somewhat unconventional form and the number of parameters is four compared to three in equation (11).

Even though the observed values were not available from the studies of Struve, Elle and Fagerholm, and thus not included in the regression, the equations describing the heat transfer for their respective fluids (equations (1), (4), and (5)) could be used to check the validity of equation (11). The result is found in Table 1. Nusselt numbers were calculated given in the respective references for the Reynolds numbers covered in the respective studies. Comparison was made with equation (11) using the appropriate Prandtl number. The maximum error is around 15%. It is the results from Fagerholm *et al.* [12] which differ most. Fagerholm *et al.*'s results overestimate the heat transfer compared to equation (11). This could be explained by the fact that the observed Nusselt numbers are average values over the whole tube and thus higher than the Nusselt numbers for fully developed flow.

CONCLUSIONS

The well-known correlation from Chun and Seban (equation (3)) [3], describing heat transfer in surface evaporation for turbulent falling films, is shown to predict the heat transfer reasonably well for refrigerants if use of the correlation is limited to the Reynolds and Prandtl numbers covered in the original study. Large errors could occur when the correlation is extrapolated to higher Reynolds numbers. A new correlation, using the data from Chun and Seban together with data from Munch Berntsson and the present study, describes the results from six different studies within 15%. The new recommended equation describing surface evaporation of turbulent falling films reads:

$$Nu = 0.012\, Re^{0.28}\, Pr^{0.53}$$

$$SE_{\text{ESTIMATE}} = 0.028.$$

The turbulent regime starts according to Chun and Seban at

$$Re = 5800\, Pr^{-1.06}$$

and the equation is derived under this assumption. The maximum Reynolds number in the data was $Re = 27\,000$ and the Prandtl number range covered by the data was 1.8–5.7.

Table 1.

Study	Pr	Re	Equation number	Error (%)†
Chun	5.7	8000	3	−15
		2000		+3
	1.77	21 000		−12
		4000		+8
Present	3	18 000	10	+10
		2000		+3
Struve	4.1	8000	1	±0
		3200		+11
Elle	4.2	6000	4	−2
		3200		+6
Fagerholm	5.4	11 000	6	−17
		3000		−13

† Error is the error compared to equation (11).

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EVAPORATION EN SURFACE DES FILMS TOMBANTS TURBULENTS

Résumé—Dans les applications de réfrigération et de pompe à chaleur, les évaporateurs à film tombant sont très intéressants par leur coefficient de transfert thermique élevé avec une faible perte de charge. On présente des résultats expérimentaux avec du R12. Les résultats sont comparés à des formules existantes et il apparait la nécessité d'une nouvelle corrélation. L'équation suivante est établie avec des données provenant de différentes sources (incluant la présente étude et celle de Chun et Seban) :

$$Nu = 0{,}012\,Re^{0,28}\,Pr^{0,53}.$$

Cette équation est validée par trois études additionnelles.

OBERFLÄCHENVERDAMPFUNG AN TURBULENTEN FALLFILMEN

Zusammenfassung—Aufgrund der sehr großen Wärmeübergangskoeffizienten bei gleichzeitig vernachlässigbarem Druckabfall sind Fallfilmverdampfer für kältetechnische Anlagen und Wärmepumpen sehr gut geeignet. In der vorliegenden Arbeit werden experimentell mit dem Kältemittel R12 ermittelte Ergebnisse vorgestellt. Diese Ergebnisse werden mit Korrelationsgleichungen aus der Literatur verglichen—daraus folgt die Notwendigkeit einer neuen Korrelation. Die folgende Gleichung wird aufgrund von Versuchsdaten aus drei Quellen entwickelt (dies sind außer der vorliegenden Arbeit die Untersuchungen von Chun und Seban):

$$Nu = 0{,}012Re^{0,28}\,Pr^{0,53}.$$

Diese Gleichung wird mittels Daten aus drei zusätzlichen Untersuchungen bestätigt.

ПОВЕРХНОСТНОЕ ИСПАРЕНИЕ ТУРБУЛЕНТНЫХ СТЕКАЮЩИХ ПЛЕНОК

Аннотация—Благодаря высоким коэффициентам теплопереноса и пренебрежимо малому перепаду давления пленочные испарители могут эффективно использоваться в холодильной технике и тепловых насосах. В данной работе представлены результаты экспериментального исследования хладона R12. Проводится сравнение этих результатов с имеющимися в литературе обобщающими соотношениями, и признается необходимость получения новых зависимостей. По данным трех различных источников (включая настоящее исследование, а также работу Чана и Себана) выведено следующее уравнение:

$$Nu = 0{,}012Re^{0,28}\,Pr^{0,53}.$$

Справедливость уравнения подтверждается результатами трех дополнительных исследований.