STUDY ON HEAT TRANSFER IN TURBULENT FILM FLOW

Stasys Sinkunas
Kaunas University of Technology, Department of Thermal and Nuclear Energy, Donelaicio 20, LT-44239, Kaunas, Lithuania
stasin@ktu.lt

Jonas Gylys
Kaunas University of Technology, Department of Thermal and Nuclear Energy, Donelaicio 20, LT-44239, Kaunas, Lithuania
jonas.gylys@ktu.lt

Algimantas Kiela
Kaunas College, Faculty of Technology, Pramones 22, LT-50387, Kaunas, Lithuania
algimantas.kiela@tm.kauko.lt

Abstract. *The purpose of the present study is to obtain a comprehension for the heat transfer developments in gravitational liquid film flow. Analytical study of the heat transfer for the turbulent film flow was performed. The method for the calculation of the local heat transfer coefficient to turbulent film falling down a vertical plane surface was proposed. The decrease of the heat flux density in the film was observed as the distance from the wetted surface increased. Experimental investigation of the heat transfer in the entrance region for the turbulent film has been carried out as well. The description of experimental set-up is presented in the paper. The experiments were performed in water film flowing down a surface of vertical tube. The results of experiments are discussed with respect to the local heat transfer coefficient dependence upon Reynolds number and initial velocity of the film. The heat transfer stabilization length was determined experimentally.*

Keywords*: heat transfer, turbulent film, entrance region, initial velocity, stabilization length*

1. Introduction

Heat transfer phenomena play an important role in many industrial and environmental problems. As an example, consider the vital area of energy production or conversion. There is not a single application that does not involve heat transfer effects in some way. In the generation of electrical power, whether it is performed by nuclear fission or fusion, the combustion of fossil fuels, magnetohydrodynamics processes, or by the use of geothermal energy sources, there are numerous applications where heat transfer problems must be solved. These problems involve conduction, convection, radiation processes and are related to the design of systems such as boilers, condensers and evaporators. One is often confronted with the need to maximize heat transfer rate and to maintain the integrity of materials in high temperature environments. Heat transfer processes also affect the performance of propulsion systems, such as the internal combustion, gas turbine and rocket engines. Heat transfer problems arise in the design of conventional space and water heating systems, refrigeration and air conditioning systems, cooling of electronic equipment.

Turbulent film flow is of great technological importance, as it occurs frequently under normal operating conditions for a variety of heating and cooling devices in such diverse fields as aerospace, naval, nuclear, mechanical and chemical engineering. The main advantage of turbulent over laminar flows is that they are capable of providing enhanced heat and mass transfer rates. However, this is at the expense of the increased friction losses accompanying the turbulent flows.

The heat transfer in the film flow down a vertical surface has been the subject of analytical and experimental investigations (Tananayko and Vorontszov, 1975; Gantchev, 1987), but these studies have been done in the stabilized film flow region. At the same time, there are few data about the behavior of liquid film in the entrance region. An estimation of the entrance region length is important in many engineering applications (Gimbutis, 1988). Practically, the operation of real film heat exchangers is based on vertical tubes (Alekseenko *et al*., 2001). In the case when liquid film flows down a vertical tube the curvature of its surface and the film itself affects heat transfer characteristics on the surface, shear stress distribution and correspondingly thickness of the liquid film. Simultaneously the intensity of the heat exchange between a wetted surface and liquid film is altered.

Some papers concerning the film flow on vertical or inclined plane surfaces have been published. They are collected in references (Bertani and De Salve, 2001; Chinnov *et al*. 2001).

Heat transfer mechanism of a liquid film flowing down horizontal tubes was investigated (Rifert *et al*., 2001). A method based on the breakdown of the heat boundary layer by longitudinal fins in order to enhance the local heat transfer coefficient was developed in this study. The comparison of various correlations for the calculation of the heat transfer in liquid films with high Prandtl number was carried out (Wadekar, 2000).

2. Analytical determination of heat transfer in a turbulent film

The heat flux across the turbulent film on a vertical plane surface can be determined by the expression

$$q = -c\rho\left(a + a_t\right)\frac{dT}{dy} \tag{1}$$

or

$$q = -\frac{c\rho v^*}{Pr}\left(1+\frac{Pr}{Pr_t}\frac{\nu_t}{\nu}\right)\frac{dT}{d\eta} \tag{2}$$

where c - specific heat, J/kg·K; ρ - liquid density, kg/m^3; a - thermal diffusivity, m/s^2; a_t - turbulent thermal diffusivity, m/s^2; $v^* = \left(\tau_w/\rho\right)^{1/2}$ - dynamic velocity; τ_w - shear stress on tube surface, Pa, $Pr = \nu/a$ - Prandtl number; $Pr = \nu_t/a_t$ - turbulent Prandtl number; ν - kinematic viscosity, m^2/s; ν_t - turbulent kinematic viscosity, m^2/s.

Integrating the Eq (2) and making simple rearrangements we obtain the expression, defining the temperature field in the film

$$\int_{T_w}^{T} dT = -\frac{q_w Pr}{c\rho v^*}\int_0^{\eta}\frac{q/q_w}{1+\frac{Pr}{Pr_t}\frac{\nu_t}{\nu}}d\eta \tag{3}$$

or

$$T_w - T_f = -\frac{q_w Pr}{c\rho v^*}\int_0^{\eta}\frac{q_w/q}{1+\frac{Pr}{Pr_t}\frac{\nu_t}{\nu}}d\eta \tag{4}$$

where q_w and q - heat flux density on the wetted surface and in the film respectively, W/m^2; T_w and T_f - temperature of the wetted surface and the film respectively, K; $\eta = v^* y/\nu$ - dimensionless distance from the wetted surface; y - distance from the wetted surface, m.

The temperature difference between the wetted surface and the gravitational liquid film one can define as

$$\Delta T = \frac{\int_0^{\delta} w\left(T_w - T_f\right)dy}{\int_0^{\delta} w dy} = \frac{\int_0^{\eta_\delta}\varphi\left(T_w - T_f\right)d\eta}{\int_0^{\eta_\delta}\varphi d\eta} \tag{5}$$

where w - film velocity, m/s; $\varphi = w/v^*$ dimensionless film velocity; δ - film thickness, m; $\eta_\delta = v^*\delta/\nu$ - dimensionless film thickness.

Let us denote that

$$\int_0^{\eta}\frac{q/q_w}{1+\frac{Pr}{Pr_t}\frac{\nu_t}{\nu}}d\eta = \psi \tag{6}$$

then from Eqs. (4) and (5) we obtain

$$\Delta T = \frac{q_w Pr \int_0^{\eta_\delta} \psi\varphi d\eta}{c\rho v^* \int_0^{\eta_\delta} \varphi d\eta} \tag{7}$$

Considering that $\alpha = q_w / \Delta T$, the Eq. (7) will appear as follows

$$\alpha = \frac{0.25 c\rho v^* Re}{Pr \int_0^{\eta_\delta} \psi\varphi d\eta} \tag{8}$$

where α - heat transfer coefficient, W/(m²·K); $Re = 4\Gamma/(\rho\nu)$ - Reynolds number; Γ - wetting density, kg/(m·s).

The hydraulic diameter of plane film d or the modified thickness of the film $(\nu^2/g)^{1/3}$ can be used for the heat transfer calculation in the Nusselt number as a characteristic variable keeping in mind that surrounding density is small as compared to liquid film density, the gravity and friction forces keep their balance as well. However, from practical point of view is more reasonable using the modified film thickness. This allows avoiding the complicated calculations of liquid film hydraulic diameter. It is convenient to calculate the heat transfer coefficient using the modified Nusselt number

$$Nu_M = (\alpha/\lambda)(\nu^2/g)^{1/3} \tag{9}$$

where λ - thermal conductivity, W/(m·K).
Equation (9) can be rearranged as follows

$$Nu_M = \frac{0.25\sqrt[3]{\eta_\delta}}{\int_0^{\eta_\delta} \psi\varphi d\eta} \tag{10}$$

As it follows from Eq. (6) estimation of the variation of ratios q/q_w and ν_t/ν across the film is necessary for the heat transfer coefficient calculation.

The heat balance equation for the elementary volume of the film $bdxdy$ can be written as

$$-\frac{\partial q}{\partial y} dy(bdx) = c\rho w(bdy)\frac{\partial T}{\partial x} dx \tag{11}$$

Let us write the heat balance equation for the elementary volume of the film $b\delta dx$

$$q_w(bdx) = cG\frac{\partial T}{\partial x} dx \tag{12}$$

where b - elementary width of the film, m; G - liquid mass flow rate, kg/s.
Assume that $\partial T/\partial x = dT_f/dx$ (Kays, 1972). Then taking into account Eqs. (11) and (12), we obtain

$$\int_{q_w}^{q} dq = -\frac{q_w b\rho}{G}\int_0^{Y} wdy \tag{13}$$

Considering that $G/b = \Gamma$ the Eq. (13) can be written as follows

$$\frac{q}{q_w} = 1 - \frac{\rho}{\Gamma} \int_0^y w dy \tag{14}$$

The dimensionless form of the Eq. (14) will appears as

$$\frac{q}{q_w} = 1 - \frac{4 \int_0^\eta \varphi d\eta}{Re} \tag{15}$$

Variation of the heat flux ratio in the turbulent film flow on vertical plane surface for the different Reynolds numbers is shown in Fig. (1)

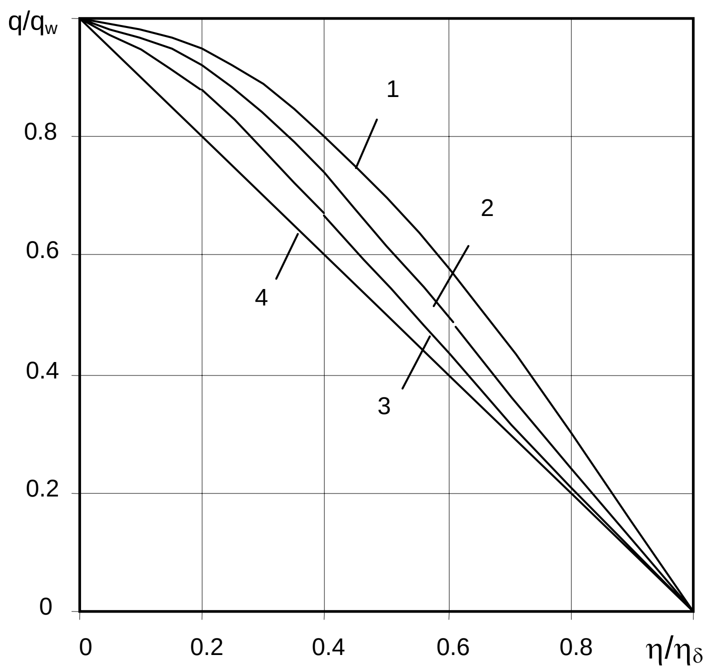

Figure 1. Dependence of the heat flux variation upon the distance from the wetted surface for the different Reynolds numbers: *1* – *Re*=1620; *2* – *Re*=3123; *3* – *Re*=9440; *4* - $q/q_w = 1 - \eta/\eta_\delta$

3. Experimental set-up and procedure

In order to generate the liquid film flow the experimental arrangement (Fig. 2) was applied. The stainless steel tube 30 mm in outside diameter with the length of 1000 mm was employed in the experiment as a calorimeter. The fixing bolts at the end of tested tube allowed the possibility to regulate and to guarantee verticality of the tube. Water was pumped up to a liquid distributor by feed-pump. At the top end of the tube a slot distributive mechanism was installed to generate the uniform film flow. After flowing down the test tube, the water was gathered back to the reservoir. The gutter at the calorimeter end ensured a smooth falling of the water into the reservoir. The surplus water was discharged to the sewerage by exhaust-pump while the fresh water was supplied from water supply directly.

Preliminary investigation has shown that the use of water from plumbing did not influence on the demanded experiment accuracy. That is why fresh water was employed in the research. The temperature of falling down film was measured by two calibrated thermocouples. The location of a thermocouple in the liquid distributor ensured the measurement of film temperature at the inlet. The thermocouple installed at the end of calorimeter had determined a film temperature at the exit correspondingly. As heat flux along the tested section did not change, so to the demanded accuracy of the experiment it was assumed that the bulk mean temperature of the liquid film confirms linear regularity. That circumstance allowed determining the liquid film temperature at any cross-section of the tested section accurately.

For the heat transfer research of falling down water film on the surface of vertical tubes the electric circuit (Fig. 3) was applied. The electric current supplied for the calorimeter provided a steady heat flux on the experimental section. In order to convert an alternating current into a direct current the rectifier was used. Voltages drop on the known value resistor fitted in as shunt determined the electric current strength in the circuit. Its readings were taken from the millivoltmeter. Voltage value on the calorimeter was measured by voltmeter.

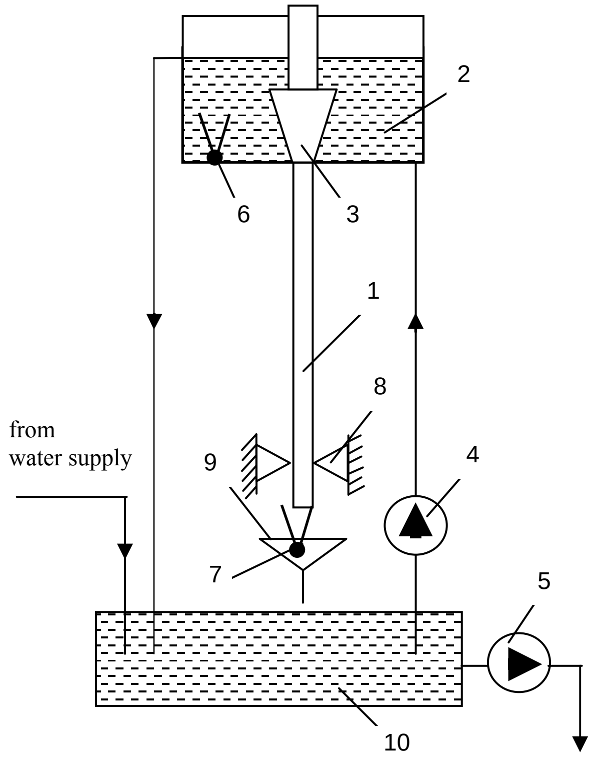

Figure 2. Schematic diagram of experimental set-up: *1* - calorimeter; *2* - liquid distributor; *3* - slot distributive mechanism; *4* - feed-pump; *5* - exhaust-pump; *6* - inlet thermocouple; *7* - outlet thermocouple; *8* - centering bolts; *9* - gutter; *10* - liquid reservoir

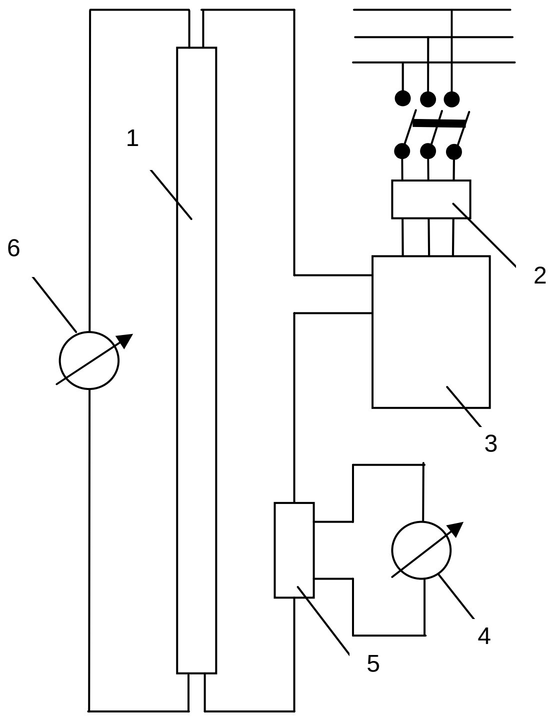

Figure 3. Electric circuit: *1* - calorimeter; *2* - voltage regulator; *3* - rectifier; *4* - millivoltmeter; *5* - shunt; *6* - voltmeter

To investigate the heat transfer intensity to the liquid film falling down a vertical surface the calorimeter (Fig. 4) was constructed. The stainless steel tube 30 mm in outside diameter and 1000 mm length was employed in the experiment as the test section. The temperature of inner tube wall was determined by 0.12 - 0.15 mm copper-constantan thermocouples fixed by pressing them to the inner wall surface with elastic plastic rings. In order to avoid the electric current influence on the thermocouples their heads were covered with a thin layer of dielectric lacquer. Thirty thermocouples were located along the inner surface of the calorimeter, by three of them in each of ten cross sections respectively. The millivoltmeter registered the values of thermocouples electromotive force.

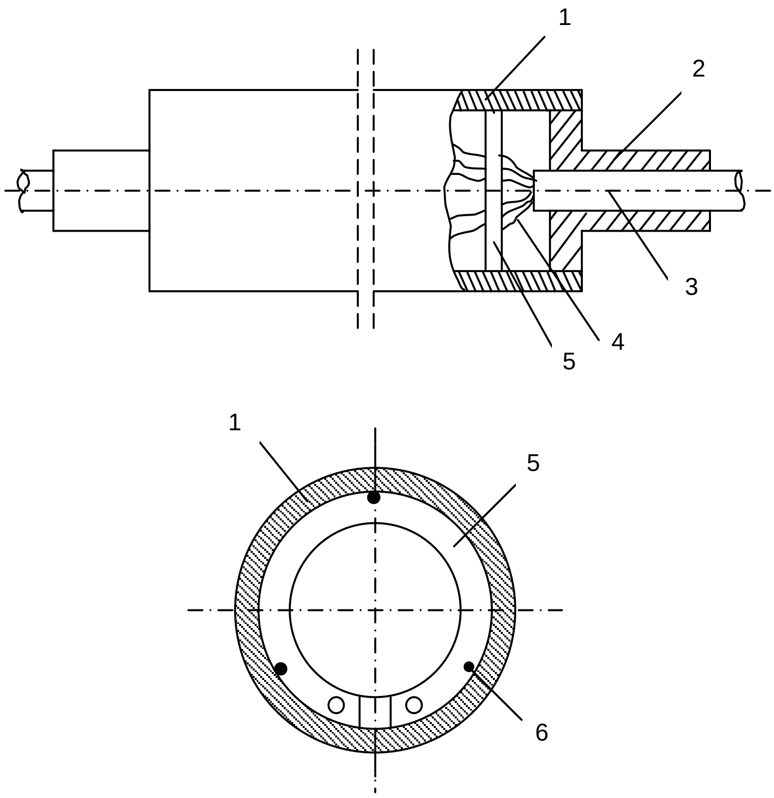

Figure 4. Calorimeter: *1* - thin-walled stainless steel tube; *2* - tip; *3* - elastic tube; *4* - thermocouples; *5* - elastic plastic ring; 6 - head of thermocouple

4. Local heat transfer for a turbulent film flow on a vertical surface

The water film flowing down a surface of vertical tube was used in experiments. The experiments were provided for Reynolds number ranged from $9{\cdot}10^3$ to $28{\cdot}10^3$. The temperature of the tube (calorimeter) surface and the film, electric current, voltage were measured and recorded during the experiment. After registration of electric current and voltage the heat flux density on the calorimeter surface was calculated. When records of heated tube surface and film flow temperatures were performed, the difference of temperature ΔT (between the mean temperatures of film $\overline{T}_f$ and tube surface T_w) was calculated. Local heat transfer coefficient was computed by formula

$$\alpha = q_w / \Delta T \tag{16}$$

Experimental data are presented in Fig. 5. As we can see, alteration of the local heat transfer coefficient in the entrance region is complicated. Three different regions may be delineated along the length of film flow in a case when initial average velocity of the film in the liquid distributor is less than an average velocity of stabilized flow. In the first region the significant decrease of the local heat transfer coefficient while reaching minimal value at some distance from liquid distributor is seen. This phenomenon one can explain by the development of thermal boundary layer and its laminar nature. In the second region, fluid fluctuations begin to develop while heat transfer increases to a maximum value. The beginning of heat transfer stabilization takes place in the third region of the film flow. Augmentation of a thermal boundary layer terminates with the film thickness. The variation of local heat transfer in the entrance region when initial average velocity of the film exceeds or is equal to an average velocity of stabilized film is not high. As we can see from Fig.5, in all cases the heat transfer stabilization takes place at the distance 0.5 m from the liquid distributor when Re > 10^4.

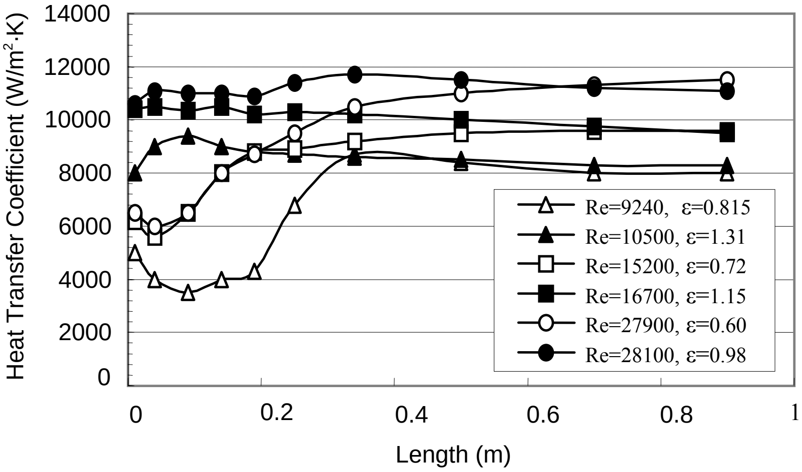

Figure 5. Variation of local heat transfer coefficient in the entrance region of the film flow down a vertical surface: $\varepsilon = \overline{w}_d / \overline{w}_{stab}$ - ratio of the film mean velocities in the liquid distributor and for stabilized flow

4. Conclusions

Analytical predictions of the heat flux distribution for the turbulent liquid film flow on a vertical plane surface are made. The decrease of the flux density in the film is observed as the distance from wetted surface increases.

The turbulent film flow is very complicated and difficult for analytical study. The determining effect of the film generation on the heat transfer intensity takes place in the entrance region of the film flow down a surface of vertical tube. The experimental data revealed that Reynolds number and the initial velocity of the film have a significant influence to the local heat transfer. It is obtained that heat transfer stabilization occurs at 0.5 m distance from the liquid distributor when $Re > 10^4$.

With large Reynolds numbers the inertia forces are sufficiently large to cause the disturbances and a transition to turbulence takes place very fast.

5. References

Alekseenko, S., Cherdantsev, S., Kharlamov, S. and Markovitch, D., 2001, "Characteristics of Liquid Film in a Vertical Pipe with the Pressure of Gas Flow", Proceedings of 5th World Conference on Experimental Heat Transfer, Fluids Mechanics and Thermodynamics, Vol.3, Thessaloniki, Greece, pp. 1996-1999.

Bertani, C. and De Salve, M., 2001, "On Freely Falling Liquid Film along Vertical Flat Plate", Proceedings of 5th World Conference on Experimental Heat Transfer, Fluids Mechanics and Thermodynamics, Vol.3, Thessaloniki, Greece, pp. 1953-1958.

Chinnov, E., Zaitsev, D., Sharina, I., Marchuk, I. and Kabov, O., 2001, "Heat Transfer and Breakdown of Subcooled Falling Liquid Film on a Middle Size Heater", Proceedings of 5th World Conference on Experimental Heat Transfer, Fluids Mechanics and Thermodynamics, Vol.3, Thessaloniki, Greece, pp. 1985-1990.

Gantchev, B.G., 1987, "Cooling of Nuclear Reactor Elements by Falling Film Flow", Energoatomizdat, Moskva, 192 p.

Gimbutis, G., 1988, "Heat Transfer of a Falling Fluid Film", Mokslas Publishers, Vilnius, 233 p.

Kays, W.M., 1972, "Convective Heat and Mass Transfer", Energy, Moskva, 446 p.

Rifert, V., Sidorenko, V., Usenko, V. and Zolotukhin, I., 2001, "Methodic of Measurements and Experimental Results of Local Heat Transfer at Liquid Film Flow on the Horizontal Finned Tubes", Proceedings of 5th World Conference on Experimental Heat Transfer, Fluids Mechanics and Thermodynamics, Vol.3, Thessaloniki, Greece, pp. 997-2000.

Tananayko, J.M. and Vorontszov, E.G., 1975, "Analytical and Experimental Methods in a Fluid Film Flow", Technika, Kiev, 312 p.

Wadekar, V.V., 2000, "Heat Transfer to Falling liquids Films with High Prandtl Numbers" Proceedings of 3rd European Thermal Sciences Conference, Vol.2, Heidelberg, Germany, pp. 851-856.

6. Responsibility notice

The authors are the only responsible for the printed material included in this paper.