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Comparison of Different Concepts of Condensation Heat Exchangers With Vertically Oriented Pipes for Effective Heat and Water Regeneration

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This paper presents a theoretical study focused on the possible enhancement of the heat transfer in a vertical tube during condensation of steam. Two different concepts of condenser operation are compared. The first concept considers a condenser pipe with steam inside and coolant flowing along the external pipe wall. The second concept considers steam inside the pipe and the coolant flowing along the inner wall together with the steam. During the condensation process, a liquid film is formed on the pipe wall. This fluid film represents a thermal resistance that increases with the increasing thickness of the film. The operating parameters of both considered concepts are compared with the utilising of a developed analytical procedure. The analytical model is focused on the detail evaluation of the liquid film parameters on the heat flux, intensity of the condensation process and output temperature of coolant leaving the condenser.

1. Introduction

Condensation is a process which is an important part of many technological devices, for example, steam power plant thermal cycles, waste gas condensers or natural gas condensation (Bao et al., 2018).
The condensation process is significantly affected by the intensity of heat transfer through the condensate film and the heat exchange surface wall. This paper focuses on a detailed assessment of the impact of liquid film parameters on the condensation process. Pure steam without the presence of non-condensable gases is considered for this study as the condensing component. In many applications, condensate film creates the most significant thermal resistance against heat transfer in the condensation exchanger. On the steam-condensate interface, the steam jet creates interfacial shear stress which deforms the speed profile and affects the character and thickness of the condensate film. In the case of a laminar film flow, the resistance against heat conduction increases in direct proportion to its thickness and in the case of a turbulent film flow, thermal resistance is lower (Al-Shammari, 2004).
The theory of membrane condensation on a vertical wall is based on historical relationships deduced by W. Nusselt (1916). He deduced the speed and thickness of condensate film as the gravitational acceleration function while the impact of shear stress and momentum change in the film was neglected. Due to different steam pressures in the core of the steam jet and on the interface, a radial speed component is formed, which increases the amplitude of the wave motion of the film. Chen and Ke (1993) designed a mathematical model for heat transfer through the condensate film during turbulent steam flow. Oh and Revankar (2005) designed a modification of a laminar film model including the effect of mass transfer on interfacial shear stress. Kubín and Hirš (2016) compared a classical thermal resistance method with the Wilson plot method. On the field of CFD simulation, the condensation of steam was solved by Phan and Won (2018) whose compared the influence of a mesh size of the 3D model on results of thermal transfer.
Most vertical heat exchangers in which steam condensation occurs use coolant flowing from the outside of the condenser pipe wall for heat rejection. For certain types of application, this way of cooling is not suitable. If the condensing mixture contains substances which might disturb the pipe material, it is suitable to distribute

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the coolant directly into the inner space of the condenser pipe. As the coolant runs along the inner pipe wall, it protects the pipe from the impact of the condensing gas, abrasion by solid substances carried by the flowing gas, and erosion of water drops.

When coolant is used inside the pipe, it is not necessary in certain cases to use steel, copper, or aluminium as the construction material. Cheaper materials may be used such as PPR plastic. A disadvantage of this configuration in some applications is the contact of the coolant with the condensing medium, which leads to its pollution.

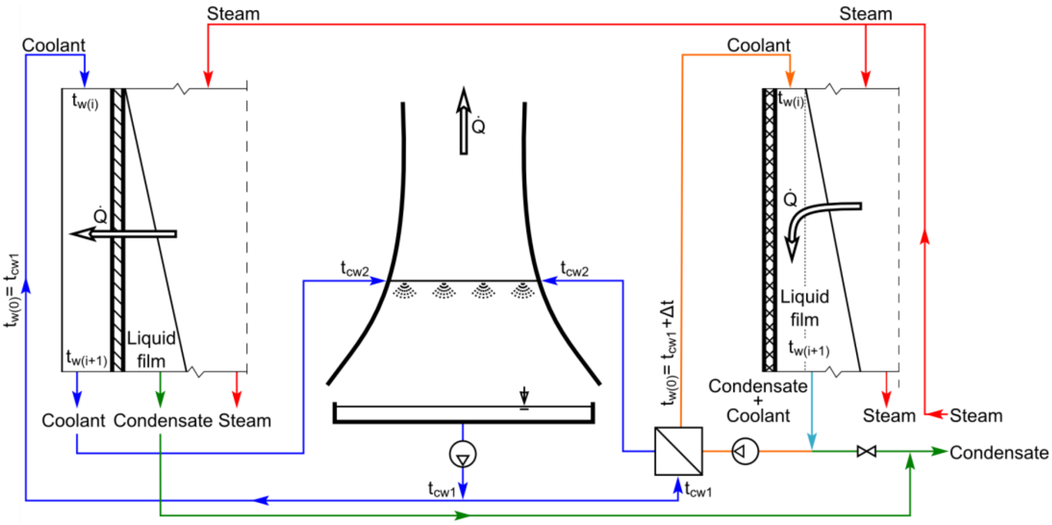
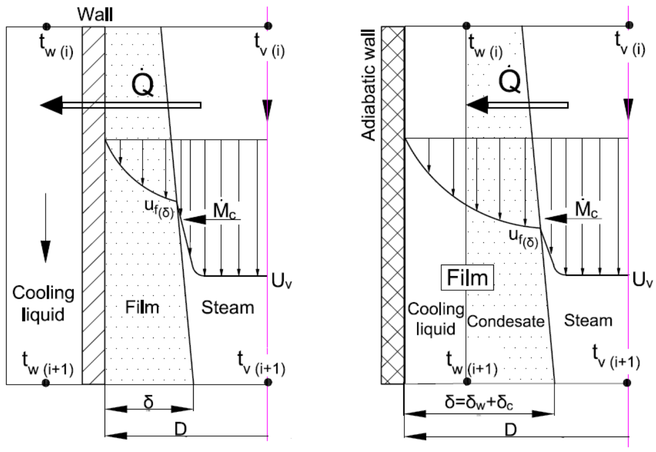


Figure 1: Circulation of the coolant, steam, and condensate

This paper contains an analytical comparison between the two above mentioned configurations of cooling water steam. In both cases, it is condensation of pure water steam in a one meter long vertical pipe. In the first configuration (mode 1), the coolant runs along the outside pipe wall. In the second configuration (mode 2), the coolant runs along the inside pipe wall and the outside pipe coat is insulated perfectly. Figure 1 shows the principle of cooling in both modes. Thanks to physical separation of the coolant from the steam flowing in the pipe, it is possible, in mode 1, to cool the pipe using chemically untreated water, which is cooled directly in the cooling tower.



a) Mode 1. *b) Mode 2.*

Figure 2: Scheme of different concepts of condensation heat exchangers

In mode 2, it is necessary to divide the cooling circuit into two parts. In the primary part, the same coolant may be used as in mode 1. Using a heat exchanger, this water cools the secondary water, which must have the same quality and composition as the steam and condensate. When using an equally efficient cooling tower and an equal flow rate of the primary coolant, the steam in mode 2, compared with mode 1, will be cooled by warmer coolant, the difference being the temperature drop necessary for the operation of the heat exchanger. When making an economic comparison of the cooling modes, it is also necessary to consider the power input of the pump providing the circulation of the secondary coolant.

2. Analytical model

Figure 2 shows a detail of a model of the condensate film, steam, and coolant flowing in both of the steam cooling configurations compared. In this model, the steam and the interface between the steam and the film are considered in a saturated condition. The heat transfer is considered one-dimensional. Both steam and the condensate film flow in the same direction.
The mathematical description uses empirical relationships to describe the behaviour of the liquid film and the heat flow transfer. It is solved in several subsequent steps through an iterative procedure and the following chapters describe and compare the partial relationships for both condensation configurations.

3. Calculation procedure

The physical properties of the liquid phase are related to the mean temperature. The physical properties of the gaseous phase are related to the input parameters of steam. At first, the output temperature of the coolant is proposed and the logarithmic temperature drop in the condenser is calculated from the equation:

$$\Delta t_{ln} = \frac{t_{w2} - t_{w1}}{ln\frac{t_s - t_{w1}}{t_s - t_{w2}}} \tag{1}$$

The transferred condensing power is then calculated from the following equation:

$$\Delta\dot{Q} = kl\Delta t_{ln} \tag{2}$$

From known transferred condensing power can be calculated the increase in the weight volume of the condensate from equation:

$$\Delta\dot{M}_{f(i)} = \frac{\dot{Q}_{(i)} l_u}{h_c} \tag{3}$$

For more accurate results the tube is virtually divided into several parts, which are solved independently. In this case, the total volume of the condensate flowing through the particular section is calculated from equation:

$$\dot{M}_{f(i)} = \dot{M}_{f(i-1)} + \Delta\dot{M}_{f(i)} \tag{4}$$

The Reynold's number for both modes is calculated from Eq(5) and the total coefficient of heat transfer for both modes from Eq(6). (Havlík, 2015 and Havlík, 2018).

$$\left.\begin{aligned} &\text{mode 1:} \quad Re_f = \frac{4\dot{M}_f}{\pi\eta_f d_i} \\ &\text{mode 2:} \quad Re_f = \frac{4(\dot{M}_f + \dot{M}_w)}{\pi\eta_f d_i} \end{aligned}\right\} \tag{5}$$

$$\left.\begin{aligned} &\text{For mode 1:} \quad \mathrm{k} = \frac{\pi}{\frac{1}{\alpha_f d_i} + \frac{1}{2\lambda}\ln\left(\frac{d_o}{d_i}\right) + \frac{1}{\alpha_o d_o}} \\ &\text{For mode 2:} \quad \mathrm{k} = \frac{\pi}{\frac{1}{\alpha_f d_i}} \end{aligned}\right\} \tag{6}$$

The condensing power and logarithmic temperature drop from heat calculation must be compared with proposed values. If the values from comparison are different, the temperature of the coolant has to be changed until a match is found. The next variable which influences the heat calculation is the film thickness. The film of the condensate removes space in the tube where the gas flows and that leads to an increased speed of the gas, heat transfer and the intensity of condensation. The film thickness must be estimated until the value of the weight volume of the condensate from Eq(7) is matches the value from Eq(4).

$$\dot{M}_{f(i)} = \frac{\pi\left[d_i - \left(d_i - 2\delta_{(i)}\right)\right]^2}{4}\bar{u}_{f(i)}\rho_{f(i)} \tag{7}$$

3.1 Calculation setting

For both cooling modes, we calculated the condensation of steam in a copper pipe with the outside diameter of 44 mm, wall thickness of 1.0 mm and total length of 1.0 m. The pipe was divided into ten sections, each being 0.1 m long for the purpose of the calculation. At the entry to the first section the temperature of the coolant was set to $t_{w(0)}$ = 20.0 °C and its flow rate is $\dot{V}_{w(0)}$ = 20.0 l/min.

In mode 1, there is constant pressure of $p_{w(0)}$ = 3.0 bars in the cooling system. In mode 2, the pressure of the coolant is equal to the saturation pressure of the steam condensing in the particular section. The saturation temperature of the steam at the entry to the pipe is set to $t_{s(0)}$ = 40.0 °C, the mass flow rate of the steam through the condenser is $\dot{M}_{s(0)}$ = 25.0 kg/h and the steam dryness at the entry into the condenser was set to $x_{s(0)}$ = 0.999. To calculate the physical properties of the steam phase and the liquid phase, the xSteam tables implemented in MS Excel were used (Holmgren, ver. 2.6).

4. Results and discussion

An analytical calculation of modes 1 and 2 was carried out with the same input parameters aiming to compare the two modes.

Figure 3 shows the developments of the heat flow density and the dryness of steam along the modelled pipe. The dryness of steam may be considered the most relevant parameter in comparing the efficiency of both the cooling modes. For mode 2, the dryness of steam decreases along the pipe much more rapidly than for mode 1. Also, for mode 2 the steam reaches the dryness of only 5 % at the outlet, whereas for mode 1 the output dryness of steam reaches 75 %. For mode 2, the maximum density of the heat flow is identified in the input section of the pipe. The density of heat flow reaches here approx. 2 kW/m and the density of heat flow follows a downward trend along the pipe, where in the output section of the pipe it reaches approx. 1.2 kW/m. Along the one meter long pipe in mode 1 the total thermal output transferred reaches 25% of the output transferred in mode 2. The downward trend of the density of heat flow is not so strong in mode 1 due to lesser warming of the coolant, which produces a more balanced temperature drop along the pipe.

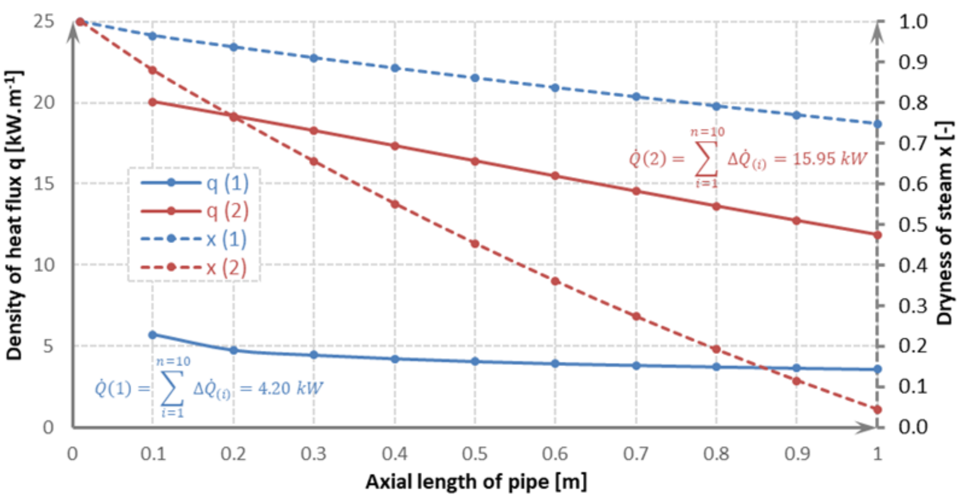
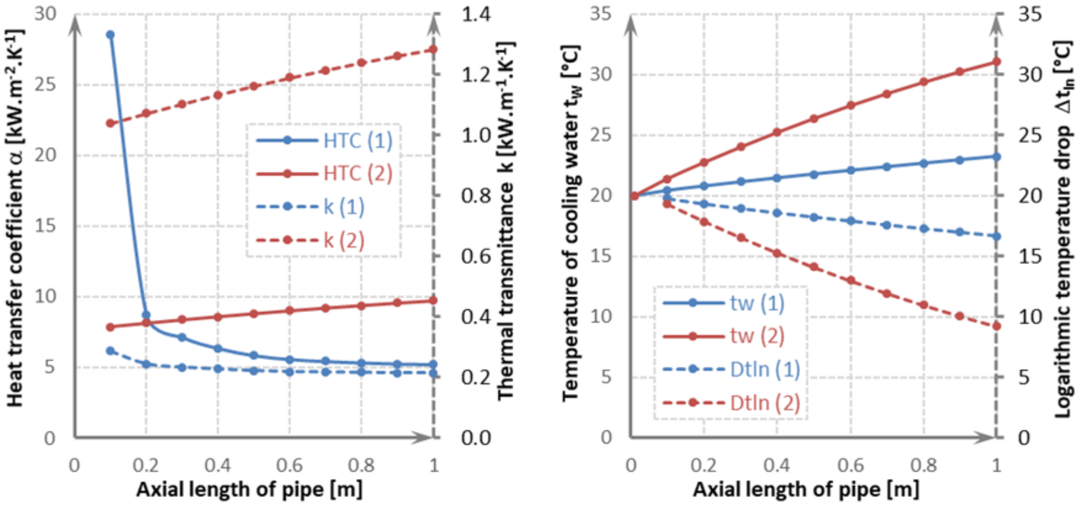


Figure 3: Graphical comparison of the transferred heat output and the dryness of steam expressed as a function of the pipe length

Figure 4(a) compares the coefficients of heat transfer on the inner pipe wall and the overall coefficients of heat transfer. HTC for mode 1 reaches a high value of around 28 kW/(m^{2}K) in the input section of the pipe and then it falls rapidly to approximately 7.5 kW/(m^{2}K). Subsequently, the value tends to decrease slowly until the end of the pipe is reached. This phenomenon is caused by the fact that in the input section of the pipe there is no condensate flowing and the heat is transferred directly from the steam jet to the wall. HTC for mode 1 follows a decreasing trend due to the growing thickness of the film, which increases the resistance against heat transfer in the mode of laminar flow. HTC for mode 2 reaches the initial value of approximately 7.5

kW/(m²K) because the coolant, which is already present in the input section, acts against heat transfer, like the condensate film, and follows an increasing trend. The reason is that an increased flow of the coolant along the pipe wall results in turbulent flow, which in turn leads to the reduction in heat resistance due to further steam condensation as the flow rate increases.



a) Heat transfer coefficient for modes 1 and 2. *b) Coolant temperature and temperature drop.*

Figure 4: Graphical comparison of calculated state parameters along the tube length.

The heat transfer coefficient for mode 1 reaches a four times lower value than the heat transfer coefficient for mode 2 because when cooling by water inside the pipe there is no heat transfer through the pipe wall and heat transfer to the coolant. Figure 4(b) shows the development of the coolant temperature and the mean logarithmic temperature drop for modes 1 and 2. It is apparent from the figure that the coolant temperature in mode 2 increases by 11.1 °C. This water also contains condensate, which causes an increase in the mass flow rate of the coolant in the axial direction. The coolant temperature in mode 1 increases by 3.2 °C and its mass flow rate is constant. As for the mean logarithmic temperature, it follows a decreasing trend along the pipe due to the warming of the coolant. Since the coolant is warmed up less in mode 1 than in mode 2, the drop in logarithmic temperature for mode 1 is more balanced with the output value of approx. 16.7 °C. For mode 2 the output logarithmic temperature drop is 9.2 °C.
The results presented above describe a situation where, for both configurations, we considered the same temperature of the coolant at the entry to the first section of the pipe ($t_{cw1} = t_{w(i=0)}$). Since cooling mode 2 needs an intercooling circuit including a pump and a heat exchanger (Figure 1) for its operation, it is not possible to realistically reach the same temperature of the coolant at the entry to the pipe. With the same cooling circuit parameters for modes 1 and 2, the coolant in mode 1 flows directly into the condenser, $t_{cw1} = t_{w(i=0)}$, whereas in mode 2, the primary coolant flows into a heat exchanger, where the secondary coolant is cooled for cooling the steam itself. When considering the heat exchanger temperature drop to be 3 °C in mode 2, the input temperature of the coolant will be

$$t_{w(i=0)} = t_{cw1} + \Delta t = 23.0\ °C \qquad (16)$$

For this reason, the cooling performance of the pipe will decrease by 12 % and the output steam dryness will increase to 16 %. Due to a lower transmitted power, the secondary coolant will be warmed less. Its temperature will increase by 9.7 °C instead of the theoretical value of 10.1 °C.

5. Conclusions

The paper presents two concepts of heat rejection when condensing steam in a vertical pipe. Mode 1 (the first case) describes standard heat rejection through a condenser pipe wall using a coolant flowing along the external pipe wall. Mode 2 (the second case) describes steam condensation in a vertical pipe where the coolant flows along the inner pipe wall together with the steam. An analytical procedure was developed and presented for the purpose of this paper.

A basic comparison shows that, on average, in mode 2 the heat transfer coefficient is four times greater although the steam related HTC is only greater by 75 %. This was achieved due to the fact that the heat does not pass through the pipe wall and thus heat transfer meets with less heat resistance. The cooling performance in mode 1 is one quarter of that in mode 2, which leads to a greater heating of the output coolant and a smaller temperature drop. Also, the output dryness of steam in mode 2 is almost zero, which means that almost all the steam condensates. In mode 1, for the same amount of steam to condensate, it would be necessary to use a longer pipe, a higher flow rate or lower temperature of the coolant.
When increasing the flow rate of the coolant, the output dryness of steam in mode 1 decreases. The impact of a change in the flow rate of the coolant on the dryness of steam in mode 1 is not as significant as in mode 2. In order to increase the condensing power of a pipe condensers, the presented comparisons show that mode 2 seems to be a suitable solution. Its application seems to be potentially suitable for heat exchangers regenerating thermal energy released by condensation of steam with minimal temperature drops. For the reasons mentioned, this type of heat exchanger can be recommended for the implementation of flue-gas condensers.

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