Analysis on the Calculating of Air Leakage Threshold which Caused the Condenser Vacuum Deterioration

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Abstract. There is an air leakage threshold under certain condenser heat load and air ejector output. When the amount of air leakage is less than the threshold, the condenser pressure is essentially the same, we can maintain a stable condenser vacuum, once exceed the threshold, condenser pressure begin to gradually larger, condenser vacuum will deteriorate. Considered the air leakage influences on the heat transfer coefficient of condenser, this paper had been established the one-dimensional steady-state heat transfer model of condenser to obtain the condenser pressure curve along with the amount of air leakage, the turning point on the curve is the air leakage threshold.

1. Introduction
A thermal power plant is a complex whole consisting of multiple equipment and systems. The performance of any one of the equipment and systems will affect the economic efficiency of the thermal power plant.

The condenser is an important auxiliary machine of the thermal power unit. It plays an important role in the thermal cycle of the power plant. The deterioration of its operating conditions will directly cause the steam exhaust pressure of the turbine to rise, the heat consumption of the unit will increase, the output will decrease, and the safe operation of the unit will be compromised. If the condenser vacuum deteriorates by 1 kPa, the heat consumption of the unit will increase by about 0.8%. Therefore, the normal operation of the condenser has an important impact on the safe and economic operation of the thermal power plant.

The heat transfer process of the condenser is quite special, and the shell side working fluid is accompanied by by phase transition during flow heat transfer, and because the condenser shell side is in a vacuum state, the outside air inevitably leaks through the steam or through the pipe gap. The relative amount of air increases with the condensation of steam along the process, gradually forming a multi-component complex steam stream on the shell side. The presence of non-condensing air also increases the heat transfer resistance of the shell side, forming a gas film on the surface of the pipe, thereby reducing the heat transfer coefficient.

The effect of the amount of air leaking into the condenser device on the heat of steam condensation is not only the effect of the amount of air leakage, but also the flow velocity of the steam-air mixture, that is, the suction capacity of the air ejector. The working condition of the ejector directly affects the degree of accumulation of air in the condenser. Figure 1 is a characteristic diagram of a typical air ejector and condenser.
As shown in the Figure 1, curve 1 is the characteristic curve of the extraction pressure, curve 2 is the characteristic curve of the condenser pressure, and curve 2 is composed of three sections: the AB section is a horizontal straight line called the thermal characteristic zone, the pressure in the condenser remains essentially the same in this zone. The pressure is mainly related to the flow of the discharged steam, the flow rate of the cooling water, the inlet temperature of the cooling water and the dirt on the pipe wall. The increase in air leakage has little effect on the condenser pressure and can be ignored. This area is the best working area for the condenser.

When the air leakage continues to increase and reaches a threshold $G_{a0}$, ie, point B, the air leakage begins to affect the condensation heat transfer of the steam, the condenser pressure and the extraction pressure curve gradually begin to approach. The BC section is called the deterioration transition zone, and condenser pressure begins to increase as the amount of air leakage increases. When the air leakage continues to increase to $G_{a2}$, ie, point C, condenser pressure and extraction pressure curve basically tend to be parallel. The CF section is called a vacuum deterioration zone, and as the amount of air leakage continues to increase, the vacuum deteriorates rapidly.

It can be seen from the above that there is a threshold for the effect of air leakage on the condenser. The condenser can maintain a normal vacuum before the threshold and begins to deteriorate after the threshold. Therefore, determining the critical amount of air leakage that causes the condenser vacuum to deteriorate can provide guidance for the safe and economic operation of the power plant condenser. Most of the existing literatures qualitatively analyze the influence of air leakage on the condensation heat transfer of the condenser. How to quantitatively calculate the critical air leakage has not been proposed yet. This paper proposes a method for calculating the critical air leakage that causes the condenser vacuum to deteriorate. Under the load and the output of the extractor, the method establishes the one-dimensional steady-state heat transfer model of the condenser to calculate the change of the condenser pressure when the air leakage changes, thus determining the critical amount of air leakage that causes the condenser vacuum to deteriorate.

2. **One-dimensional steady-state heat transfer model of condenser**

The air leakage mainly affects the condensation heat release of the steam side. The steam sweeps through the condenser tube bundle and the proportion of air in the steam-air mixture increases continuously,
thereby forming an air film on the surface of the cooling tube bundle, which impeding contact heat transfer between steam and cooling water wall. With the continuous condensation of steam, the influence of air leakage on the heat release of steam condensation increases gradually, which leads to the decrease of the heat transfer coefficient of condenser along the process. The effect of leaking air on the heat release coefficient of the steam-air mixture is shown in the figure 2 below:

![Figure 2](image-url)

**Figure 2.** Effect of air content on heat release coefficient of steam-air mixture [3]

In this paper, the heat exchange area S of the cooling pipe is taken as an independent variable, and each cooling water pipe is calculated from the outside to the inside in the direction of the steam flow. The steam flows through each of the cooling water pipes and condenses a part, so that the relative content of the air gradually increases. The heat release coefficient of the steam-air mixture flowing through the next tube bundle can be obtained from Figure.2, and the condensation amount of steam flowing through the tube bundle can be obtained. The above calculation is repeated until the suction capacity of the air ejector is satisfied to obtain a convergent solution, that is, the mass of the uncondensed steam-air mixture is equal to the mass of the steam-air mixture extracted by the air ejector.

The ultimate goal of this modeling is to obtain the pressure of the condenser under a certain amount of air leakage. With the continuous condensation of steam and the increase of air relative content, the heat transfer coefficient and steam partial pressure of condenser change all the time. Therefore, the mathematical models of heat transfer coefficient and steam partial pressure should be established respectively.

The heat transfer coefficient of the condenser is calculated by the division formula:

$$k = \frac{1}{a_s + \frac{1}{a_w} \frac{d_1}{d_2} + \frac{d_1}{d_2} \frac{1}{2\lambda} \ln \left( \frac{d_1}{d_2} \right) + R_f}$$

(1)

Where: $a_s$ — Steam side heat release coefficient, W/(m².℃); $a_w$ — Cooling water side heat release coefficient, W/(m².℃); $\lambda$ — Cooling pipe thermal conductivity, W/(m².℃); $d_1$ — Cooling tube outer diameter, m; $d_2$ — Cooling tube inner diameter, m; $R_f$ — Pipe fouling coefficient, W/(m².℃).
The fouling coefficient is calculated by the following formula:

$$R_f = \left( \frac{1}{a_s} + \frac{1}{a_w} \right) \frac{d_1}{d_2} \frac{d_1}{2\lambda} \ln \left( \frac{d_1}{d_2} \right) \left( \frac{1}{\beta} - 1 \right)$$

Where $\beta$ is the cleaning coefficient of the pipe, generally 0.85, so the fouling coefficient of the pipe is a single-valued function of the steam side heat release coefficient. The water side heat release coefficient is a fixed value, the fouling coefficient is only related to the steam side heat release coefficient, and the steam side heat release coefficient is a single value function of the relative air content. So the heat transfer coefficient of the condenser along the process can be calculated only by the relative air content along the process.

Relative air content along the process is calculated by the following formula:

$$\varepsilon_i = \frac{G_o}{G_o + G_s - G_{c(i-1)}}$$

Where $\varepsilon$ — Relative air content; $\varepsilon_i$ — Relative air content when flowing through the i-th cooling water pipe; $G_o$ — Air leakage flow ,kg/s; $G_s$ — Condenser steam flow ,kg/s; $G_{c(i-1)}$ — Steam condensation amount flowing through the i-1th cooling water pipe ,kg/s.

Along the steam flow, each part of the steam condenses through a cooling water pipe. Therefore, the steam flow of the condenser is always changing, and the relative content of the air in the steam-air mixture is gradually increased, so the relative content of air flowing through the lower tube can be calculated only by obtaining the condensation amount of steam flowing through the upper tube.

The experimental results [3] show that the heat release coefficient of pure steam is 17500 W/(m²°C), and the polynomial is approximated according to figure 2.calulation.

$$y = -0.0326x^5 + 0.7576x^4 - 6.7934x^3 + 30.04x^2 - 69.605x + 91.621$$
Where: \( y \) represents the ratio of the heat release coefficient of the air-containing steam \( a_s \) to the heat release coefficient of pure steam \( a_{s,0} \), and \( x \) represents the relative content of air leakage.

Therefore, according to the formula (4), the relative air content of each calculation area is obtained, and the steam side heat release coefficient can be calculated by the fitting formula (5). Because the experiment in Figure 1 only achieves the effect of the relative air content of 7% on the heat release coefficient of the steam-air mixture, when the steam has already condensed more than 99%, that is, the steam has passed over 99% of the cooling pipe, so it is close to the air cooling zone where the relative content of air at the suction port can generally reach more than 30%. In this area, the heat release coefficient has tended to the heat release coefficient of the air, and the forced convection heat release coefficient of the air is only 20-100 W/(m²·°C), therefore, when the relative content of air is more than 7%, the heat release coefficient of the steam-air mixture is treated as half of the heat release coefficient when the relative content of air is 7%. Here, the heat release coefficient at the suction port is approximately zero, because this tube bundle accounts for a small proportion of the total bundle, this treatment is more reasonable.

The basic heat transfer equation of the condenser can be deduced as the following formula:

\[
G_c = \frac{D_w c_w (t_s - t_i)(1 - \frac{1}{\exp\left(\frac{kS}{D_w c_w}\right)})}{h_e - h_c} \tag{6}
\]

Where: \( D_w \)—Cooling water flow, kg/s; \( c_w \)—Cooling water specific heat capacity, J/(kg·°C); \( t_s \)—Saturation temperature under steam partial pressure, °C; \( t_i \)—Cooling water inlet temperature, °C; \( S \)—Cooling tube heat exchange area, m²; \( h_e \)—Low pressure cylinder exhaust enthalpy, kJ/kg; \( h_c \)—Enthalpy of condensate, kJ/kg.

The steam partial pressure is obtained according to the ideal gas equation:

\[
P_s = \frac{P_c}{1 + 0.622 \frac{1}{x} \frac{G_a}{G_s}} \tag{7}
\]

Where: \( P_s \)—Steam partial pressure, kPa; \( P_c \)—Condenser pressure, kPa; \( x \)—Steam dryness in condenser.

The shell side steam is in a saturated state, so the temperature of the steam-air mixture is only related to the steam partial pressure. The partial pressure of the steam is obtained according to the above formula, and then the water vapor table can be used to obtain the saturation temperature \( t_s \) under the steam partial pressure. The relative amount of air in the steam discharged into the condenser is very low, so \( P_c \) is the saturation pressure corresponding to the initial value of \( t_s \).

According to the ideal gas equation, the density of the steam-air mixture at the suction port can be calculated:

\[
\rho_{mix} = \rho_s + \rho_a = \rho_s \left(1 + \frac{P_c M_s}{P_s M_a}\right) = \rho_s \left(1 + 1.61 \frac{P_c}{P_s}\right) \tag{8}
\]
Where: $\rho_s$ — Steam density, kg/m$^3$; $\rho_a$ — Air density, kg/m$^3$; $\rho_{mix}$ — Steam-gas mixture density kg/m$^3$; $P_s$ — Steam partial pressure at the suction port, kPa; $P_a$ — Air partial pressure at the suction port, kPa;

Saturated water steam is an ideal gas, the density of saturated water steam at standard atmospheric pressure is 0.597 kg/m$^3$. Therefore, if the pressure of water steam at the suction port is known, the density can be determined according to the ideal gas state equation, and the density of the steam-gas mixture can be obtained.

$$\rho_s = 0.961 \frac{P_s}{101.33}$$  \hspace{1cm} (9)

If the suction output $V_H$ of the air ejector is known, the flow of the steam-air mixture drawn by the air ejector can be calculated:

$$G_{mix}^H = \rho_{mix} V_H$$  \hspace{1cm} (10)

Where: $G_{mix}^H$ — the mass of the steam-air mixture that can be extracted by the air ejector, kg/s.

The above formula (1)—formula (10) constitutes a one-dimensional steady-state heat transfer model of the condenser. Under a certain amount of air leakage, set the saturation temperature $t_s$ corresponding to the pressure of the condenser to be the initial parameter of the iteration. The cooling water pipes of the condenser are sequentially calculated, and the iterative calculation is convergent under the condition that the mass of the steam-air mixture that is not condensed at the suction port is equal to the mass of the steam-air mixture extracted by the air ejector. The pressure corresponding to the convergence solution is the condenser pressure that the condenser can maintain under this amount of air leakage. By changing the amount of air leakage and repeating the above process, the pressure that the condenser can maintain under different air leakage amounts can be obtained.

Drawing the characteristic curve of condenser pressure and air leakage, the air leakage corresponding to the inflection point where the condenser pressure begins to increase is the critical air leakage of condenser vacuum deterioration.

3. Calculation method for critical air leakage of condenser

The specific method for calculating the critical air leakage of the condenser by using the one-dimensional steady-state heat transfer model of the condenser is as follows. The raw data required for calculation includes the amount of steam discharged into the condenser (or condenser heat load), the heat exchange area of the condenser, the flow rate of the cooling water, the temperature of the cooling water inlet, the amount of air leakage, and the output of air ejector.

1) The saturated steam temperature $t_s$ corresponding to the condenser pressure is set as the initial iteration parameter, and the initial temperature difference of the condenser is obtained by subtracting the cooling water inlet temperature from the steam side temperature.

2) According to the formula (5), $a_s$ is first calculated as the pure steam heat release coefficient 17500 to calculate the initial heat transfer coefficient $k$, and according to the formula (6), the steam condensation amount $G_c$ is calculated.

3) Calculate the relative air content $\varepsilon_1$ of the steam flowing through the first cooling pipe according to the formula (4), and calculate the $a_s$ corresponding to $\varepsilon_1$ according to the formula fitted in Figure.

2. Use formula (1) to calculate $k$ again and calculate $G_c$ according to formula (6). Compared with
the $G_c$ calculated in the second step, if the difference is greater than 1%, the average of the $G_c$ calculated in two steps is returned to the third step to calculate until the condition is met.

The relative air content $\varepsilon_1$, heat transfer coefficient $k$, steam condensation $G_c$ and steam flow rate $G_s$ after the first iteration are calculated.

4) According to the air relative content $\varepsilon_1$ of the third step, the present steam partial pressure $P_s$ is calculated according to formula (7), and then the saturated temperature $t_s$ under the partial pressure of the steam is obtained, which is used as the initial temperature of the next iteration, and the initial temperature difference of the next iteration is obtained by subtracting the inlet temperature of the cooling water.

5) Return to the second step, each cooling tube is calculated iteratively, and the number of iterations is the total number of cooling pipes. When the relative air content is greater than 7%, the steam heat release coefficient $a_s$ is half of the steam heat release coefficient when 7%. The mass flow of the steam-air mixture $G_{mix}$ at the suction port is calculated by iteration and compared with the maximum suction flow of the air ejector $G_{mix}^{\text{H}}$. If $G_{mix} = G_{mix}^{\text{H}}$, output $t_s$ and corresponding condenser pressure, the condenser vacuum corresponding to this pressure is the vacuum that can be maintained under given air leakage $G_a$, condenser heat load and exhaust volume. If $G_{mix}$, $G_{mix}^{\text{H}}$ are not equal, return to the first step and reset the iteration initial value until the condition is met.

6) Change the value of the air leakage $G_a$ and repeat the calculation process from step 1) to step 5) to obtain the pressure that the condenser can maintain under different air leakage, and plot the relationship curve between the condenser pressure and the air leakage, in which the air leakage corresponding to the inflection point is the air leakage threshold which causes the condenser vacuum deterioration.

4. Calculation example of air leakage threshold of a 600MW unit condenser

The original design data of a 600MW supercritical unit condenser is shown in Table 1:

| Number | Project                          | Unit   | Data   |
|--------|---------------------------------|--------|--------|
| 1      | Steam turbine exhaust volume    | kg/s   | 294.3  |
| 2      | Area of cooling surface         | m²     | 34600  |
| 3      | Process number                  |        | 1      |
| 4      | Cooling water flow              | kg/s   | 16700  |
| 5      | Cooling water inlet temperature | °C     | 24.7   |
| 6      | Cooling tube effective length   | m      | 26.2   |
| 7      | Cooling tube number             |        | 16830  |
| 8      | Cooling water flow velocity     | m/s    | 2.2    |
| 9      | Cooling tube specification      | mm     | Ø25x0.5|
| 10     | Tube plate thermal conductivity | W/(m²°C)| 13.8   |
| 11     | Cleaning factor                 |        | 0.85   |
| 12     | Air leakage                     | kg/h   | 81.6   |
| 13     | Water jet air ejector Extraction efficiency | m³/s | 1.67   |
| 14     | Air ejector working water temperature | °C | 24.7   |

4.1. Condenser pressure vs. air leakage

Using the one-dimensional steady-state heat transfer model of the condenser, the characteristic curves of the condenser pressure variation with the change of air leakage can be obtained when the inlet temperature of cooling water is 24.7°C at 120%, 100%, 80% and 60% load.
As shown in Figure 3, the characteristic curve of the condenser pressure calculated by the model with the amount of air leakage is in good agreement with the experimentally obtained curve. Under a certain steam load, the pressure of the condenser changes with the amount of air leakage. The pressure in the condenser is basically unchanged when the air leakage is small. Once the air leakage reaches a threshold, the pressure rises rapidly and the vacuum of the condenser begins to deteriorate.

Taking the characteristic curve of condenser pressure under 100% steam load as an example, when the air leakage rate varies in the range of $0 \sim 0.045 \text{ kg/s}$, unchanged and stable, and the condenser runs in the thermal characteristic zone. When the air leakage varies from $0.045 \sim 0.055 \text{ kg/s}$, the condenser pressure has shown a slow upward trend and the condenser vacuum begins to deteriorate, but the condenser vacuum remains at a higher level, and the condenser operates in a deteriorating transition zone. When the air leakage rate varies in the range of $0.055 \sim 0.075 \text{ kg/s}$, the condenser pressure rises sharply, so that the condenser cannot maintain normal working vacuum. The condenser then operates in the vacuum deterioration zone.

As can be seen from Figure 3, there is an inflection point in the characteristic curve of condenser pressure under each load, and the corresponding air leakage at the inflection point is the critical air leakage which causes the deterioration of condenser vacuum. At 60% load, the critical air leakage is 0.025 kg/s, at 80% load, the critical air leakage is 0.035 kg/s, at 100% load, the critical air leakage is 0.045 kg/s, and at 120% load, the critical air leakage is 0.055 kg/s. It can be seen that the susceptibility of condenser vacuum to air leakage decreases and the critical air leakage increases with the increase of load.

4.2. Influence of cooling water inlet temperature

As the season and weather change, the cooling water inlet temperature will also change. When the cooling water inlet temperature becomes $15^\circ\text{C}$, the curve of the condenser pressure with the amount of air leakage under different steam loads is shown in Figure 4:
Compared with the inlet temperature of the cooling water at 24.7 °C, it can be seen that the critical air leakage from the thermal characteristic zone into the deteriorated transition zone is increased. Taking the 100% load as an example, when the inlet temperature of cooling water is 24.7 °C, the critical air leakage is 0.045 kg/s, and when the inlet temperature of cooling water is 15 °C, the critical air leakage is increased to 0.055 kg/s.

It can be seen that under the condition of low inlet temperature of cooling water, the influence of air leakage volume on condenser heat transfer is reduced. This is because the cooling water temperature is lower, the initial temperature difference of heat transfer is larger, and the influence of air leakage on condensation heat transfer is reduced to a certain extent by the increase of initial temperature difference.

4.3. Influence of the output of the air ejector.
The mixture of uncondensed steam and air at the suction port is extracted by the air ejector, so the performance of the air ejector affects the accumulation of air leakage in the condenser. When the output of the air ejector increases from 1.67 m³/s to 3.34 m³/s, the curve of condenser pressure with air leakage at the cooling water inlet temperature of 24.7 °C and 100% steam load is shown in Figure 5:
It can be seen from Figure 5 that when the extraction output is 1.67 m$^3$/s, the condenser enters the deterioration transition zone with the air leakage of 0.045 kg/s. When the extraction output is increased to 3.34 m$^3$/s, the condenser starts to enter the deterioration transition zone when the air leakage is 0.1 kg/s. It shows that the extraction output is increased, the thermal characteristic zone of the condenser is extended, and the critical air leakage is increased. This is because as the extraction output increases, in the case of a small amount of air leakage, a smaller extraction output can timely and continuously evacuate the leaked air, and as the amount of air leakage increases, the smaller extraction output cannot completely extract all the leaked air, resulting in air accumulation, which makes the condensation heat transfer of the condenser deteriorate and enters the deteriorated transition zone. The larger extraction output can still extract all the leaked air, thus maintaining the normal operation of the condenser.

5. Conclusion

In this paper, a one-dimensional steady-state heat transfer model of condenser is established, and the method of calculating the air leakage threshold by model is introduced in detail. The characteristic curve of condenser pressure with air leakage is calculated by example, which is in good agreement with the experimental curve, and the accuracy of this method is verified.

The main conclusions of this paper are as follows:

1) There is a threshold for the influence of air leakage on the condenser. Before the threshold, the condenser pressure is basically constant, after the threshold, the condenser pressure begins to increase rapidly.
2) When the cooling water flow, inlet water temperature and air ejector output remain unchanged, the air leakage threshold increases with the increase of load.
3) When the load, cooling water flow and inlet water temperature remain unchanged, the air leakage threshold increases with the increase of air ejector output.
4) When the load, cooling water flow and air ejector output remain unchanged, the air leakage threshold increases as the cooling water inlet temperature decreases.

References

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