New Tube Arrangement of Condenser for Power Stations*

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Steam pressure loss in a tube bundle significantly influences condenser performance. Based on this fact, a new tube arrangement will be developed. The analytical study shows the advantage of this improvement: up to 20% higher heat transfer performance than from our former design⁽¹⁾. The method applied to estimate performance is a steam flow analysis using the finite volume method. To confirm the analytical results, hydraulic analogue tests with water are carried out. This new type condenser will be operated in power stations in the near future.

Key Words: Condenser, Power Station, Tube Arrangement, Numerical Analysis, Heat Transfer

1. Introduction

The condenser is one of the largest heat exchangers in a power station. Therefore, it is very attractive to increase the heat transfer performance of condenser. This would increase the thermal efficiency of a power station by reducing turbine exhaust pressure and the size of the condenser itself. Increased thermal efficiency decreases fuel consumption and contributes to environmental protection, because it reduces emissions of carbon dioxide. A condenser has a large number of tubes (titanium, aluminum brass, and so forth). If a condenser has a higher heat transfer performance, it needs less cooling area, which would reduce the cost and the size of the condenser. Furthermore, it would decrease the construction cost of the plant.

The performance of the condenser is greatly affected by the tube arrangement. If the tubes are not arranged properly, the amount of heat transferred decreases and turbine exhaust pressure, namely condenser pressure, increases. As a result, thermal

efficiency of power station is decreased.

Development of a high heat transfer performance condenser consists of the following three steps:

- 1) Determining basic specification of the tube arrangement by numerical analysis
- 2) Confirming analytical results by hydraulic analogue tests with water
- 3) Proving validity of new tube arrangement by a model condenser test under actual steam conditions

Step 3) is being planned. This paper reports on steps 1), 2), and a design of a condenser tube arrangement for 290 MWe power station.

Nomenclature

- c_{PL} : specific heat of condensate
 d_i : inner diameter of tube
 d_o : outer diameter of tube
 g : gravitational acceleration
 G : condensate flow rate
 Gal : Galileo number ($=g \cdot d_o^3 / \nu_L^2$)
 H : phase change number ($=c_{PL}(T - T_w)/L$)
 h_i : heat transfer coefficient of cooling water
 h_o : condensing heat transfer coefficient with non-condensable gas
 h_s : condensing heat transfer coefficient without non-condensable gas

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Fig. 1 Effects of tube bundle thickness on pressure loss and heat duty

- K : overall heat transfer coefficient
 L : latent heat of condensation
 Nu : Nusselt number for cooling water
 $(= h_i \cdot d_i / \lambda_c)$
 Nu_L : Nusselt number for condensation
 $(= h_s \cdot d_o / \lambda_L)$
 Pr : Prandtl number of cooling water
 Pr_L : Prandtl number of condensate
 Re : Reynolds number of cooling water
 Re_L : two-phase Reynolds number $(= d_o \cdot v / \nu_L)$
 R_f : fouling factor
 S : heat transfer area
 T : condensing temperature
 T_c : cooling water temperature
 T_w : temperature at the outer surface of tube
 v : Steam flow velocity in tube bundle
 W : mass fraction of air
 ν_L : kinematic viscosity of condensate
 λ_c : thermal conductivity of cooling water
 λ_m : thermal conductivity of tube
 λ_L : thermal conductivity of condensate

2. Concept of New Tube Arrangement

Figure 1 shows the effect of tube rows (namely tube bundle thickness) on pressure loss and heat duty when the total number of tubes is fixed. As the tube bundle is thinner, a lower pressure loss can be obtained, resulting in a higher heat duty. This is the calculation result from SPOC1 which is discussed later. From these kinds of basic calculations, the following requirements for a good tube arrangement are deduced.

- (1) Minimize pressure loss of the steam flow inside tube bundle
- (2) Prevent air blanketing of the non-condensable gases (mainly air)

Furthermore, the overall configuration of the tube

Fig. 2 Tube arrangement for model condenser

bundle should be designed considering the static pressure distribution around the tube bundle.

Figure 2 shows the tube arrangement of the model condenser with the above taken into account and Table 1 shows the specification. As the steam from the turbine into the condenser and then passing around the tube bundle flows downward, it gradually decreases in flow rate. Based on this flow rate variation, a wedge-shaped tube arrangement was adopted. This configuration prevents static pressure from decreasing due to an excessively high steam velocity around the tube bundle, which causes the steam flow into the tube bundle to decrease. The steam is flowing into the tube bundle and condensing on the surfaces of the cooling tubes. The steam flows into the upper and

Table 1 Specification of model condenser

Steam Pressure	5065Pa
Cooling Water Flow Rate	1110 kg/s
Cooling Water Temperature	295 K (22°C)
Heat Duty	22 MWt
Tube Material	Titanium (JIS TTH340W)
Tube Outer Diameter	22 mm
Tube Thickness	0.4 mm
Effective Tube length	7420 mm
Number of Tubes	1852 (ACZ:120)

lower steam aisles at the center of the tube bundle, then into the right and left air cooling zones. While flowing up through the cooling tubes in the air cooling zone, the steam including air further condenses, causing air concentration to rise. The passages of the air cooling zones are tapered upward to prevent the volumetric flow rate from decreasing due to condensation, causing heat transfer performance to decrease.

The steam including air to 20 - 30% through the air cooling zones is led to the venting pipes at the center of the tube bundle. Even extraction from each air cooling zone to the venting pipes is obtained by flow restrictions. They are connected to a vacuum pump or steam ejector.

Condensate on the surfaces of the cooling tubes in the tube bundle flows down while maintaining contact with the steam across the tube bundle. This contact generally prevents condensate depression; that is, cooling of the condensate below saturation temperature. The condensate ultimately flows into the hot-well at the bottom of the condenser.

3. Numerical Analysis for Model Condenser

3.1 Analysis method

To determine the tube arrangement of the model condenser, fluid analysis code SPOC1 and STAR-CD were applied. As a special-purpose code, SPOC1 is easy to use, but does not deal with steam velocity distribution at the inlet boundary. In contrast, STAR-CD, which is a general-purpose code, has no such restrictions, but is more laborious than SPOC1 in preparing meshes for various tube arrangements and furthermore must be customized to deal with condensation and pressure loss in the tube bundle. Thus, SPOC1 is used to optimize the tube arrangement by carrying out a parameter survey. Next, STAR-CD is applied to a detailed analysis for the optimized tube arrangement.

3.2 Analysis by SPOC1

SPOC1 (Simulation of Power Condensers) is a performance calculation code for condensers developed by Heat Transfer and Fluid Flow Service (HTFS) in the UK. This computer code calculates the detailed two-dimensional steady-state flow and the

Fig. 3 Velocity distribution with SPOC1

(UNIT:wt%)

Fig. 4 Concentration distribution with SPOC1

heat transfer distribution in a wide range of power plant condenser configurations with numerical differential equations using well-established iterative methods. As SPOC1 can handle steam and air mixture flow in condenser tube bundles, one can easily identify an area of air blanketing. SPOC1 is a compact program and it is easy to change the configuration of the tube bundle and the internal baffles. This program is applied to investigate the optimum configuration of tube bundles under ideal conditions. The SPOC1 model can have only one air extraction point in the tube bundle and cannot deal with a maldistributed steam flow from turbine exhaust. Therefore, only half of the tube bundle (Fig. 3) can be calculated with a uniform velocity distribution from turbine.

Figure 3 shows SPOC1 calculation result of velocity distribution. Steam flows into the tube bundle from the entire peripheral area. As the steam flows upward and the condensate flows downward in the lower part of the tube bundle, they contact each other in this area. This contact prevents the condensate depression, which causes loss of the heat rate and dissolution of oxygen to condensate. Figure 4 shows the air concentration distribution. The highest concentration area of the air only occurs in the outlet of the air cooling zone and there is no air blanketing.

From analytical result, the steam temperature drop due to pressure loss 325 Pa between the condenser inlet and the ACZ (Air Cooling Zone) outlet is at most 2°C, which permits a larger temperature difference between steam and cooling water. This is the reason for the high performance.

3.3 Analysis by STAR-CD

The general-purpose analysis code STAR-CD has the following features:

- 1) Covers transients, compressibility, turbulence, heat transfer, chemical reaction and multi-phase flow.
- 2) Permits not only BFC (body fitted coordinates) lattice but nonstructural lattices to be used, and is also applicable to models having complicated shapes.
- 3) Can be used as an open system linked with the user's subroutine or another CAE system.

When applying STAR-CD to a condenser, the fluid in the condenser is a compressive mixture of steam and air. Besides, both fluids have the same velocity. The condensation and pressure loss in the tube bundle were dealt with using the user's subroutine. The condensate flow rate is given by

$$G = S \cdot K(T - T_c)/L \quad (1)$$

and the overall heat transfer coefficient is given by

$$1/K = 1/h_o + d_o/d_i \cdot 1/h_i + d_o/(2\lambda_m) \cdot \ln(d_o/d_i) + R_f \quad (2)$$

where for the heat transfer coefficient of the cooling water and the condensation, Dittus-Boelter's equation⁽²⁾ and Fujii's following equations^{(3),(4)} were used respectively:

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (3)$$

$$h_o = h_s \cdot A \cdot \exp(-m \cdot W) + B \cdot \exp(-n \cdot W) \quad (4)$$

$$A = 0.83(T - T_w)^{-0.15} + 0.1 \ln(v) \quad (5)$$

$$B = 0.21(T - T_w)^{0.25} - 0.09 \ln(v) \quad (6)$$

$$m = 3.7v^{-0.12} \quad (7)$$

$$n = 19(T - T_w)^{0.20} v^{-0.3} \quad (8)$$

$$Nu_L = 1.0125 - 0.2875 Y^{0.25} (Y^{0.58} - Re_L^{1.16}) / (Y^{0.58} + Re_L^{1.16}) \quad (9)$$

$$Y = Ga_L \cdot Pr_L / H \quad (10)$$

The velocity distribution was obtained with the tube arrangement of the model condenser on the assumption that there was a uniform steam flow at the condenser inlet. The results are shown in Fig. 5. Because there was no velocity distribution at the condenser inlet, the flow pattern is symmetrical with respect to the vertical center line and nearly the same as the results with the half tube bundle shown in Fig. 3 obtained by SPOC1.

4. Hydraulic Analogue Test with Water

Thus far, the steam flow in the condenser has been obtained by fluid flow analysis. On the other

Fig. 5 Velocity distribution with STAR-CD

hand, in addition to the small pressure loss, the steam flow in the tube bundle is so complicated and delicate that fluid analysis might not represent it accurately. It is useful to confirm the steam flow by Drummond's method⁽⁵⁾, which simulates a steam flow with a water flow. The test apparatus is shown in Fig. 6. It has a tube arrangement similar to that of the model condenser and its acrylic tubes have small holes to suck in water. Water suction corresponds to condensation in the condenser, establishing hydraulic analogue relation. The amount of water sucked into acrylic tubes is proportional to the water head, which corresponds to the relation between the amount of condensation and the temperature difference of steam and cooling water in the condenser. Furthermore, the water flow rate in the hydraulic analogue test is determined to provide the same Reynolds number in the condenser. In the test, nylon powder floating on the free surface of the flow were used to observe the stream line and non-stagnation.

Figure 7 shows the result of observation with a uniform velocity distribution, which agrees well with the results in Fig. 3 and Fig. 5. To simulate the maldistributed steam flow at the inlet of the condenser, the flow passage was narrowed to one-third with blocks laid at the inlet of the test apparatus. Also to prevent the flow from right to left around the tube bundle, a partition plate was installed under the tube bundle. The flow pattern observed in this formation is shown in Fig. 8. In the lower part of the tube bundle, water flows into the central aisle from right

Fig. 6 Hydraulic analogue test facility

Fig. 7 Flow pattern with uniform inflow

Fig. 8 Flow pattern with maldistributed inflow and partition plate

Fig. 9 Velocity distribution with maldistributed flow and partition plate with STAR-CD

Fig. 10 Condenser tube arrangement for 290 MWe power station

and left. On the other hand, in the upper part of the tube bundle, water flows to the center from right and left, except for some flow across the tube bundle at the top. The results here agree well with those of the STAR-CD analysis shown in Fig. 9.

5. Condenser Configuration for 290 MWe Power Station

The tube arrangement of the condenser for 290

MWe power station is shown in Fig. 10. It consists of four sets of tube bundles and two bundles each have a common waterbox at both ends. Each tube bundle has the similar configuration to the model condenser. Venting pipes from each tube bundle are connected to a vacuum pump or steam ejector. The specification of this condenser is shown in Table 2.

SPOC1 calculation results of velocity and air concentration distribution in the tube bundle are

Table 2 Specification of condenser for 290 MWe power station

Steam Pressure	5065Pa
Cooling Water Flow Rate	16500 kg/s
Cooling Water Temperature	295 K (22°C)
Heat Duty	130 MW
Tube Material	Titanium (JIS TTH340W)
Tube Outer Diameter	25.4 mm
Tube Thickness	0.5 mm
Effective Tube length	14350 mm
Number of Tubes	15400(ACZ:810)

Table 3 Comparison of SPOC1 and STAR-CD with HEI

	SPOC1	STAR-CD	HEI
Overall Heat Transfer Coefficient(W/m ² K)	4320	4490	3590
Pressure Loss (Pa)	(1.20)	(1.25)	(1)
Pressure Loss (Pa)	425	360	NA

*Fouling factor is not assumed in the above calculation.

Fig. 11 Velocity distribution with SPOC1 for 290 MWe condenser

Fig. 12 Air concentration distribution with SPOC1 for 290 MWe condenser

Fig. 13 Velocity distribution with STAR-CD for 290 MWe condenser

shown in Fig. 11 and Fig. 12 with uniform velocity distribution. Steam flows smoothly into the air cooling zone and no air blanketing can be seen. In the actual condenser, the velocity distribution of steam from the turbine is not uniform. Figure 13 shows the velocity distribution in the right half (two tube bundles) of a vertically symmetrical condenser obtained by a STAR-CD analysis with the velocity distribution at the condenser inlet taken into account. Furthermore, partition plates are installed under the tube bundle. Condenser performance, which is obtained from SPOC1 and STAR-CD, is shown in Table 3 compared to HEI⁽⁶⁾.

6. Conclusions

A new tube arrangement is developed to enhance the heat transfer performance of a condenser for a power station. The objectives of this tube arrangement are to minimize the pressure loss of steam across the tube bundle and enhance air extraction from the bundle. The steam pressure loss and the heat transfer performance of the tube arrangement is confirmed by a fluid flow analysis (see Table 3) and a hydraulic analogue test with water. These found:

1. Steam pressure loss across tube bundle is significantly reduced.
2. Tube bundle has more than 20% higher overall heat transfer coefficient than our former design

(HEI).

3. Non-condensable gas is smoothly exhausted to the venting pipe through the air cooling zone.

To confirm the performance of new tube arrangement, a test facility with power plant steam conditions has been built. The facility has the model condenser tube arrangement shown above and can carry out many kinds of test on new design condensers.

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