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Intensification of Heat Transfer during Steam Condensation in Process Condenser of NPP Unit Cooling System

Tetiana Rymar*, Anna Malysheva

Lviv Polytechnic National University, 12 S. Bandery St., Lviv, 79013, Ukraine

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Abstract

The paper investigates heat transfer during condensation of water vapor on vertical pipes of the process condenser of the cooling system of a nuclear power unit. The numerical study was performed at different mass flow rates of steam in the range of its change from 20 kg/s to 40 kg/s. Intensification of heat exchange is provided by the use of highly efficient heat exchange profiled tubes. The study used a set of heat exchange tubes (25 mm in diameter and 1.4 mm wall thickness) with the following values of the distance between the grooves of the profiled tube: 0.007075 m, 0.00875 m, 0.00925 m, 0.0105 m. The effect of the groove depth (from 0.0007 to 0.0009 m) on heat transfer during water vapor condensation on vertical pipes was also studied. The study was carried out for the range of changes in the Reynolds number for the condensate film from 5254.2 to 10508.5. The study obtained data indicating an increase in the heat transfer coefficient on the pipe with an intensifier compared to the heat transfer coefficient on a smooth pipe. This analysis did not take into account the change in tube wall temperature.

Keywords: heat transfer; film condensation; profiled heat exchange tubes; vertical tube condenser; water vapor.

1. Introduction

The interest in the problem of intensification of heat transfer during condensation of water vapor in surface-type condensers is constantly growing [1] – [8]. Condensers are usually made as tube condensers with steam condensing outside the tubes. The thermal resistance in the case of surface condensation on the side of the condensing steam consists of the thermal resistance of the condensate film and the air gap. The thermal resistance of the film can be reduced by destroying or turbulizing the condensate film, while the thermal resistance of the interlayer can be reduced by ensuring reliable air removal, maintaining sufficient velocities of the steam-air mixture in the pipe bundle, and rational layout of the pipe bundle. The most rational methods of intensifying heat transfer from the condensing steam side are: creating droplet condensation [1], [2]; using low-finned pipes [2]; creating vibration of the heat transfer surface [2]; using the tilt of the pipe bundle [2]; developing effective air removal schemes [3], etc.

2. Analysis of publications and research

The use of profiled or finned pipes increases the heat transfer surface per unit length and intensifies the heat transfer of steam and water. Today, a large number of types of finned surfaces are known. Pipes with different rolling shapes, where fins are present on both the outer and inner sides, can be actually used in condensers (for example, the pipes shown in Fig. 1).

Paper [1] investigated the intensification of heat transfer in the condenser of a steam turbine unit of a nuclear power plant by replacing film condensation with droplet condensation. The authors found that changing the state of the outer surface of heat exchange tubes increases the heat transfer coefficient during water vapor condensation by almost three

* Corresponding author. Email address: tetiana.i.rymar@lpnu.ua

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times, which ensures an increase in heat transfer efficiency (more than 10 %) and, accordingly, makes it possible to reduce the working surface of the condenser. Paper [2] analyzes the methods for identifying heat transfer in nuclear power plant equipment, in particular, various types of fins, turbulent generators and swirlers of single- and two-phase flows, heating surface coatings, and mixed-flow systems. Calculation recommendations for determining the intensity of heat emission from promising types of heat exchange surfaces, namely bundles of low-melting tubes and spiral coils, are given.

The influence of the technical condition of steam turbine condensing devices on the amount of electricity losses, the amount of electricity generated, and the reliable and economical operation of NPP power units was studied in [3]. An empirical dependence of the electric power of a turbine unit on the presence of air leaks, inlet water temperature, and contamination of the heat exchange surface has been established. Taking into account the influence of contamination and steam load, when the inlet water temperature increases by one degree, the increase grows by a value in the range from 30 to 47 kW. Taking into account contamination and temperature changes, in the case of an increase in steam load, the increase grows by an amount ranging from 1.17 to 2.66 MW. Paper [4] proposes the use of nanofluids as a heat carrier in heat exchangers for scheduled and emergency cooling of a nuclear power unit in order to improve the safety of nuclear power plants. The intensity of heat transfer depends on the thermo-physical properties of the heat carriers, and these properties depend on the nature of the heat carrier. The addition of nanoparticles to the base coolant significantly increases its viscosity, which is accompanied by an increase in hydraulic resistance and an increase in pumping power.

The theoretical study of laminar film condensation on the outer surface of a vertical tube in the presence of flowing steam was studied in [5]. It was determined that the condensation heat transfer coefficient decreases with increasing cooling water velocity and depends on the conduction through the tube wall and convection of cooling water in the condensation area at the inlet. Work [6] investigated the condensation heat transfer of steam on vertical microtubes with diameters: 0.608 mm, 0.793 mm, 1.032 mm, and 1.221 mm. It was determined that with an increase in the temperature difference between the steam and the surface and with a decrease in the tube diameter, the condensation heat transfer coefficient monotonically decreased. A new correlation of the condensation heat transfer of steam is proposed, taking into account the influence of the tube diameter and steam velocity.

The study of the mechanical properties of high-efficiency heat exchange tubes, which are widely used in heat exchangers to enhance heat transfer and ensure reliable operation of heat exchangers, is presented in [7]. The analysis was performed for two types of high-efficiency heat exchange tubes: a spirally grooved (SG) tube and a converging-diverging (CD) tube. Paper [8] analyzes the heat transfer during the condensation of water vapor inside a vertical tube condenser. It was found that the presence of non-condensable gas reduces the temperature of steam condensation and reduces the value of the convective heat transfer coefficient.

The advantages of evaporative-condensation heat transfer devices as promising means of passive heat removal and thermal protection in nuclear power are substantiated in [9]. The author considers the main thermo-physical factors limiting the heat transfer capacity of evaporative-condensation systems and presents examples of circuit design solutions for evaporative-condensation systems for passive heat removal and thermal protection in nuclear power. Paper [10] presents results on the heat transfer of supercritical water heated above the pseudo-critical temperature or affected by mixed convection flowing up and down in vertical tubes with an internal diameter of 6.28 mm and 9.50 mm. It is shown that: the heat transfer coefficient in the downward flow of water can be higher by about 50% compared to the upward flow coefficient; the deterioration of the heat transfer regime is affected by the direction of flow, i.e., under the same operating conditions, the deterioration of heat transfer can be delayed in the downward flow compared to the upward flow.

3. Goal of the paper

The existing data on heat transfer enhancement during water vapor condensation is incomplete, and further research on other parameters is needed to apply profiled or finned tubes to increase the heat transfer surface. The operating conditions of tubes in a bundle in real condensers differ significantly from those of a single tube. At the same time, most studies of water vapor condensation intensification have been performed for single tubes. There are practically no data on the effect of air velocity in bundles of rolled or finned tubes on heat transfer. The deterioration of heat transfer due to flooding in bundles of finned pipes is 2 to 2.5 times less than in a bundle of smooth pipes [11]. One of the effective ways to reduce the impact of flooding for profiled pipes can be to tilt the pipe bundle horizontally by a small angle. In general, the efficiency of heat exchangers with profiled tubes increases by 25-50 %, although there are currently not enough practical recommendations in terms of quantifying this issue. Therefore, the aim of this work is to numerically study heat transfer during condensation of water vapor on vertical pipes of the process condenser of the cooling system of a nuclear power unit at different mass flow rates of steam. Methodological approaches to solving the problems of calculating the justification for the intensification of heat transfer on vertical tubes through the use of highly efficient heat exchange profiled tubes are used in this work.

4. Presentation and discussion of the research results

The basic technological scheme of cooling the first circuit of a NPP power unit with a process condenser is shown in Fig. 2.

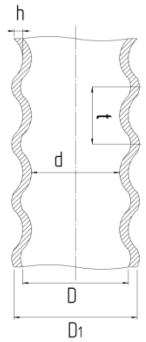

Fig. 1. Longitudinal section of the profiled pipe used to condense the coolant on the outer surface: *h* - groove depth of the profiled pipe, m; *t* - distance between grooves of profiled pipe, m; *d* - diameter of profiled pipe, m; *D* - internal diameter, m; D_1 - external diameter, m.

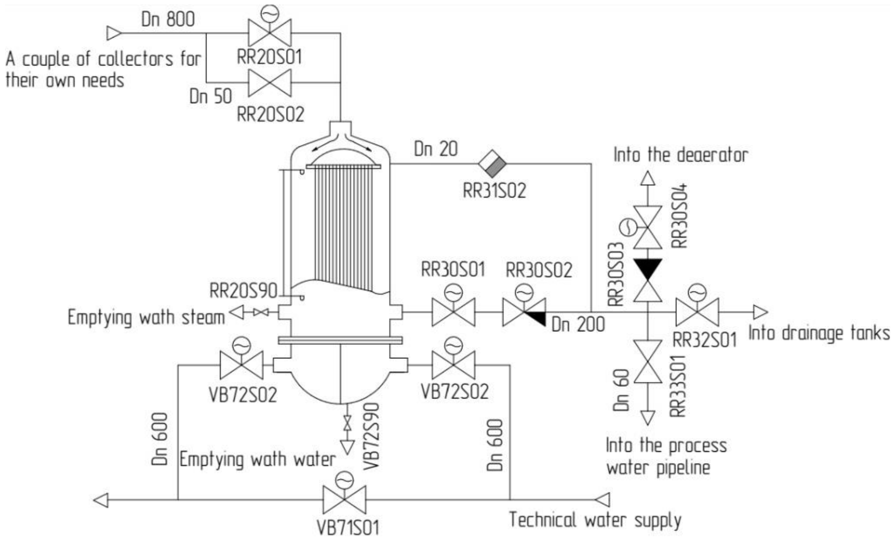

Fig. 2. Scheme of the system for cooling the first circuit of a NPP power unit through a process condenser.

The power unit cooling system is designed to ensure non-stationary power unit operation by regulating steam pressure in the second building, by diverting part of the steam to the process condenser in case of shutdown or maintaining the power unit in a «hot» state. The cooling system ensures that the first circuit is cooled down from nominal parameters to a temperature of 150 °C at a rate of 15 °C/h in the event of a power unit shutdown.

The process condenser is a single-shell, vertical, shell-and-tube heat exchanger with a floating head, single-pass in the intertube space and double-pass in the tube space. The condenser condenses steam and then cools the condensate as it passes through the intertube space along the vertical tubes. The heat exchange surface is made of pipes ø 25x1.4 mm, which are placed with a spacing of 32 mm.

In order to justify the choice of optimal parameters of the profiled pipes, theoretical studies were performed in a wide range of changes in operating and geometric parameters. The study of heat transfer intensification on vertical pipes of a process condenser was carried out during the condensation of water vapor. Cooling water was supplied inside the tubes, and water vapor was supplied to the intertube space from above.

The authors of [11] propose the following dependence for practical calculations of the effects of heat transfer intensification during water vapor condensation when using smooth pipes

$$\overline{Nu_0} = 0.925 \cdot Re_{film}^{1/3} \cdot \left[1 + 0.04 \cdot Re_{film}^{0.2} + 2.23 \cdot 10^{-3} \cdot Re_{film}^{0.8} \cdot Pr_{film}^{0.6}\right] \cdot \left[\left(\frac{\lambda_w}{\lambda_f}\right)^3 \cdot \left(\frac{\mu_f}{\mu_w}\right)\right]^{1/8}, \quad (1)$$

where $Re_{film} = \frac{G_s}{\mu_f}$ is the Reynolds number for the film; Pr_{film} is the Prandtl number for the film; G_s is the specific condensate flow rate, kg/(s·m); λ_f is the condensate thermal conductivity at its average temperature, W/(m·K); λ_w is the condensate thermal conductivity coefficient at the wall temperature, W/(m·K); μ_f is the dynamic condensate viscosity at its average temperature, Pa·s; μ_w is the dynamic condensate viscosity at the wall temperature, Pa·s.

The results of studies on the average heat transfer during condensation of water vapor on a profiled pipe are summarized by the formula [11]

$$\overline{Nu} = \overline{Nu}_0 \cdot \left[1 + \frac{5.4 \cdot 10^3}{exp\left(1.4 \cdot \frac{t}{h}\right)}\right] \cdot Re_{film}^{\overline{\frac{0.127}{exp\left[\frac{9}{h} \cdot \left(\frac{\nu_f^2}{g}\right)^{1/3}\right]}}}, \quad (2)$$

where ν_f is the kinematic viscosity of condensate at its average temperature, m^2/s; g is the acceleration of free fall, m/s^2; h is the depth of the groove of the profiled pipe, m; t is the distance between the grooves of the profiled pipe, m.

To compare the thermal efficiency of structurally different intensifiers based on experiments conducted by different authors at different average flow temperatures of the medium and ranges of Reynolds and Prandtl numbers, it is recommended to use the following relationship

$$\frac{\overline{Nu}}{\overline{Nu}_0} = f(Re_{film}), \quad (3)$$

where the index “0“ characterizes a smooth heat transfer surface. Dependence (3) characterizes the increase in the heat transfer coefficient on a pipe with an intensifier compared to the heat transfer coefficient on a smooth pipe.

Figures 3 – 5 show the results of numerical calculation of heat transfer intensification during condensation of water vapor on vertical pipes ø25 x 1.4 mm of a process condenser with an intensifier (in the range of changes in the value of the distance between the grooves of the profiled pipe t = 0.007075 ÷ 0.0105 m and in the range of changes in the depth of the groove of the profiled pipe h = 0.0007 ÷ 0.0009 m) in comparison with the heat transfer coefficient on a smooth pipe.

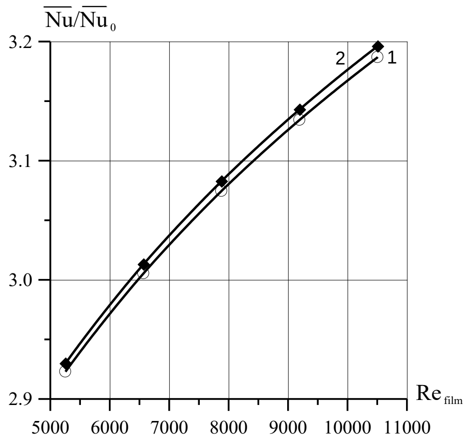

Fig. 3. Intensification of heat transfer during condensation of water vapor on vertical pipes ø25 x 1.4 mm of a technological condenser in the range of changes in the value of the distance between the grooves of a profiled pipe t from 0.007075 to 0.0105 m at the depth of the groove of a profiled pipe (h = 0.0007 m): 1 - t/h = 10.11; 2 - t/h = 15.

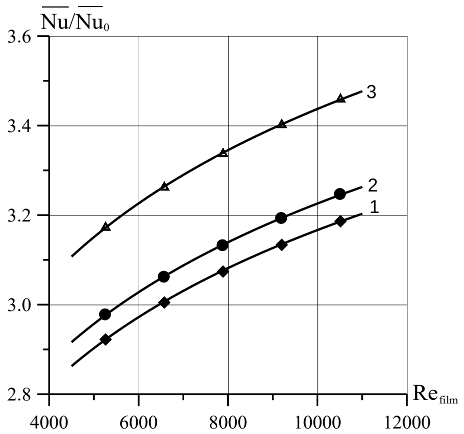

Fig. 4. The effect of changing the depth of the groove of a profiled pipe (in the range of change h = 0.0007 ÷ 0.0009 m) on heat transfer during the condensation of water vapor on vertical pipes ø25 x 1.4 mm of a process condenser at a constant value of the distance between the grooves of the profiled pipe (t = 0.007075 m): 1 - t/h = 10.11; 2 - t/h = 8.84; 3 - t/h = 7.86.

From the analysis of Figures 3 – 5, we see that the Reynolds number for the film has a significant effect on the intensification of heat transfer. From the analysis of Fig. 3, we see that with the increase of the Re_{film} Reynolds number from 5254.26 to 10508.51 (twice), the heat transfer coefficient of the profiled pipe increases 2.923 – 3.19-fold compared to the smooth heat exchange surface. The effect of the distance between the grooves of the profiled pipe t from 0.007075 to 0.0105 m at the depth of the groove of the profiled pipe h = 0.0007 m does not significantly affect the intensification of heat transfer, in particular, changing the pitch of the turbulators from t/h = 10.11 to t/h = 15 (1.48-fold increase) leads to an increase in Nu/Nu_0 by 0.24 ÷ 0.29 %, depending on the Reynolds number Re_{film}. The analysis of Figures 3 – 5 showed that the highest heat transfer intensity is observed for profiled tubes of the process condenser with the depth of the profiled tube groove h = 0.0009 m and the distance between the grooves of the profiled tube t = 0.007075 m (with the turbulizer pitch t/h = 7.86), for which the intensification of heat transfer of the profiled tube compared to the smooth tube is 3.173 ÷ 3.46 times. The lowest heat transfer efficiency is achieved by profiled tubes of a process condenser with a depth of the profiled tube groove h = 0.0007 m and a distance between the grooves of the profiled tube t = 0.007075 m (with a turbulizer pitch t/h = 10.11), for which the intensification of heat transfer of the profiled tube compared to a smooth tube is 2.923 ÷ 3.187 times.

The analysis of Figures 3 – 5 showed that the turbulizer pitch t/h does not unambiguously affect the intensification of heat exchange of water vapor condensation in the process condenser. An increase in the turbulizer pitch t/h (Fig. 3) due to an increase in the distance between the grooves of the profiled pipe t from 0.007075 to 0.0105 m (1.48 times) leads to an intensification of heat transfer (an increase of 0.29 %). Increasing the turbulizer pitch t/h by increasing the depth of the groove of the profiled pipe in the range of changes in h from 0.0007 to 0.0009 m leads to a decrease in heat transfer during the condensation of water vapor on the vertical pipes of the process condenser (Fig. 4, Fig. 5). However, the results of numerical modeling showed that for the value of the distance between the grooves of the profiled pipe t = 0.0105 m, a 1.29-fold increase in the turbulizer pitch t/h does not significantly affect the intensification of heat transfer (an increase of 0.04 %). And at a value of the distance between the grooves of the profiled pipe t = 0.007075 m, a 1.29-fold increase in the turbulizer pitch t/h significantly affects the intensification of heat transfer (an increase of 8.55 %).

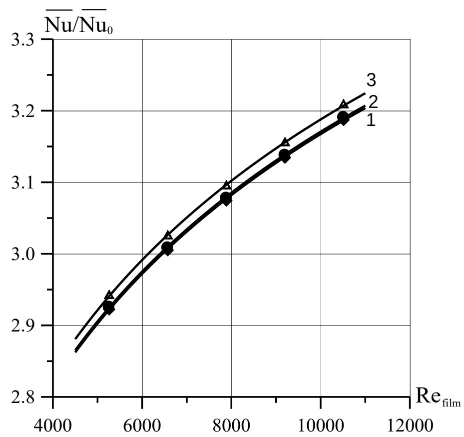

Fig. 5. The effect of changing the depth of the groove of a profiled pipe (in the range of change h = 0.0007 ÷ 0.0009 m) on heat transfer during the condensation of water vapor on vertical pipes ø25 x 1.4 mm of a process condenser at a constant value of the distance between the grooves of the profiled pipe (t = 0.0105 m): 1 - t/h = 15; 2 - t/h = 13.13; 3 - t/h = 11.67.

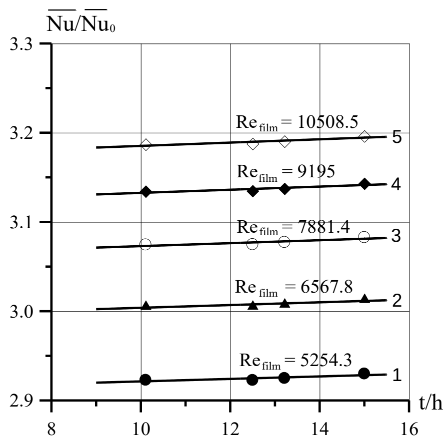

Fig. 6. Effect of the turbulizer pitch t/h on the intensification of heat transfer during the condensation of water vapor on vertical pipes ø25 x 1.4 mm of the process condenser at the depth of the groove of the profiled pipe h = 0.0007 m: 1 - Re_{film} = 5254.3; 2 - Re_{film} = 6567.8; 3 - Re_{film} = 7881.4; 4 - Re_{film} = 9195; 5 - Re_{film} = 10508.5.

Fig. 6 shows the effect of the turbulizer pitch t/h on the intensification of heat transfer during the condensation of water vapor on vertical pipes ø25 x 1.4 mm of the process condenser compared to a smooth pipe at a groove depth of the profiled pipe h = 0.0007 m. The analysis of Fig. 6 showed that an increase in the Reynolds number of the condensate film (an increase in the specific condensate flow rate) has a significant impact on the intensification of heat transfer. With an increase in the Reynolds number of Re_{film} from 5254.26 to 10508.51 (twofold), the heat transfer coefficient of the profiled pipe increases 2.923-3.19-fold compared to a smooth heat exchange surface. A 1.48-fold increase in the turbulizer pitch t/h leads to an increase in heat transfer by only 0.24%.

5. Conclusion

The use of profiled tubes increases the heat transfer surface per unit length and also intensifies the heat transfer of water vapor condensing on the outer surface of the vertical tubes of the process condenser of the cooling system of a nuclear power unit. The use of profiled tubes with annular grooves makes it possible to intensify heat transfer during condensation up to 3.46 times. The analysis of the numerical calculation of heat transfer of water vapor condensation on vertical pipes of the process condenser showed that the highest heat transfer intensity is for pipes with a groove depth of the profiled pipe h = 0.0009 m and a distance between the grooves of the profiled pipe t = 0.007075 m, for which the intensification of heat transfer of the profiled pipe compared to the smooth pipe (Nu/Nu_0) is 3.173 ÷ 3.46 times. A significant effect on the intensification of heat transfer is exerted by an increase in the specific condensate flow rate (Reynolds number for the film). It was found that with a twofold increase in the Reynolds number of Re_{film}, the heat transfer coefficient of the profiled pipe increases 2.923 ÷ 3.19-fold compared to a smooth heat exchange surface. The increase in heat transfer during steam condensation is due to the action of surface tension forces in areas of low curvature. No significant intensification is observed in areas of greater curvature.

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Інтенсифікація теплообміну під час конденсації пари у технологічному конденсаторі системи розхолоджування енергоблоку АЕС

Тетяна Римар, Анна Малишева

Національний університет «Львівська політехніка», вул. С. Бандери, 12, Львів, 79013, Україна

Анотація

У роботі досліджено теплообмін під час конденсації водяної пари на вертикальних трубах технологічного конденсатора системи розхолоджування енергоблоку АЕС. Чисельне дослідження виконано за різних масових витратах пари в діапазоні їх зміни від 20 кг/с до 40 кг/с. Інтенсифікація теплообміну передбачена за рахунок застосування високоефективних теплообмінних профільованих трубок. Для дослідження використано комплект теплообмінних трубок діаметром 25 мм і товщиною стінки 1,4 мм з різними значеннями відстані між канавками профільованої труби: 0,007075 м; 0,00875 м; 0,00925 м; 0,0105 м. Також досліджувався вплив глибини канавки профільованої труби (у діапазоні від 0,0007 до 0,0009 м) на теплообмін під час конденсації водяної пари на вертикальних трубах. Дослідження виконано для діапазону зміни числа Рейнольдса для конденсатної плівки від 5254,2 до 10508,5. У роботі отримано дані, що вказують на зростання коефіцієнта тепловіддачі на трубі з інтенсифікатором порівняно з коефіцієнтом тепловіддачі на гладкій трубі. У цьому аналізі не враховувалася зміна температури стінки трубки.

Ключові слова: тепловіддача; плівкова конденсація; вертикальний трубчастий конденсатор; профільовані теплообмінні трубки; водяна пара.