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Influence of Tube Diameter and Steam Flow Rate on Heat Transfer in a Vertical Pipe of Condenser: Experimental Investigation of Copper Pipes

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The overall condensing power at condensers is affected by many factors. A condensate film on the pipe wall plays a crucial role in heat transfer. The velocity of the gas phase inside the tubes has a fundamental influence on the movement of the liquid film and the specific course of the velocity profile in the condensate film. The magnitude of the shear stress at the steam-condensate interface affects the film thickness and its integrity. This paper presents a study to evaluate the effect of the flow velocity inside a vertical pipe on the heat transfer coefficient during water vapour condensation. Specifically, steam flow on heat transfer for two different pipes is evaluated, namely with inner diameters 16 and 26 mm. A common feature is a detailed investigation of the steam condensation process for a parallel flow and counterflow of steam and liquid film. Furthermore, the influence of the temperature and flow direction of the water cooling the outer side of the condenser tube on the transmitted power is evaluated. The condensation process is experimentally investigated on a copper pipe-in-pipe heat exchanger with a possible change of the direction of the cooling water flow. Determination of the condensation heat transfer coefficient is based on experimental identification of the overall heat transfer coefficient and subsequent inverse calculation of the condensation heat transfer coefficient. The condensation heat transfer coefficient ranges from 3,000 to 6,500 W/(m^2·K) for all configurations measured. The results generally show that as the Reynolds number of steam flow increases, the condensation heat transfer coefficient increases too. At Reynolds number of 35,000 the same heat transfer coefficient value is identified either for parallel flow or counterflow of cooling water. For higher Reynolds numbers, the parallel flow of cooling water enables to reach the higher heat transfer coefficient compared to counterflow configuration. At lower Reynolds numbers, the dependency is reversed.

1. Introduction

Condensation is a process which is an important part of many technological devices, for example, in the thermal circuits of steam power plants, in flue gas condensers or humid air condensers. The condensation process takes place in heat exchangers called condensers. One of the most widely used types of heat exchangers is a pipe heat exchanger. To produce a thermal design of a pipe exchanger with phase change requires good knowledge of the phase change process in pipes. This paper focuses on an experimental analysis of film condensation of water vapour in a vertical pipe depending on the Reynolds number.

The first derivation of the analytical relation for laminar film condensation was made by Nusselt in 1916. Nusselt derived the relation for the heat transfer coefficient during condensation on a vertical wall with the vertical steam velocity being zero. Many other scientists then used his relations as a basis. Rohsenow focused on non-linear temperature field distribution in the condensate film. Sparrow and Gregg (1959) included in Nusselt's theory a change in the momentum of a liquid film in a gravitational field caused by the acceleration of the film flow. A considerable change in the consideration of liquid film is respecting the fact that the surface of a liquid film is not perfectly smooth and due to gravity and the friction of the gas and liquid molecules inside the film, it becomes deformed and its surface becomes wavy. This problem was mathematically commented on by Aktershev and Alekseenko (2005), who describes the effect of flowing gas on the film of a liquid for parallel flow and counterflow. In articles by Aktershev and Alekseenko (2013) and Aktershev and Alekseenko (2017), used the finite element method to investigate the formation of natural waves and expresses the conditions for the stability of the condensate film flow. The undulation of the surface significantly increases the heat transfer.

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According to Whitham, the increase reaches up to 20.6 % in comparison with the original Nusselt's theory. Blangetti and Naushahi (1980) included in the calculations the suction effect, which occurs during mass transfer due to condensation. Due to the different steam pressures in the core of the steam flow and at the steam-condensate interface, a radial component of velocity occurs increasing the amplitude of the film undulation. The suction effect is expressed by a so-called blowing parameter, which primarily depends on the steam velocity and the condensation intensity. In the recent years, new numerical models of water vapour condensation have been developed. Lee and Son (2018) studied the influence of the flow on the condensation process during parallel flow and counterflow of steam and condensate. They compared the analytical approach proposed by Nusselt and the numerical approach proposed in . Their work primarily shows the necessity of a correct setting of the numerical simulations, where, for example, even the size of the numerical network is very important. The numerical approach was also used by Hssain and Hammami (2019), who developed a model for predicting water vapour condensation in relation to the relative humidity and the input Reynolds number of steam. The simulation showed that with the Reynolds number and the relative steam humidity increasing, the condensation intensity grows. A common feature of the above-mentioned studies is a detailed investigation into the steam condensation process for a parallel flow configuration of the condenser pipe cooling. None of the studies focused on assessing the impact of a change in the cooling configuration (parallel flow/counterflow) on the resulting heat transfer coefficient. This study focuses on assessing the influence of a change in the character of the cooling (parallel flow/counterflow), in the Reynolds number, and in the vertical pipe diameter on the condensation heat transfer coefficient at atmospheric pressure. The results obtained allow the formulation of suggestions leading to minimizing the material demands of steam condensers.

2. Experiment a data treatment

2.1 Experimental device

Figure 1 shows the testing apparatus. It is based on a pipe-in-pipe heat exchanger, where the condensation process is monitored. The steam runs through the inner pipe and the cooling water flows in the inter-annular space of the shell of the exchanger. Both the concentric pipes are made of copper. It is possible to change the direction of the cooling water flow in the experimental device and thus to change the cooling flow configuration of the condenser pipe from parallel flow to counterflow. The experimental section, where the controlled condensation takes place, is 1 m long. The source of the water steam is a steam generator with a maximum amount of the generated steam of 0.0097 kg/s. The output of the steam generator is controllable with a step of $2.7 \cdot 10^{-5}$ kg/s of steam.

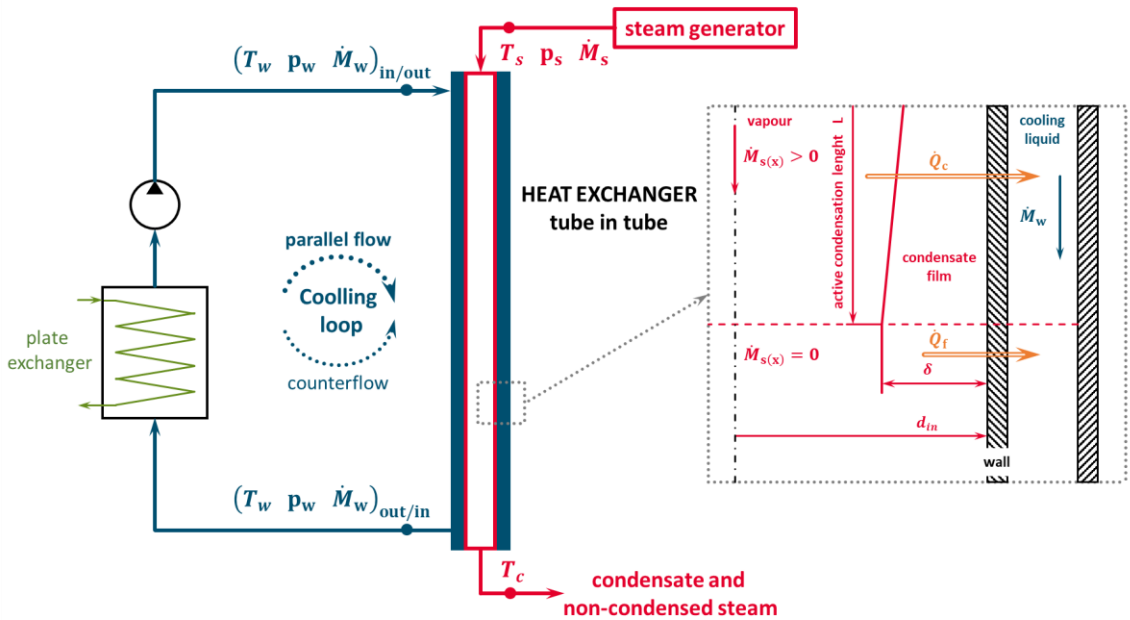

Figure 1: Experimental loop scheme

At the steam inlet into the exchanger, the temperature (T_s) and pressure (p_s) of the incoming steam are measured and the temperature of the outgoing condensate (T_c) is measured at the exchanger outlet. The cooling loop maintains the required temperature of the cooling water at the inlet into the shell of the exchanger ($T_{w,in}$) and at the outlet ($T_{w,out}$) with the mass flow (M_w). Further, at the side of the cooling loop, water pressures are measured at the inlet ($p_{w,in}$) and outlet ($p_{w,out}$). Heat is removed from the cooling loop using a plate exchanger into the central cooling water available in the laboratory.

2.2 Determining the condensation heat transfer coefficient

To correctly determine the condensation heat transfer coefficient, it is necessary to know the overall heat transfer coefficient. It depends both on the intensity of cooling from outside and on the material and thickness of the pipe wall. The overall heat transfer coefficient is given by Eq(1).

$$k_c = \frac{\dot{Q}_c}{L \cdot \Delta T_{ln}}, \tag{1}$$

where $\dot{Q}_c$ is the heat output of a phase change. Two situations occurred during the experiments. In the first case, such an amount of steam was let into the pipe that, even after running through the exchanger, it could not condense completely. In such a case, the active length L equals the entire experimental length of the exchanger, and the output $\dot{Q}_c$ equals the output transmitted to the cooling water. In the other case, such an amount of steam was used that all the steam condensed in the exchanger and the condensate started to subcool. In this case, the length L equals only the condensation length shortened by the length needed for the subcooling of the condensate. This length is different for each of the measured states and it depends on the outlet temperature of the condensate. The heat transfer in the condensate during subcooling is based on the available analytical relationships for the heat transfer in the liquid film. The output $\dot{Q}_c$ is only the output of the phase change. After identifying the overall heat transfer coefficient, it is possible to determine the condensation heat transfer coefficient using Eq(2).

$$\alpha_{in} = \frac{1}{d_{in} \cdot \pi \cdot \left[\frac{1}{k_c} - \frac{1}{d_{out} \cdot \pi \cdot \alpha_{out}} - \frac{ln\left(\frac{d_{out}}{d_{in}}\right)}{2 \cdot \pi \cdot \lambda_{cu}}\right]} \tag{2}$$

Where α_{out} [W/(m^2·K)] is heat transfer coefficient which is calculated according to the relations for forced convection in the annulus from the VDI heat atlas (2010)

2.3 Predicted condensation heat transfer coefficient

The condensation heat transfer coefficient can be determined using analytical equations derived from physical phenomena or obtained from experiments. This text presents two approaches. The first one is based on the Eq(3) published by Nusselt, who derived the condensation heat transfer coefficient on a vertical wall for immovable steam. This equation is still used in practice.

$$\alpha_{in,nu} = 0{,}943 \left[\frac{g \cdot \rho_c \cdot h_c \cdot \lambda_c^3}{\nu_c \cdot (t_s - t_w) \cdot L}\right]^{0{,}25} \tag{3}$$

Where h_c [J/kg] is latent heat. As can be seen, Nusselt's equation does not include the influence of the steam velocity and thus the possible effects of the moving steam on the condensate film. From more recent analytical relations, Eq(4) by Pashkevich and Muratov (2015) is used. He proposed the relation for the mean value of the condensation heat transfer coefficient depending on the mass flow rate of steam at the inlet. He validated this equation using a vertical pipe with a diameter of 20 mm.

$$\alpha_{in,pa} = \frac{\dot{M}_v \cdot h_c}{\pi \cdot d_{in} \cdot (t_s - t_w) \cdot L} \tag{4}$$

3. Results

In this study, the impact of the Reynolds number at the inlet into the condensation section on the condensation heat transfer coefficient was measured in vertical pipes of two different inner diameters. The first measurement included a vertical exchanger with the pipe inner diameter of 16 mm. The other case included an exchanger with the pipe inner diameter of 26 mm. The mass flow rates of the steam were set for the given diameters during the measurements, as is shown in Table 1. The determination of the mass flow rates of steam was based on the pipe with a diameter of 26 mm and then the flow rates were recalculated for the pipe with a smaller diameter maintaining the same input steam velocity for both pipes. Each of the studied mass flow rates of steam was assessed for three different temperatures of the incoming cooling water, particularly, 30, 40, and 50 °C. All the combinations of the parameters tested were evaluated for both the parallel flow and the counterflow of the cooling water. The fluid properties were calculated for middle temperature. The first configuration was parallel flow cooling, when the cooling water flows in the same direction as the steam and the condensate. The other configuration is counterflow cooling, when the cooling water flows in the opposite direction to the steam and the condensate. The mass flow rate of the cooling water was adjusted so that the intensity of cooling was equal for both heat exchangers.

Table1: The measured input steam flow rate and velocity

	Pipe 26 mm			Pipe 16 mm		
Case	$\dot{M}_s$ [kg/s]	u_s [m/s]	Re [−]	$\dot{M}_s$ [kg/s]	u_s [m/s]	Re [−]
1	0.0055	17.7	22,348			
2	0.0069	22.3	28,158	0.0026	22.3	21,658
3	0.0083	26.6	33,585	0.0031	26.6	25,834
4	0.0097	31.3	39,519	0.0037	31.3	30,399
5				0.0047	40.0	38,849
6				0.0059	50.0	48,561

The result of the measurement is a set of identified heat transfer coefficients. Using a regression analysis, they were used to establish a continuous dependence of the heat transfer coefficient on the monitored parameters. See the diagram in Figure 2.

3.1 Impact of the Reynolds number and the cooling water temperature on heat transfer

Figure 2 shows the influence of the Reynolds number at the inlet into the condensation section on the condensation heat transfer coefficient depending on the inner pipe diameter and the cooling water temperature. The states for the Reynolds number ranging from 22,500 to 48,000 were measured experimentally. For all the states measured, the condensation heat transfer coefficient ranged from 3,000 to 6,500 W/(m·K), which matches the values presented by other authors. The Figure clearly shows that as the Reynolds number increases, the condensation heat transfer coefficient increases as well. This is true for both the diameters and for different cooling water temperatures.

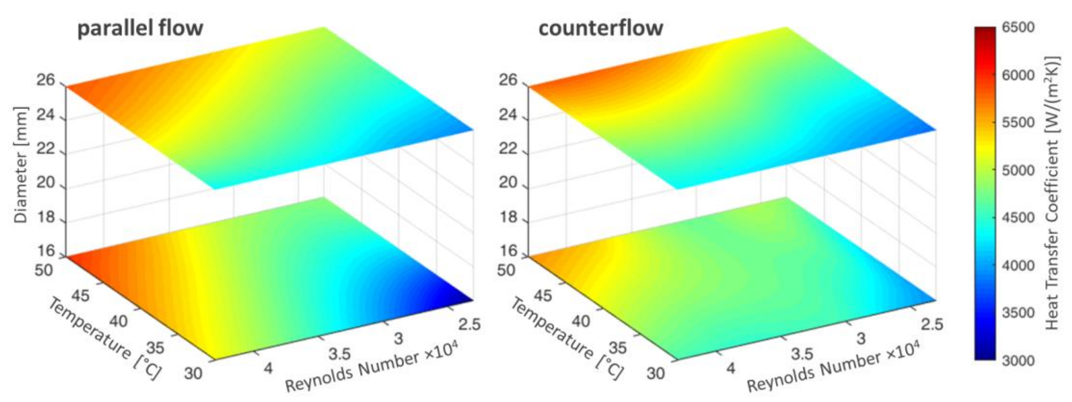

Figure 2: Influence of the Reynolds number on the condensation heat transfer coefficient depending on the inner pipe diameter and the cooling water temperature

The condensation heat transfer coefficient inside the pipe also increases with a growing temperature of the cooling water at the inlet. This results from the fact that when the heat drop is small, less steam condenses and thus less condensate, which works as additional resistance to heat conduction, is formed. With the resistance decreasing, the heat transfer coefficient inside the pipe increases but the total output transmitted in the pipe decreases with the cooling water temperature increasing.

3.2 Impact of the pipe diameter on the heat transfer

The Figure 2 also shows that the heat transfer coefficient in a pipe with a smaller diameter is more sensitive to a change in the Reynolds number or in the steam velocity than in a pipe with a larger diameter. For the inner diameter of 16 mm, there was a 42 % increase in the condensation heat transfer coefficient when the Reynolds number increased by 50 %, whereas for the inner diameter of 26 mm, there was a 10.7 % increase in the condensation heat transfer coefficient when the Reynolds number increased by 50 %. This is probably caused by two effects. Firstly, in a pipe with a smaller diameter, the condensate film is more disturbed because, in a smaller space, the steam vortices have less room to swirl and, thus, the condensate film is more disturbed and becomes wavier, which reduces the resistance to heat conduction. Secondly, due to a greater heat transfer, the condensation is more intense and, thus, the thickness of the condensate film grows more than in a pipe with a larger diameter. A greater film thickness results in a smaller cross section of the pipe, thus contributing to an increase in the steam velocity, which leads to greater turbulence inside the pipe. Therefore, if we are not bound by pressure losses during the flow in the pipe, it is more suitable to use a smaller pipe diameter to achieve a more efficient heat transfer.

A pipe with a smaller diameter is also more sensitive to the cooling configuration. In a pipe with a diameter of 26 mm, there is not much difference between the parallel flow and counterflow cooling, whereas in a pipe with a diameter of 16 mm, there are obvious differences between the cooling configurations. Figure 3 shows a detailed comparison of the impact of the parallel flow and counterflow cooling configurations for a pipe with the inner diameter of 16 mm and the input cooling water temperature of 30 °C. The experimentally obtained points were fitted with a linear curve.

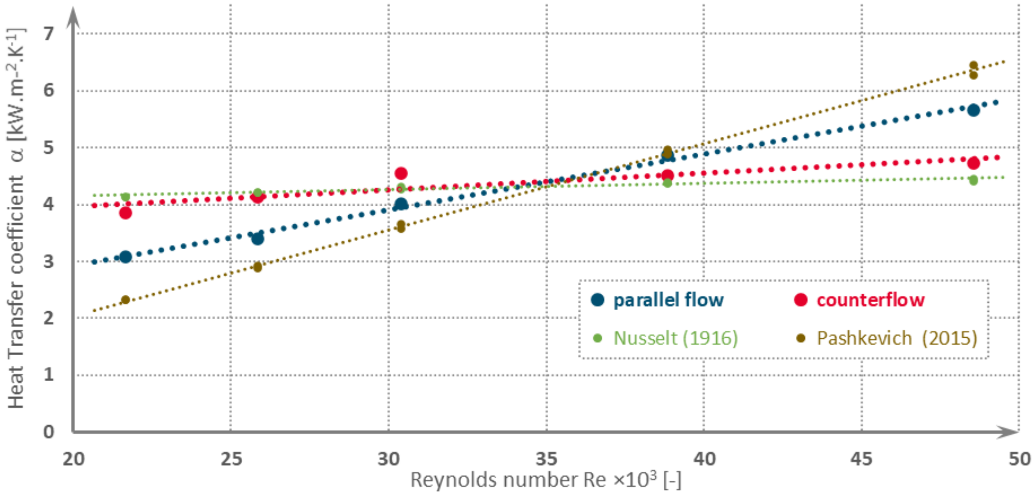

Figure 3: Dependence of the heat transfer coefficient on the Reynolds number - parallel flow and counterflow cooling configurations for the pipe diameter of 16 mm and the cooling water temperature of 30 °C.

In classic heat exchangers without phase change, counterflow cooling is usually more efficient due to a greater heat drop. For condensation heat exchangers, this statement cannot be unequivocally confirmed. The main difference, in comparison to exchangers without phase change, is the presence of the condensate film inside the pipe, which increases the resistance to heat conduction. In the counterflow cooling configuration, cold water enters the exchanger where the condensate film layer is the thickest, i.e., at the point of the highest heat resistance to heat conduction, which reduces the advantage of a greater temperature drop. For this reason, the counterflow cooling configuration is better for low values of the Reynolds number or of the mass flow rate of steam at the inlet into the pipe, see Figure 3. This is because, during condensation in the range of these steam flow rates, a sufficient layer of condensate is not formed and there is still the advantage of a greater heat drop in comparison to the resistance to heat conduction. As the mass flow rate of steam increases, the efficiency of both configurations becomes more and more equal. In our case, the point of equal efficiency corresponds to the Reynolds number of 35 000. At this point, it cannot be determined which cooling configuration is better. When the mass flow rate of steam is further increased, the liquid film thickness also increases and better efficiency of cooling is achieved by parallel flow cooling.

Figure 3 shows the theoretical behaviour. Conditions for both the parallel flow and counterflow cooling were calculated. Since neither of the authors focused on the cooling configuration, there is not a big difference between these two cases and the points almost coincide. The only difference, when these relations are applied to both cooling configurations, results from differences in the wall temperature and the mean temperature for determining the physical properties of the liquid film. However, these differences are insignificant.

When comparing the experimentally obtained data with the behaviours described by Nusselt and Pashkevich, we can see a relatively close correspondence. Nusselt's relation corresponds to the experimental data for the counterflow cooling, where the greatest deviation of the data measured from Nusselt's equation does not exceed 10 %. Therefore, it is suitable to apply his equation when calculating film condensation for counterflow cooling. Pashkevich's relation has a much greater progression, and its behaviour corresponds more to the parallel flow cooling configuration. When the Reynolds number is 35 000, the experimental data intersect Pashkevich's relation. However, since the graphs have different gradients, the deviation increases in both directions from this point. The deviation from the data obtained is the greatest when the Reynolds number is low, around 22 300, when it reaches approximately 31 %.

4. Conclusion

This paper evaluates the influence of the Reynolds number and the pipe diameter on the condensation heat transfer coefficient in a vertical pipe-in-pipe heat exchanger. Experiments were performed for two diameters (16 mm and 26 mm) of the inner pipe. It was proven that as the Reynolds number increases and the pipe diameter decreases, the condensation heat transfer coefficient increases. A bigger proportional increase in the inner heat transfer coefficient occurred with a smaller pipe diameter. The data presented above suggest that the most efficient configuration for dissipating the most power from the steam core is using a condensation pipe with a smaller diameter when the Reynolds number of the steam at the inlet into the condensation section is high. The experimental comparison of the parallel flow and the counterflow cooling configurations with a maximum heat output transmission suggests that using the parallel flow cooling of water with lower input temperature is more appropriate. This configuration provides a solution for transferring the maximum heat flow while minimizing the material demands on the condensation exchanger.

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