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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

No. 109, 2001, 101-120

MARIAN TRELA[1], DARIUSZ BUTRYMOWICZ[1]

# Enhancement of condensation heat transfer – a review

A state-of-the-art review on the enhancement of condensation heat transfer techniques developed recently is presented in the paper. The classification of enhancement techniques is provided. It encompasses both the passive and active methods. The particular attention is paid to the problems where the authors have been carried out theoretical as well as experimental research. As an example of the passive technique the results of condensate drainage enhancement by using the drainage strip are presented. For an active method of heat transfer enhancement a novel EHD technique is described and some experimental results are provided.

## Nomenclature

$a$ – capillary constant, $a = \sqrt{\sigma/(\rho g)}, m$
$c_p$ – specific heat, J/kg·K
$C$ – constants in Eq. (7), given in Tab. 1
$Ca$ – capillary number, $Ca = \mu \bar{u}_N/\sigma$
$D$ – outer diameter of the tube, $m$
$E$ – electric field strength, $V/m$
$f$ – unit force, $N/m^3$
Fr – Froude number based on film thickness $\delta_N$, $\mathrm{Fr} = \bar{u}_N^2/(g\delta_N)$,
$\mathrm{Fr}_a$ – Froude number based on capillary constant, $\mathrm{Fr}_a = \bar{u}_N^2/(ga)$
$g$ – gravitational acceleration, m/s$^2$
Ga – Galileo number, $\mathrm{Ga} = gD^3/v^2$
$h_{fg}$ – latent enthalpy of vaporisation, J/kg
$H_f$ – fin height, m
$K$ – dimensionless film curvature, $K = \kappa a$, see Eq. (5)
$K$ – phase change number, $K = h_{fg}/(c_p \Delta T)$
Nu – Nusselt number, $\mathrm{Nu} = \alpha D/\lambda$
Pr – Prandtl number, $\mathrm{Pr} = c_p\mu/\lambda$
$q$ – electric charge density, $C/m^3$
$s$ – electrode distance, $m$
$T$ – temperature, $^o$C
$\Delta T_c$ – difference between saturation temperature and outside tube wall temperature, $K$
$U$ – electric potential difference between electrode and tube, kV
$\bar{u}_N$ – average velocity for Nusselt-type film flow, $\bar{u}_N = \delta_N^2 g/(3v), m/s$
$W_t$ – fin spacing at fin tip, $m$
We – Weber number, $\mathrm{We} = \rho\bar{u}_N^2\delta_N/\sigma$
$x$ – rectangular coordinate (along the strip), $m$
$\Delta x_h$ – height of the retained condensate on the strip, $m$

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$\alpha$ – film heat transfer coefficient, $W/(m^2 \cdot K)$
$\alpha$ – dimensionless parameter, $\alpha = We/Fr$
$\beta_{exp}$ – ratio defined by Eq. (12), dimensionless
$\beta_{Nu}$ – ratio defined by Eq. (11), dimensionless
$\delta_N$ – condensate film thickness for Nusselt-type film flow, $\delta_N = [3\ \Gamma v/(\rho g)]^{1/3}$, $m$
$\varepsilon$ – electric permittivity, $F/m$
$\varepsilon$ – dimensionless parameter, defined by Eq. (2)
$\varepsilon_d$ – drainage parameter, defined by Eq. (3)
$\varepsilon_0$ – electric permittivity of free space, $\varepsilon_0 = 8.854 \times 10^{-12}$ F/m
$\varepsilon_r$ – relative permittivity, $\varepsilon_r = \varepsilon/\varepsilon_0$
$\Phi$ – flooding angle, rad; $\Phi_f = \Phi/\pi$
$\gamma$ – dimensionless parameter, $\gamma = Fr_a/Ca^2$
$\Gamma$ – mass flow rate per unit length, kg/(m·s)
$\kappa$ – film curvature, 1/m
$\lambda$ – thermal conductivity of condensate, W/(m·K)
$\mu$ – dynamic viscosity of condensate, Pa·s
$\nu$ – kinematic viscosity of condensate, $m^2/s$
$\theta$ – fin tip half angle, rad
$\sigma$ – surface tension, N/m
$\rho$ – condensate density, $kg/m^3$
$\Omega$ – dimensionless heat transfer enhancement factor, see Eq. (6).

Subscripts

$e$ – electrohydrodynamic (EHD) force,
exp – experimental value,
$f$ – finned tube, flooding zone
$h$ – hydrostatic zone on the strip.
$m$ – maximum,
Nu – based on the Nusselt model
$T$ – constant temperature,

## 1. Introduction

The goals of enhanced heat transfer can be described as the desire to encourage or accommodate high heat fluxes. The heat flux can be increased by elevating the heat transfer coefficient for fixed temperature difference, for either constant heat transfer rate or constant area. Alternatively, enhancement permits accommodation of high heat fluxes at moderate temperature differences with fixed heat generation. Another possibility for a two-fluid heat exchanger is to use enhancement to reduce the temperature difference for a fixed heat flux. This reduces the entropy generation and increases the secondary law efficiency. Generally one can consider two different enhancement techniques: passive and active ones. The passive techniques do not require the application of the external power, whereas the active techniques require activator/power supply to bring about the enhancement.

Bergles [1] suggested the following methods of the passive enhancement techniques: treated surfaces, rough surfaces, extended surfaces, displaced enhancement devices, swirl flow devices, coiled tubes, surface tension devices, additives for fluids. Next he proposed the following active techniques: mechanical aids, surface vibration, fluid vibration, electrostatic fields, suction or injection, jet impingement. Bergles [1] also suggested the compound enhancement methods such as application of a rough surface with a twisted-tape swirl flow device, for example.

When we attempt to enhance a rate of condensation, what we usually do is to enlarge the area of heat transfer surface as the first step, and then try to find a way of increasing the heat transfer coefficient. The former is achieved by using extended surfaces and has been employed for a long time. In order to increase the heat transfer coefficient, the thermal resistance corresponding to the condensate film should be decreased. The film thermal resistance can be reduced either by enhancing turbulent mixing in the liquid layer or by thinning the liquid film. The most found method of condensation enhancement is using tubes having extended outer surface, usually in the form of low profile fins. The integral fin tubes are available from manufacturers with the fin density varying from about 350 to approximately 2000 fins per meter. The most important phenomenon during condensation on low profile horizontal integral-fin tube is the so-called Gregorig effect which is caused by the surface tension forces. As it is depicted in Fig. 1b, the liquid film at the top of the fin is convex whereas at its base is concave. Therefore the pressure within the liquid phase is greater at the fin top and lower at the fin base in comparison with the pressure of the vapour. This leads to the pressure gradient along the fin flank which reduces the liquid film thickness. The comparison of the shape and thickness of the condensate film flowing only under gravity and under the influence of the surface tension forces is shown in Fig. 1. Due to the Gregorig effect the thinning of the condensate film is achieved what provides a very high heat transfer coefficient for horizontal integral-fin tubes. The described effect was first discovered by Gregorig almost fifty years ago [2]. Since then more elaborated and sophisticated treatments concerning the optimum fin profiles have been proposed by many investigators [3-5].

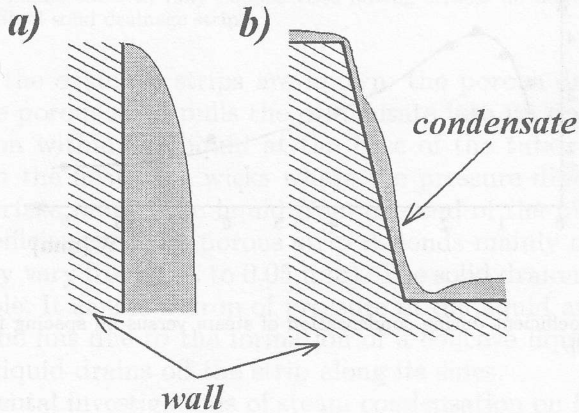



Fig. 1. The shape of the condensate film under conditions: a) gravity induced flow, b) surface tension induced flow on the fin flank – the so-called Gregorig effect.

Bearing in mind above considerations, the use of low profile fins may be regarded, in the terms of Bergles terminology, as a combined method influencing both the surface area enhancement technique and the surface tension action.

However, one should be careful about the condensate retention. It is well known that when a horizontal finned tube comes in contact with a highly wetting liquid, surface tension force will cause the liquid to be retained between the fins at the bottom of the tube. This phenomenon is known as liquid "hold-up", "retention" or "flooding". Condensate retention between the fins has been studied by many investigators, e.g. Honda et al. [6], Rudy and Webb [7]. The flooding may be characterized by the so-called flooding angle, which is measured circumferentially from the top of the tube to a position where the inter-fin space is just completely filled with the condensate. The flooding angle depends mainly on the tube geometry and the capillary constant.

Based on their own results, Webb et al. [8] found out that the heat transfer rate across the zone of retained condensate was negligible for most practical cases. Most of the available experimental data gathered from systematic tests with the family of finned tubes (e.g. Yau et al. [9]) and analytical models (e.g. Adamek and Webb [10]) suggest that the optimum fin spacing exists. Fig. 2 presents a typical variation of the heat transfer coefficient versus the fin spacing according to Adamek and Webb [10]. It is seen that as the spacing between fins decreases, the heat transfer coefficient (based upon the smooth tube surface area) increases to a maximum value at the spacing of about 1.5 mm. Further decrease in fin spacing below this optimum value results in a decrease of heat transfer due to increased flooding between the fins. Therefore it is evident that the problem of avoiding the condensate retention is very important since it can easily cancel the advantage achieved by the application of the low- finned surface. One way to diminish the retention effect relies on the use of a drainage strip.

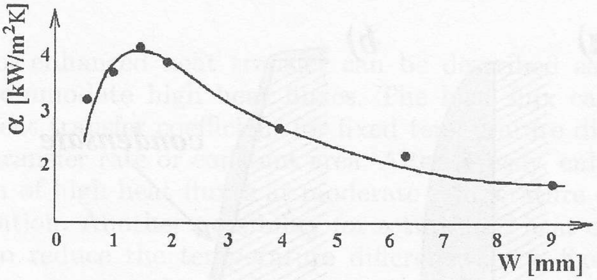



Fig. 2. Heat transfer coefficient during condensation of steam versus fin spacing for water according to Adamek and Webb [10].

## 2. Passive method of condensate drainage augmentation – drainage strip technique

Flooding may be significantly reduced or even completely removed by using the longitudinal condensate drainage strip placed at the very bottom of the tube [11]. Figure 3 shows schematically two characteristic cases. The first one (Fig. 3a) depicts the case where at some critical value of fin spacing the tube is just fully flooded (i.e. the flooding angle $\Phi = 0$). If the drainage strip is used, see Fig. 3b, than even at the critical value of the fin spacing the flooding angle is greater than zero. It is clear that the tube depicted in Fig. 3b should provide greater heat transfer rate than the partially flooded tube without the strip but of the same flooding angle $\Phi_b$ since it has more fins per unit length of the tube.

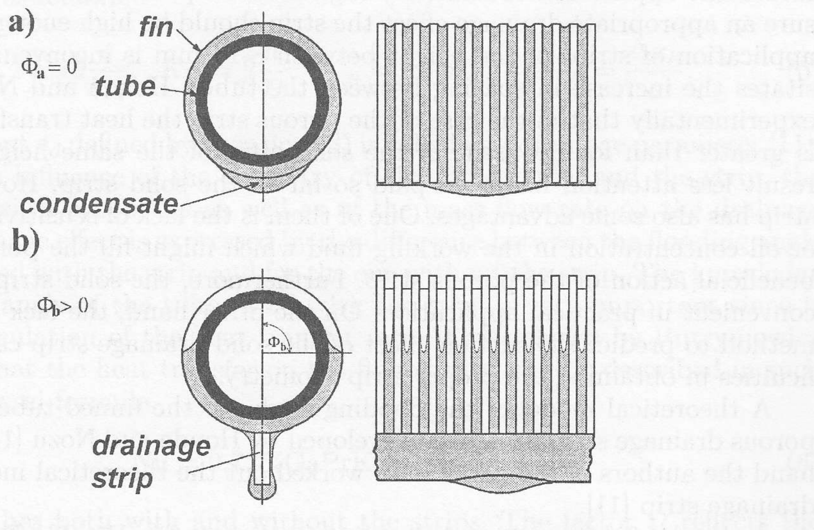



Fig. 3. The horizontal finned tube: a) fully flooded tube having critical fin density, b) the same tube fitted with the longitudinal solid drainage strip.

Two kinds of the drainage strips are known: the porous drainage strip and the solid one. The porous strip pulls the condensate into its pores and creates a low pressure region within the liquid at the base of the tube. This principle is commonly used in the heat pipe wicks where the pressure difference across the liquid-vapour interface pumps the liquid from one end of the pipe to another. It is clear that the efficiency of the porous strip depends mainly on the size of the pores (usually they vary from 0.02 to 0.05 mm). The solid drainage strip works on a different principle. It causes a drop of pressure in the liquid at the intersection of the strip and the fins due to the formation of a concave liquid meniscus. Due to this effect the liquid drains off the strip along its sides.

Some experimental investigations of steam condensation on finned tubes with solid drainage strips were presented by Marto et al. [12]. The maximum increase of the heat transfer coefficient during film condensation was 35%, compared to that of the tube without the strip. Experimental investigations with solid drainage strips for a horizontal smooth tubes were carried out by also Yau et al. [9]. It turned out that in the case of steam, as a condensing medium, the strip of 8 mm height provided only a slight increase of the heat transfer coefficient, compared to the strip of 4 mm height. The solid drainage strip of 8 mm height was also applied to intensify the condensation heat transfer at a smooth tube by Yemielianov et al. [13]. The increase amounting to 20% of the heat transfer coefficient was achieved, compared to the unstripped tube. Honda and Nozu [14] investigated the process of condensation of methanol vapour on horizontal finned copper tubes. They found a moderate increase of the heat transfer coefficient with the small increase of the flooding angle.

It is worth to note that experimental investigations carried out in [9], [12-14] favour the application of relatively high solid strips in belief that in order to assure an appropriate drainage effect the strip should be high enough. However, the application of strips of the height between 8-14 mm is inconvenient as it necessitates the increase in spacing between the tubes. Honda and Nozu [14] proved experimentally that in the case of the porous strip the heat transfer enhancement is greater than for the issue of the solid strip of the same height. Due to this result less attention has been paid so far to the solid strip. However, the solid strip has also some advantages. One of them is the lack of sensitivity to impurities or oil concentration in the working fluid which might fill the pores and stop the beneficial action of the porous strip. Furthermore, the solid strip is cheaper and convenient in practical application. On the other hand, the lack of an analytical method to predict the suction effect of the solid drainage strip causes strong difficulties in obtaining the proper strip geometry.

A theoretical model of the flooding angle for the finned tube fitted with the porous drainage strip have been developed by Honda and Nozu [14]. On the other hand the authors of the paper have worked out the theoretical model of the solid drainage strip [11].

A sketch of the considered situation is presented in Fig. 3b. Due to the action of gravity the condensate flows over the horizontal tube and then along the strip. The curvature of the liquid film flowing over the tube and the strip varies both in the value and sign. It starts from a small negative value at the tube bottom, then reaches maximum and a positive value and finally falls to zero near the end of the strip. The positive influence of the solid strip on the drainage process is therefore caused by the above curvature distribution producing a suction effect beneath the finned tube.

Considering the pressure distribution in the retained liquid phase the authors obtained the following formula for the dimensionless flooding angle [11]

$$\Phi_f = \frac{1}{\pi} \operatorname{arc\,cos}[\varepsilon(1-\varepsilon_d) - 1], \tag{1}$$

where

$$\varepsilon = K_D K_f, \tag{2}$$

$$\varepsilon_d = \frac{K_m + \Delta X_h}{K_f}. \tag{3}$$

It is important to know that the equation (1) is valid for low profile fins, i.e. when

$$W_t \leq 2\frac{\cos\theta}{1-\sin\theta} H_f. \tag{4}$$

The dimensionless quantities in equations (2) and (3) correspond respectively to the curvature of the condensate meniscus in the interfin channel $K_f$, the curvature of the tube, $K_D$, the maximum curvature on the strip, $K_m$, and the hydrostatic region length on the strip $\Delta X_h$. The definitions of these dimensionless parameters are as follows

$$K_f = 2\cos\theta \frac{a}{W_t},\;\; K_D = 2\frac{a}{D},\;\; K_m = a\kappa(x_m),\;\; \Delta X_h = \frac{\Delta x_h}{a}. \tag{5}$$

The coefficient $\varepsilon_d$ defined by equation (3) is called as a drainage parameter [11]. It describes the influence of the geometry of the finned tube and the strip, the condensate physical properties as well as of the mass flow rate on the drainage effect. The drainage effect is expressed by the difference between the flooding angle for the tube fitted with the strip and for the one without the strip. The knowledge of the flooding angle of the tube fitted with the strip is very important since it permits the calculation of the heat transfer rate. It was shown by Butrymowicz and Trela [5], that the heat transfer on the finned tube can be described in such case by the general formula

$$\mathrm{Nu} = 0.725(\mathrm{GaPrK})^{0.25}\Omega\Phi_f, \tag{6}$$

valid for the tubes both with and without the strips. The factor $\Omega$ reflects the heat transfer enhancement due to the fin geometry. Details of derivation of Eq. (6) one can find in [5].

The effect of the tube and strip geometry on the drainage process was studied both theoretically as well as experimentally by the authors [5, 11]. These investigations provided that the drainage parameter decreases with the tube curvature increase. From a physical point of view this means, that the meniscus created on the strip is "more concave" for greater tubes. However the flooding angle increases with the decrease of geometrical parameter $\varepsilon$. This parameter depends on the dimensionless tube curvature – see Eq. (2). From the above considerations the authors concluded that for the same fin geometry the flooding angle increases with decreasing tube curvature for tubes without a strip. Therefore the use of the solid drainage strip is more profitable for tubes with small outside diameters. The other important geometrical parameter is the strip height. Numerical calculations performed by the authors [5] show that, for the most practical cases, this height equals 2÷4 times the capillary constant.

In order to avoid tedious numerical calculations for the condensate drainage process the authors have developed a correlation for the drainage parameter based on the database of 594 numerical results [15]

$$\varepsilon_d = \frac{1}{K_f}\left\{C_0 K_D^{C_1}\alpha^{C_2}\gamma^{C_3} + \exp[f_1(\gamma) + f_2(\alpha)K_D\right\}, \tag{7}$$

where

$$f_1(\gamma) = C_4 + C_5 10^{-5}\gamma, \tag{7a}$$

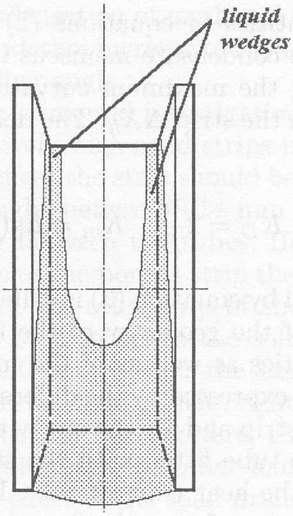



Fig. 4. Liquid wedges on the intergal-fin tube.

$$f_2(\alpha) = C_6 + C_7\alpha + C_8\alpha^2. \tag{7b}$$

The constants in the Eq. (7) are given in Tab. 1. The Eq. (7) is valid for the following range of dimensionless parameters

Table 1. Values of the constants in the Eq. (7)

| $C_0$ | $C_1$ | $C_2$ | $C_3$ | $C_4$ | $C_5$ | $C_6$ | $C_7$ | $C_8$ |
|---|---|---|---|---|---|---|---|---|
| – 4.822568 | 0.493038 | 0.607955 | 0.144530 | 0.581317 | 0.002175 | – 0.396058 | 150.389 | – 6854.0 |

$$0.042 \le K_D \le 0.756, \tag{8a}$$

$$0.169 \cdot 10^{-3} \le \alpha \le 15.49 \cdot 10^{-3}, \tag{8b}$$

$$1.04 \cdot 10^5 \le \gamma \le 5.56 \cdot 10^5. \tag{8c}$$

Observation of the static liquid retention on finned tubes, done by Masuda and Rose [16] revealed that the liquid is not only retained at the bottom part of the tube, but also on the upper part of it. In the last case the liquid has the form of small liquid wedges between the flanks of the fins and the tube surface – Fig. 4. Subsequently they developed a model of these wedges. They also defined an active surface area enhancement as the unblanked area of a finned tube (i.e. the area of the fin tips together with the unblanked portions of the fin flanks and the interfin tube surface) divided by the area of a smooth tube of fin root diameter.

Butrymowicz et al. [17] have developed also the model of the liquid wedges formed on the horizontal tube fitted with the solid drainage strip. Basing on the own model as well as own and other investigators experimental results Butrymowicz et al. [18] have concluded that the heat transfer augmentation is proportional to the increase of the flooding angle and not active surface area enhancement. Therefore the authors suggested that the problem of the effect of the liquid wedges on the condensation heat transfer on horizontal finned tube should be still regarded as an open question.

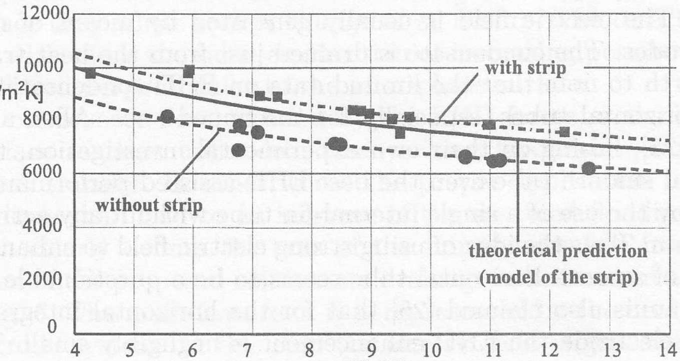



Fig. 5. Heat transfer coefficient for horizontal finned tube (OD 14.0 mm, fin density 1000 fins per meter). Dashed line corresponds to the best fit of the experimental data. The thickness of the solid drainage strip 1.0 mm, the height 4.0 mm.

Figure 5 presents the authors' results of the experimental investigations of condensation of refrigerant R11 on a single finned tube fitted with the solid drainage strip and without the strip. It appears that the solid strip application causes the increase of the heat transfer coefficient by about 19%, as compared to the tube without the strip. On the other hand, it follows from the calculations, based on the strip model, that the heat transfer coefficient should increase by about 13%. The discrepancy between the theoretical and experimental heat transfer coefficient is partially due to simplifying assumptions in the theory and partly to the limited accuracy of the prediction of the refrigerant capillary constant. Taking into account the above notes, it may be said that the experimental results agree reasonably well with the model prediction.

## 3. Active method of condensate drainage augmentation – electrohydrodynamical technique

The effect of electric field on condensation heat transfer was first reported by Velkoff and Miller in 1965 [19]. They quoted the increase of the mean heat transfer coefficient by about 200% during condensation of refrigerant R113 on a flat, vertical plate upon the application of an intense electric field perpendicular to the plate. Experimental investigations of electrohydrodynamic (EHD) enhanced condensation heat transfer have been carried out by many other researches, for example: Choi [20], Holmes and Chapman [21], Seth and Lee [22]. In all these cases, the condensation enhancement was found to be caused by an electrically induced destabilisation of the condensate film. Recently very interesting investigation has been done by Didkovsky and Bologa [23, 24]. They obtained EHD enhancement as much as 20 times during condensation on the smooth vertical tube. It should be mentioned that most investigations deal with the condensation augmentation by application of the electric field to the vertical plate or vertical tubes [25]. The electric field is usually generated by means of coaxial rod and mesh electrodes. The condensate is drained just from the heat transfer area.

It is worth to note that the limited data on EHD condensation enhancement exist for horizontal tubes, especially for the finned ones. Allen and Karayiannis concluded [25], basing on their own experimental investigation, that in the case of horizontal smooth tube even the best EHD-assisted performance falls short of that given by the use of a single internal-fin tube without any active heat transfer augmentation. Thus the idea of using strong electric field to enhance heat transfer in the case of smooth horizontal tube seems to be a questionable problem. Allen and Karayiannis also claimed [25] that for the horizontal integral-fin tube with the coaxial electrode the EHD enhancement is negligibly small. They suggested that the phenomena caused by the surface-tension forces play the dominant role also in this case.

It is quite obvious that when the strong coaxial electric field is applied to the integral-fin tube the EHD forces act against the surface tension forces, i.e. pulling out the condensate thus reducing the positive Gregorig effect. Therefore it is not now surprising, as claimed by Cooper [26], that the EHD assisted condensation enhancement applied to the horizontal integral-fin tube is negligible. It should be stressed that this conclusion is valid only for the case when the coaxial electric field caused by the use of coaxial wires or coaxial mesh electrode system is used. The above conclusions shed more light on the EHD condensation mechanism and are very helpful in formulation of a new active method for EHD enhancement of condensation on the horizontal integral-fin tube. This method is presented in a very short manner in this paper.

The physical basis of electrically enhanced condensation due to the EHD force generated by an electric field is given by the equation [25]

$$f_e = qE - \frac{1}{2}E^2 \nabla \varepsilon + \frac{1}{2} \nabla \left[ E^2 \left( \frac{\partial \varepsilon}{\partial \rho} \right)_T \rho \right]. \tag{9}$$

It may be further written in a more detailed form for non-polar fluids [27] as

$$\begin{aligned} f_e &= f_1 + f_2 + f_3 + f_4 = \\ &= qE - \frac{1}{2}E^2 \nabla \varepsilon + \frac{1}{6}\varepsilon_0(\varepsilon_r - 1)(\varepsilon_r + 2) \nabla E^2 + \frac{1}{6}\varepsilon_0 E^2 \nabla [(\varepsilon_r - 1)(\varepsilon_r + 2)] \end{aligned} \tag{10}$$

The first component represents the electrophoretic force $f_1$ exerted by an electric field upon electric charges in the fluid. The second component $f_2$, comes from the spatial distribution of electric permittivity $\varepsilon$, which are not caused by the electrostriction. This component acts perpendicularly to the inter-phase surface from the material of higher permittivity (liquid) to the material of lower permittivity (vapour). This component acts to disturb the condensate film. The third component $f_3$ is the dielectrophoretic force which is due to the non-uniformity of the electric field. This force pushes condensate towards higher electric field strength. The last component $f_4$ is the electrostriction force which occurs whether or not an applied field is uniform. This force depends on the non-uniformity of electrical permittivity. In the case of film condensation this component is significant at the inter-phase surface only when abrupt change of $\varepsilon_r$ occurs. The direction of this force coinciding with the is the same as direction of increasing permittivity (from vapour to liquid) and perpendicular to the inter-phase surface.

Due to the condensate flow, induced by the EHD forces, a significant area of the tube surface remains unflooded, thus making favourable conditions for condensation heat transfer. In other words the proper application of the EHD condensate drainage shall lead to an increase of the so-called flooding angle – a parameter which clearly defines the area of the finned tube free from the condensate. The idea of the proposed condensate drainage enhancement technique is illustrated in Fig. 6. The main feature of it is represented by the arrangement of the tube – electrode system. In this case the electric field is generated between the rod-type electrode and the lower part of the finned tube. Therefore, taking into account the EHD mechanism one may expect that the majority of the tube surface is not affected by the electric field with the exception of the small part of it at the tube bottom. As it is depicted above, this part usually plays a minor role in heat transfer also without the electric field since the interfin spaces are usually flooded totally by the condensate. Thus we have two positive phenomena with respect to the heat transfer augmentation in the considered case. The first one follows from the Gregorig effect, which takes place in the upper part of the tube, (not disturbed by the EHD forces) and second one is due to the EHD forces pulling down the condensate from the bottom of the tube. It is seen that in this method the EHD forces act only on the condensate drainage process and not on the condensation phenomenon itself. This feature expresses the main difference between the old methods of the EHD enhancement and the new one, which is proposed by the authors.

The main goal of the experimental investigations was to provide data to confirm the theoretical expectations of EHD condensation enhancement according to the idea shown in Fig. 6. The experimental investigation was conducted on the test rig depicted in Fig. 7. Experiments covered measurements of the heat transfer coefficient during condensation of refrigerant HCFC-123 at the average saturation temperature + 40$^o$C. The detailed description of the experimental rig one can find in [28]. The dimensions of the investigated tubes are given in Tab. 2. The brass rod-type electrode of the outer diameter 5.0 mm was used. Measurements covered the rate of cooling water flow in the range 3.7÷13.0 l/min.

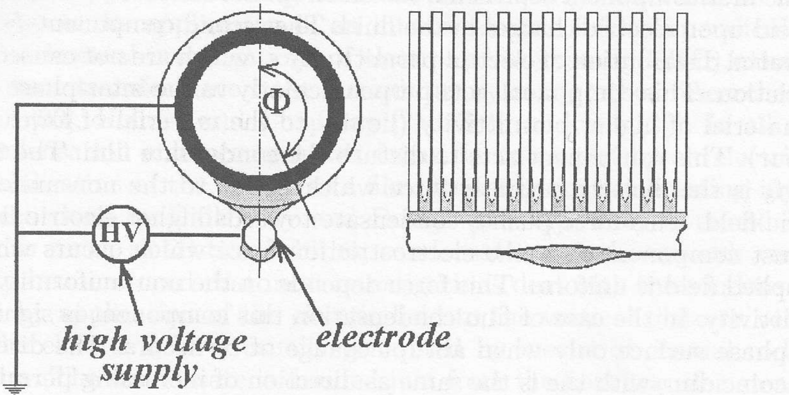



Fig. 6. The novel idea of the condensation enhancement by the EHD condensate drainage.

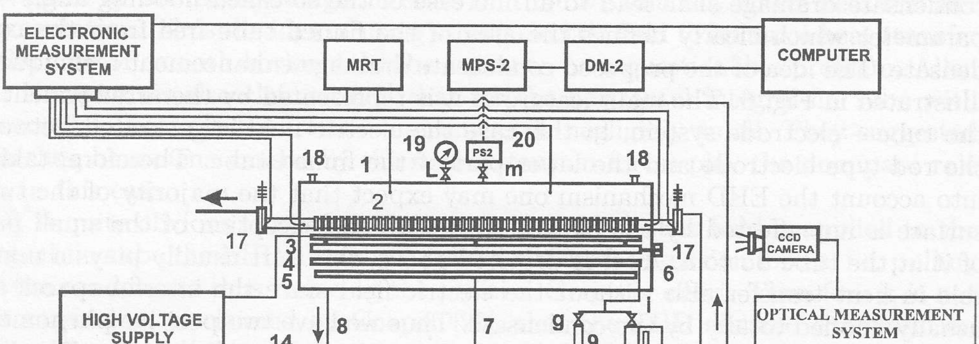



Fig. 7. The schematic diagram of the test rig: 1 – condenser, 2 – tested tube, 3 – electrode, 4 – condensate dish, 5 – vapour distributor, 6 – electrode mechanism, 7 – vapour supplying tube, 8 – condensate outflow, 9 – dehydrator, 10 – condensate calibrating tank, 11 – levelling line, 12 – vapour generator, 13 – hot water vessel, 14 – vacummeter, 15 – vacuum pump, 16 – turbine water flow rate meter, 17 – water temperature measurement tank, 18 – air purging valve, 19 – mano-vacuum meter, 20 – pressure sensor, DM-2 – electronic multi-manometr, MRT – digital encounter Minitrol.

Table 2. Geometry of the investigated tubes

| Tube | unit | 1 | 2 | 3 | 4 |
|---|---|---|---|---|---|
| fin density | fin/m | bare | 1026.7 | bare | 740 |
| inner diameter | mm | 13.0 | 11.02 | 15.60 | 13.80 |
| fin root diameter | mm | – | 13.42 | – | 16.0 |
| outer diameter | mm | 16,0 | 16.17 | 18.76 | 19 |
| fin pitch | mm | – | 0.97 | – | 1.35 |
| fin height | mm | – | 1.38 | – | 1.5 |
| fin thickness at fin root | mm | – | 0.40 | – | 0.46 |
| fin thickness at fin top | mm | – | 0.21 | – | 0.29 |
| material | | cooper | cooper | nickeline | nickeline |

The experimental data of heat transfer coefficients were compared to the ones given by the Nusselt theory for the smooth horizontal tube. Two enhancement factors $\beta_{\mathrm{Nu}}$ and $\beta_e$ – given in Tab. 3, were introduced in the analysis

Table 3. Results of condensation heat transfer measurement

| Tube | $U$ [kV] | $s$ [mm] | $\beta_{\mathrm{Nu}}$ | $\beta_{\mathrm{exp}}$ |
|---|---|---|---|---|
| No 1 | 0 | 9.5 | 1.09 | 1.00 |
| No 2 | 0 | 9.5 | 3.71 | 1.00 |
| | 25 | 9.5 | 4.71 | 1.27 |
| No 3 | 0 | 8.6 | 0.98 | 1.00 |
| No 4 | 0 | 8.6 | 2.64 | 1.00 |
| | 22 | 8.6 | 5.48 | 2.08 |

- the ratio $\beta_{\mathrm{Nu}}$ is referred to the Nusselt model for the smooth horizontal tube

$$\beta_{\mathrm{Nu}} = \frac{\alpha_{\mathrm{exp}}}{\alpha_{\mathrm{Nu}}}, \tag{11}$$

- the ratio $\beta_{\mathrm{exp}}$ is referred to the measured value of heat transfer coefficient

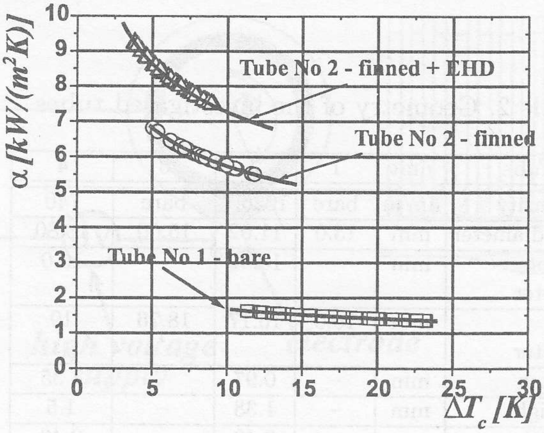



Fig. 8. Results of experimental investigations of tubes No. 1 and No. 2, $U = 25$ kV.

in the absence of electric field

$$\beta_{\text{exp}} = \frac{\alpha_{\text{exp}}}{\alpha_{\text{exp},U=0}}. \tag{12}$$

Experimental results of condensation of refrigerant HCFC-123 for tubes No. 1 and No. 2 (of OD 16 mm) are presented in Fig. 8. It is seen that the heat transfer coefficient for the finned tube increased by 240 % in comparison with the bare one in the absence of the electric field. The application of voltage of 25.0 kV caused the further increase of the condensation heat transfer coefficient by 27 %. The total increase of the heat transfer coefficient in comparison with the bare tube was found to be 330 %.

The observation of condensation phenomenon proved that the amount of the retained condensate between the fins was reduced by the EHD effect almost completely. In this case the value of the flooding angle is $\Phi_{fe} \approx 1.0$. The theoretical value of the flooding angle calculated from the Eq. (1) with the drainage parameter $\varepsilon_d = 0$ gives $\Phi_f = 0.74$ and estimated value from the photograph is equal to $\Phi_f = 0.78$. Basing on these results and the Eq. (6) one can estimate the theoretical increase of the condensation heat transfer coefficient due to elimination of the condensate retention as $\alpha_{fe}/\alpha_f = \Phi_{fe}/\Phi_f = 1.00/0.78 = 1.28$ which agrees very well with the experimental results.

Experimental results obtained for tubes No. 3 and No. 4 (i.e. tubes having OD 19 mm) are presented in Fig. 9. It is seen that the heat transfer coefficient for the finned tube increased by 170 % in comparison with the bare one. The application of voltage of 22.0 kV caused further increase of the condensation heat transfer coefficient by 110 %. This is a very impressive result. The total increase of the heat transfer coefficient (due to the fins and EHD effects) in comparison with the bare tube was found to be 460 %.

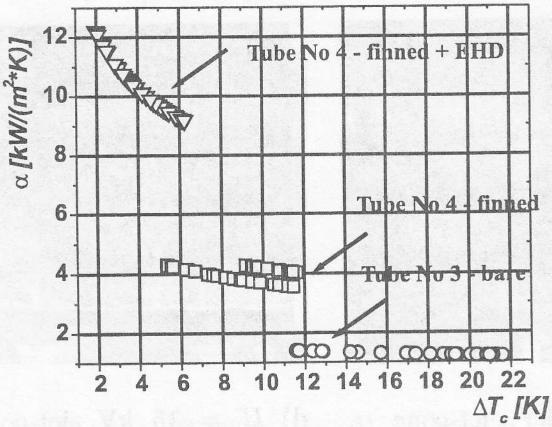



Fig. 9. Results of the experimental investigations of tube No. 3 and No. 4, $U$ = 22 kV.

It is evident that the condensation enhancement can significantly reduce mean temperature difference $\Delta T_c$ between saturated condensing vapour and the outer surface of the tube. This temperature difference may be reduced as much as three times for the EHD condensation enhancement for tube No. 2 and almost six times for the tube No. 4 in comparison with the smooth tubes of the same diameter. Since for the refrigeration or heat pump cycles the temperature difference $\Delta T_c$ in the condenser may be considered as a part of the total temperature difference between the heat source and the heat sink then the significant decrease of $\Delta T_c$ due to the heat transfer enhancement by the EHD method leads to the strong increase of the cycle efficiency.

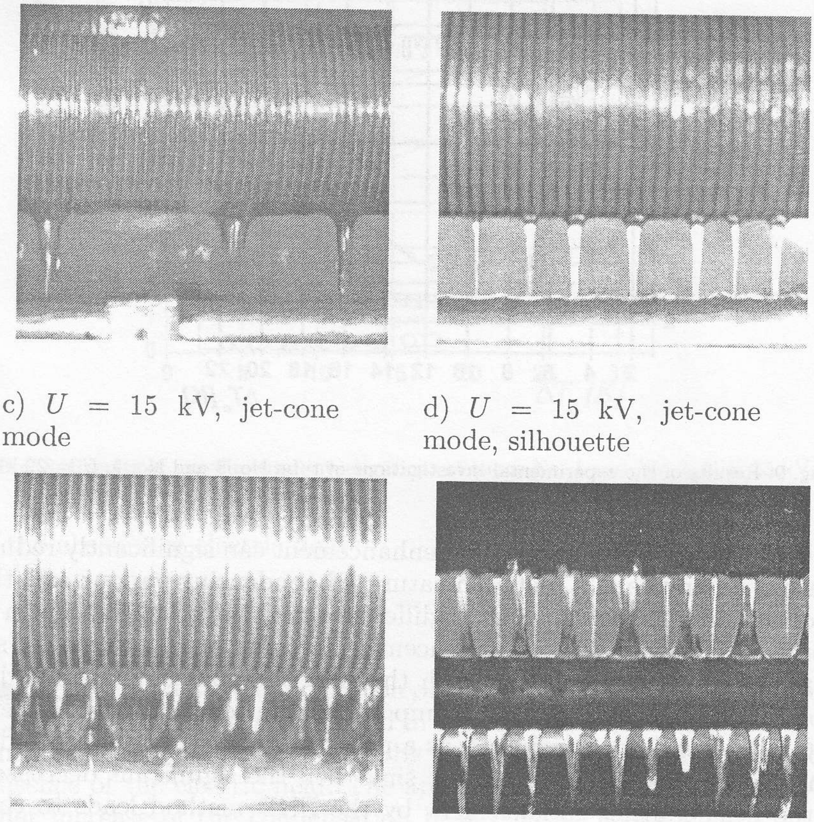
The very important role in the investigated phenomenon plays the flow pattern of the condensate leaving the tube. The condensate flow patterns for tube No. 2 and No. 4 are shown in Fig. 10 and 11, respectively. They are the same for the given range of the condensate mass flow rate and the applied voltage. It is very important to note, that the condensate flow patterns observed for tube No. 2 (Fig. 10d) were slightly different in comparison with the tube No. 4 (Fig. 11d). In the latter case the condensate jet was formed from each fin tip. Therefore the jet spacing was the same as the fin pitch. In the case of tube No. 2 the jet spacing was sufficiently larger than the fin pitch. This may be attributed as the main reason for such significant improvement of condensation heat transfer measured for tube No. 4.

## 4. Conclusions

This review has given a short overview of enhanced condensation technology by applying both passive and active techniques.

a) $U = 0$ kV, droplet mode

b) $U = 10$ kV, jet mode



c) $U = 15$ kV, jet-cone mode

d) $U = 15$ kV, jet-cone mode, silhouette

Fig. 10. The condensate flow pattern from tube No. 2: for $U = 0$: $\Gamma = 9.5 \div 19.3 \cdot 10^{-3}$ kg/$(m \cdot s)$, for $U = 10$ kV: $\Gamma = 11.4 \div 22.7 \cdot 10^{-3}$ kg/$(m \cdot s)$, for $U = 15$ kV: $\Gamma = 11.6 \div 22.2 \cdot 10^{-3}$ kg/$(m \cdot s)$.

The passive method may be achieved for instance by the application of the solid strip. Due to the action of this strip the condensate retained between the fins is drained off, leaving more surface available for heat transfer. The own rational model of the solid strip has been presented and the importance of its parameters has been discussed. The most important parameter in the model is the drainage parameter $\varepsilon_d$. It reflects the influence of fin and tube geometry as well as the physical properties of the condensate on the drainage process. The condensation heat transfer coefficient increased for investigated finned tube by approximately 20 % as the result of the application of the solid drainage strip.

The active technique of condensation enhancement may be achieved by applying the electrohydrodynamical condensate drainage augmentation. The performed experimental investigations proved the usefulness of the new idea of condensation heat transfer enhancement by the EHD condensate drainage, as depicted in Fig. 6. The effectiveness of this method depends on the geometry of the tube as well as

a) $U = 0$ kV, droplet mode b) $U = 6$ kV, jet mode

c) $U = 16$ kV, jet-cone mode d) $U = 22$ kV, jet-cone mode

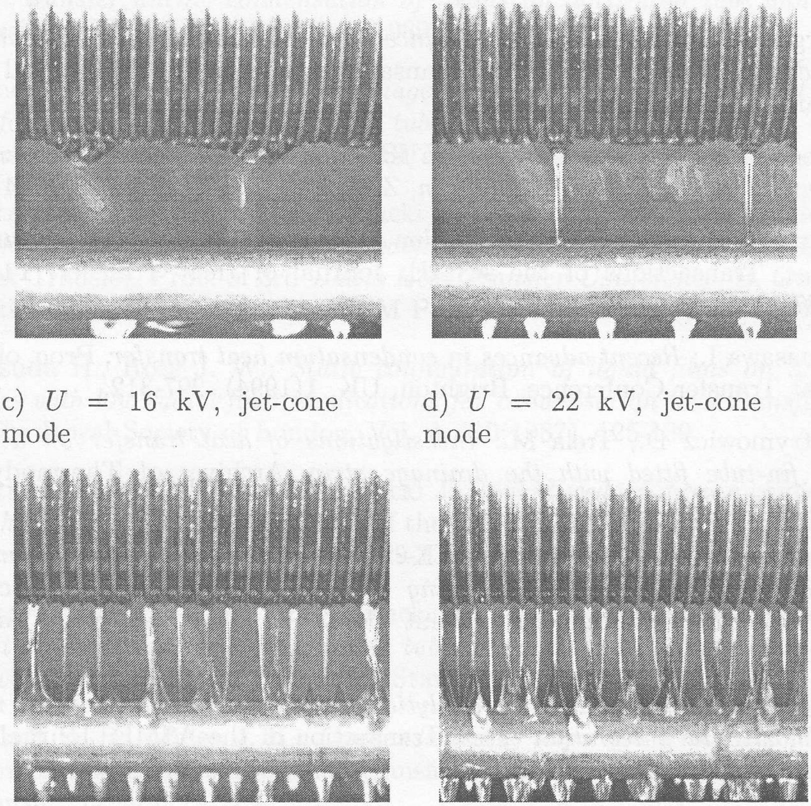


Fig. 11. The condensate flow pattern from tube No. 2: for $U = 0$: $\Gamma = 7.0 \div 14.6 \cdot 10^{-3}$ kg/$(m \cdot s)$, for $U = 22$ kV: $\Gamma = 7.5 \div 18.3 \cdot 10^{-3}$ kg/$(m \cdot s)$.

on the applied electrode potential. Enhancement of the heat transfer coefficient ranges from 27 % to 110 %. The total increase of the heat transfer coefficient in comparison with the bare tube was found to be even 460 %. The average temperature difference between condensing vapour and outside tube wall for finned tube applied with the EHD condensate drainage was reduced from three to six times in comparison with the bare tube without the EHD effect. It is obvious that many aspects of EHD condensate drainage need further investigation. The one concerns the problem of the optimal geometry of the tube – electrode system.

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