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A Numerical Study on Steam Flow and Heat Transfer of Pannier-arrangement Condensers

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Abstract — Pannier-arrangement condensers are usually adopted in the turbine generator units of combined cycle power plants. Optimization of operating performance and economy is an important goal, which requires accurate understanding of flow and heat transfer effects in the condenser. The tube bundle arrangement and steam flow behaviors of pannier-arrangement condensers are very different from those of common condensers. The physical model for existing numerical simulation program of condenser is refined by constructing the correlations for flow resistance and condensation heat exchange coefficient in which the influences of steam flow direction are considered according to available experimental data. The adaptability of the developed physical model and simulation program of pannier-arrangement condenser is verified with available experimental data.

Key words : Condenser, Flow, Heat transfer, Numerical simulation

1. Introduction

Combined cycle power generation is an important development direction for coal power generation in the 21st century. Steam extraction and heat regeneration systems aren't adopted on the steam turbines in combined cycle power plants, and steam turbines are usually designed as side exhaust steam or axial exhaust steam arrangement in order to reduce the height of workshops and be arranged conveniently. Accordingly, condensers are arranged on the ground position as high as steam turbines. In this type of pannier-arrangement condenser, steam flows across the tube bundle at the foreside and the steam flow direction turns rapidly at the tail. The physical model for steam flow and heat transfer, the method for determining heat transfer coefficient and the requirement of tube bundle arrangement are very different from those of common transverse underlung condensers. Therefore, it is necessary to develop the research of steam flow and heat transfer behavior of pannier-arrangement condensers as well as the product design. Numerical analysis method can reveal detailed steam flow and heat transfer processes inside this type of condensers, as well as predict the influence of all kinds of factors on the economy of turbine unit. The current condenser numerical simulation approaches are not suitable for pannier-arrangement condensers, since they all don't allow for the influences of steam flow direction on steam flow resistance and heat transfer coefficient.

In this paper, considering the steam flow features of pannier-arrangement condensers, the correlations for flow resistance and condensation heat exchange coefficient which allow for the influence of flow direction according to the available experimental data[1][2] are constructed. On this base, a new condenser simulation program is developed, and used to calculate and discuss the performance of a pannier-arrangement test condenser. The adaptability of the developed approach is verified by comparing the simulation results with the experimental data.

2. Physical Model and Numerical Approach

The steam flow space of a condenser is axially divided into many steam rooms by clapboards. In these steam rooms, the cooling water temperature variation is not big and the axial steam motion is lim-

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ited by clapboards. Then, the flow and condensation heat exchange process of the steam-air mixture in tube bundles can be simplified as two dimensional flow and heat transfer process which changes only on the plane vertical to the tube axis[3].

According to the two dimensional model, the steam-air mixture flow in tube bundle areas of every steam room can be simplified as two dimensional flow of perfect steam-air mixture in a porous medium with distributed resistances and distributed mass sinks.

Governing equations group which describes this kind of flow is mainly composed of mass-conservation equation, momentum equations and air component equation. These equations in Cartesian coordinates can be shown as an unified form that

$$\frac{\partial}{\partial x}(\varepsilon\rho u\varphi)+\frac{\partial}{\partial y}(\varepsilon\rho v\varphi) = \frac{\partial}{\partial x}\left(\varepsilon\Gamma_{\varphi}\frac{\partial\varphi}{\partial x}\right)+\frac{\partial}{\partial y}\left(\varepsilon\Gamma_{\varphi}\frac{\partial\varphi}{\partial y}\right)+S_{\varphi} \tag{1}$$

where u, v are components of mixture velocities in the x and y direction; ρ is the mixture density; ε is the porosity factor of the porous medium; ϕ is an independent variable; Γ_ϕ and S_ϕ are respectively the diffusion coefficient and source term, where ϕ represents u, v, the air concentration q and the number 1 respectively. Eq. (1) becomes the momentum equations, the air component equation and the mass-conservation equation respectively; the forms of Γ_ϕ and S_ϕ depend on the meaning of ϕ, and expressions for them are described by Yu[3].

After dispersing the governing equations group by control-volume integration method, the dispersed equations group is solved by semi-implict method for pressure linked equations. The obtained results include the distributions of the velocity, pressure, temperature and air concentration of the steam flow field on the condenser shell side. In the calculating process, it is necessary to determine the local steam flow resistance coefficient and steam side heat exchange coefficient by existing correlations on the basis of the local steam flow velocities, local air concentration and local tube bank condensate inundation.

At present, correlation for steam flow in the dry tubes bank is generally adopted as the correlation of steam flow resistance coefficient in tube bank area for condenser numerical simulations, but it doesn't allow for the influence of condensate and steam flow direction on the steam flow resistance. Fig. 1 shows the schematic representation of various condensate forms and flow tendencies for various steam flow directions[4]. Davidson and Rowe[5] recognized that the single phase resistance correlation can rationally represent the pressure drop of two phase flow for horizontal steam flow, but it would bring considerable error for downward and upward flow.

Fig. 2 shows the comparision of the single-phase prediction value with the experimental value of tube bank resistance for upward flow and downward flow[6]. The data shows that the single-phase prediction value is less than the experimental result for upward flow and the former is more than the latter for downward flow. In this paper, the resistance performance of upward flow and downward flow are

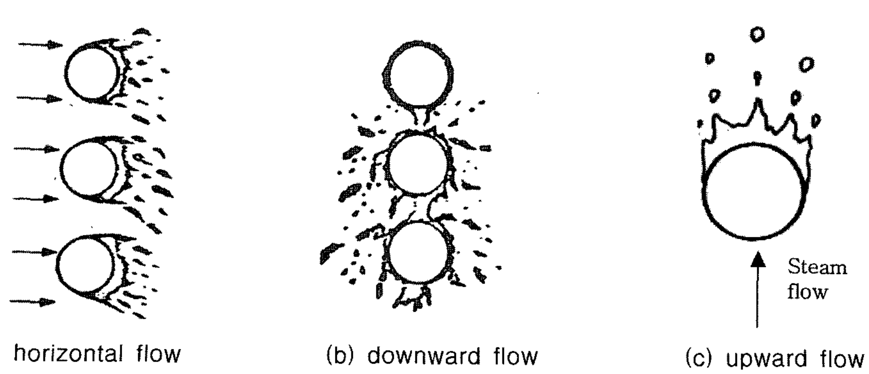
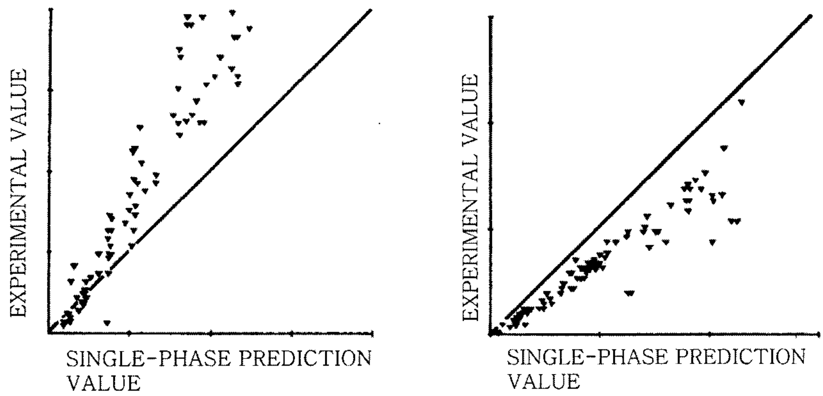


Fig. 1. Schematic representation of various condensate forms and flow tendencies for various steam flow directions.



(a) upward flow (b) downward flow

Fig. 2. Comparison of single-phase prediction value with experimental value of tube bank resistance.

revised according to the experimental data by Brickell[6], for upward flow:

$$\xi_u/\xi_D=1.732 \tag{2}$$

for downward flow:

$$\xi_d/\xi_D=0.85 \tag{3}$$

Where ξ is two-phase resistance coefficient, u represents upward flow, d represents downward flow, ξ_D is single-phase resistance coefficient.

In the absence of vapour velocity as condensate flows by gravity on to lower tubes in the bundle, the condensate should thicken around a tube and the condensate heat transfer coefficient should therefore decrease. Vapour velocity and inundation are important factors that affect the condensate heat transfer on the shell side. Computer codes for condensers are generally based on the Nusselt equation in which the influences of vapour shear and inundation, namely

$$\frac{\alpha_n}{\alpha_N}=\left(\frac{w}{c}\right)^{-s}\left[1+0.0095Re_v^{11.8/\sqrt{Nu_N}}\right] \tag{4}$$

where s is correction factor for inundation, α_n is mean heat transfer coefficient on the n^{th} tube, α_N is Nusselt heat transfer coefficient, w is total inundation rate on the n^{th} tube and c is condensation of the n^{th} tube.

None of these present day condenser numerical simulation models so far presented accounts of the interactive influence of vapour shear and inundation for the correlation of heat transfer coefficient on the shell side. The literature[6] confirms that the correction factor of inundation for upward flow is higher than that for downward flow. In this paper the influence of vapour flow direction on s is considered according to the experimental data by Brickell[6], namely
for upward flow:

$$s=\tan^{-1}|v/u|/(\pi/2)\times 0.12+0.223 \tag{5}$$

for downward flow:

$$s=0.223 \tag{6}$$

where u, v represent respectively the horizontal component and vertical component of steam flow velocity.

3. Configuration and Parameters of the Experimental Condenser

The experimental condenser is a single shell, single

Table 1. Main geometric parameters and operation conditions.

Condenser pressure (Pa)	27670	Cooling area (m^2)	38.89
Cooling water temperature (K)	290	Mass flow rate of cooling water (kg/s)	196.04
Water velocity (m/s)	1.19	Mass flow rate of exhaust steam (kg/s)	2.032
Tube length (m)	1.219	Cooling multiple	96.47
Number of tubes	400	Tube size (mm)	ϕ25.4×1.25

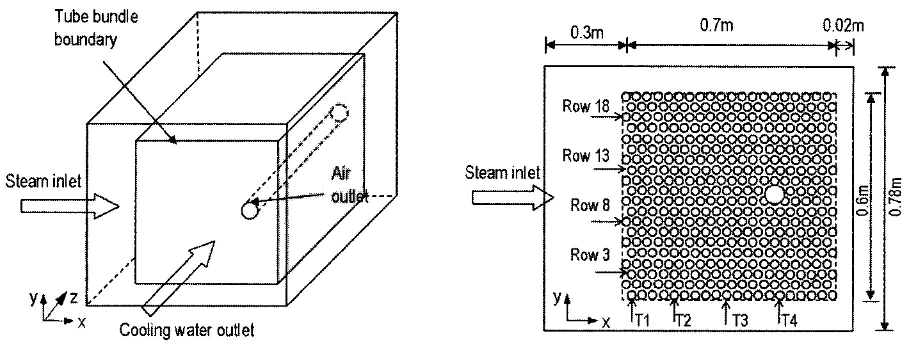


Fig. 3. Experimental condenser.

pass and pannier-arrangement condenser, whose main design parameters and operation conditions are shown in Table 1.

Fig. 3 shows the particular of tube bank. The tube bank is composed of 20×20 tubes in staggered arrangement. From Fig. 3 the main characteristic of this type of condenser is vapour flows into tube bank from the flank, and air and uncondensed vapour are drawn out at outlet. The computation supposes that the capability of the air pump is great enough to draw out all uncondensed mixture of air and vapour. The parameters of the mixture at outlet are given by computation results[5]. This experimental condenser is not a condenser with reasonable configuration, but it is suitable to examine the rationality and adaptability of a numerical simulation approach.

4. Results and Comparison

Even velocity inlet is given as the inlet boundary condition the computation is performed using a grid of 150×100 divisions. The computer code is written in Fortran 77, and the calculations are iterated for 3000 iterations. Figs. 4 and 5 show respectively the distributions of vapour velocity vectors and vapour streamlines on the shell side of the experimental condenser. From the figures it can be seen that after steam flows into tube bank area three types of flow are in existence: horizontal flow is dominant in the left part of tube bank area; downward flow is dominant in the top right part; upward flow is dominant in the under right part.

Fig. 6 shows the distributions of condensate per square meter in tube bank area are respectively obtained by the numerical model considering the influence of steam flow direction on flow resistance(not considering the influence of steam flow direction on inundation), the experiment[11][12] and the single-phase model in which the influence of flow direction on resistance is not considered. From the figure it can be

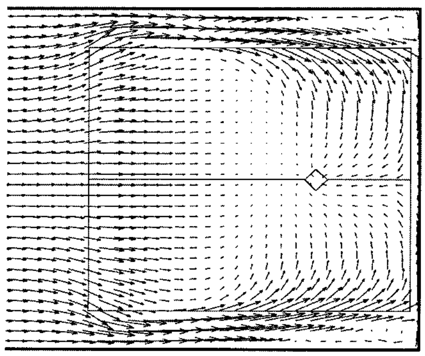


Fig. 4. Diagram of vapour velocity vectors.

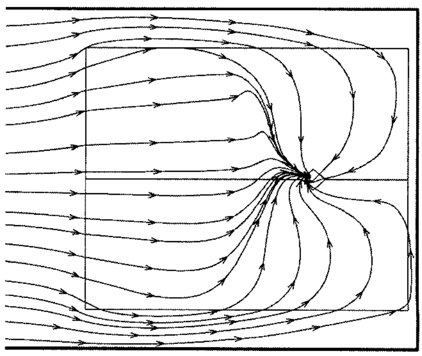
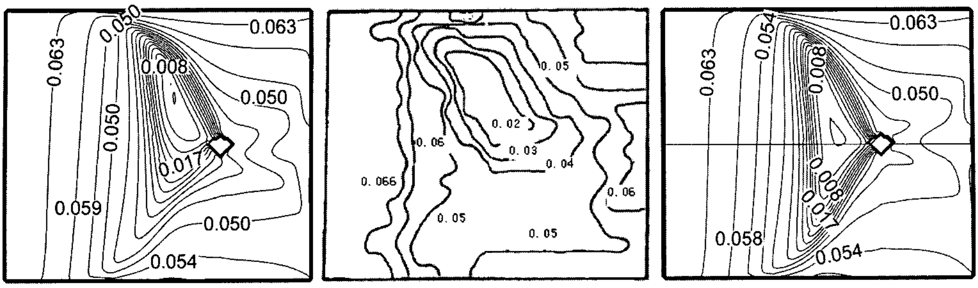
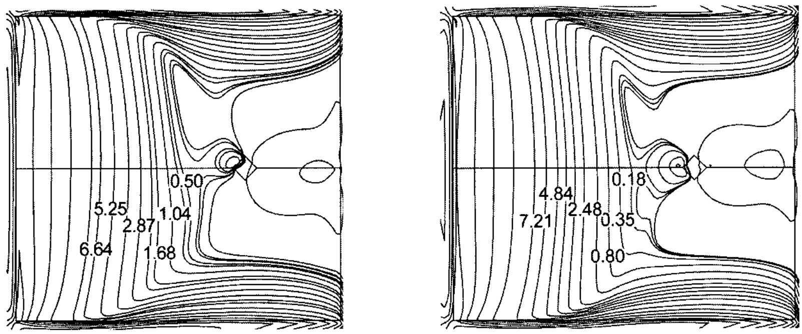


Fig. 5. Diagram of vapour streamlines.



(a) Computation Data (b) Experimental Data in [5],[6] (c) Single-phase Model Result

Fig. 6. Distribution of condensate rate (kg/m²).

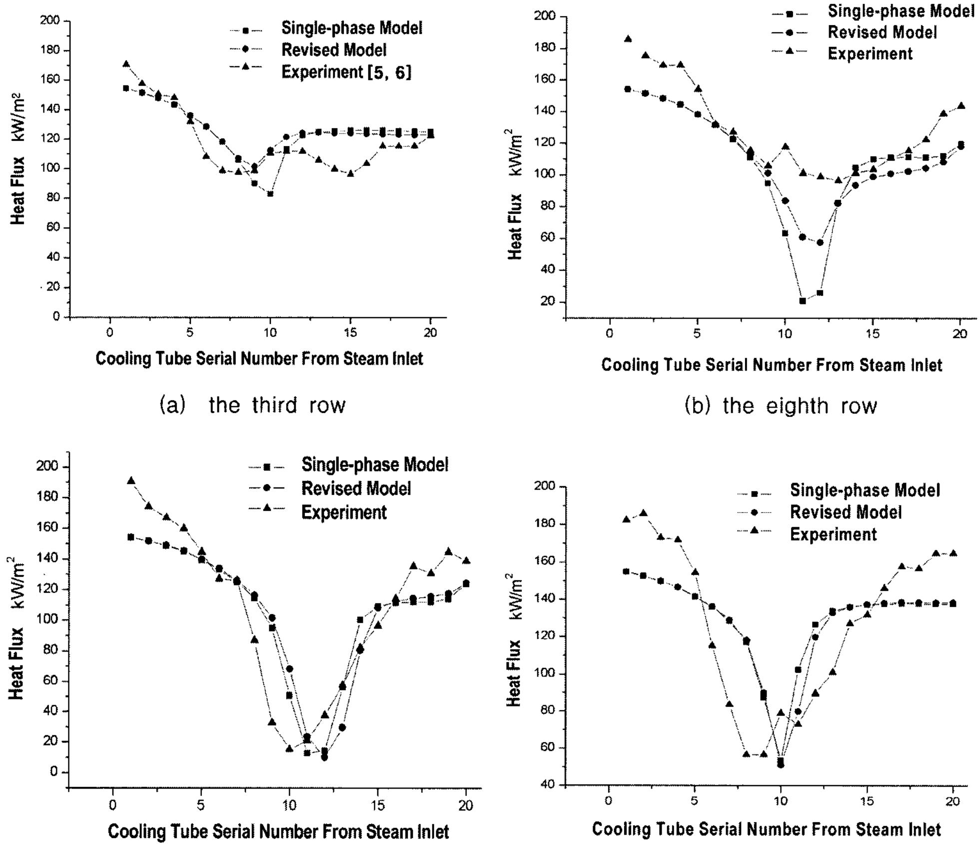


(a) model considering flow direction (b)single-phase model

Fig. 7. Comparison of distributions of horizontal components of vapour velocities (m/s).

seen that in opposition to the general thought that in the lower part of tube bank area the condensate should decrease since it is influenced by inundation, actually the least condensate rate occurs in the top part of tube bank area. The main reason may be explained as the heat transfer coefficient is lower due to the influence of inundation and therefore the variation of condensate rate in the lower part of tube bank area would be alleviative. At the periphery of the top part tube bank area, the variation gradient is greater due to higher heat transfer coefficient, the steam quantity and the vapour velocity decrease rapidly, so lower condensate rate would arise in the top middle part. On the other hand, from comparing Fig. 6(a) with 6(c) it can be seen that on the lower part of tube bank area the condensate rate which is obtained by the numerical model considering the influence of steam flow direction on flow resistance is greater than that obtained by the single-phase model. The reason can be explained as it is difficult for steam to flow into the under right part of tube bank area due to greater steam flow resistance for upward flow in this area, which would increase the horizontal flow from the under left part, and therefore the heat transfer coefficient will be increased. Fig. 7 shows comparison of distributions of horizontal components of vapour velocities which are respectively obtained by the numerical model considering the influence of steam flow direction on flow resistance and the single-phase model in which the influence of flow direction on resistance is not considered. From this figure it can be seen that at the lower middle part of tube bank area horizontal components of vapour volecities obtained by the former are greater than these obtained by the latter. It can be concluded that the reason that the least condensate rate occurs in the top part of tube bank area is there are great differences between the performances of inundation and flow resistance at the top part of tube bank area and those at the lower part.

Fig. 8 shows comparison of heat flux distributions which are respectively obtained by the revised model



(a) the third row (b) the eighth row

(c) the thirteenth row (d) the eighteenth row

Fig. 8. Heat flux distributions on tube rows from the bottom of tube bank.

considering the influence of steam flow direction on flow resistance, the experiment, and the single-phase model on four tube rows. From the figure it can be seen that for the thirteen row and the eighteen row the distribution of heat flux obtained by the revised model accords with that obtained by measured experimental data. For the tubes on the right of the third row, the heat flux obtained by revised model is a little greater than that obtained by measured experimental data. For several tubes in the middle of the eighth row, the heat flux obtained by revised model is less than that obtained by measured experimental data. The results of numerical simulation and experiment show that the area with the lowest heat flux lies in the top middle part of tube bank.

In addition, Fig. 8 shows comparision of heat flux distributions respectively obtained by the revised model and single-phase model. It can be seen that for the top of tube bank the results obtained by two models are all corresponding to that obtained by experimental data. For the middle lower part, the results obtained by single-phase model is less than those obtained by experimental data. The reason is that in this area upward flow dominates and the influence of flow direction on flow resistance and heat transfer is considerable. So the results obtained by the revised model are closer to experimental data than those obtained by the single-phase model.

Fig. 9 shows distribution of condensate rate resulted from the model considering the influence of flow direction on both flow resistance and inundation coefficient. There is no obvious difference between the distribution of condensate rate obtained by this model and that obtained by the model only considering the

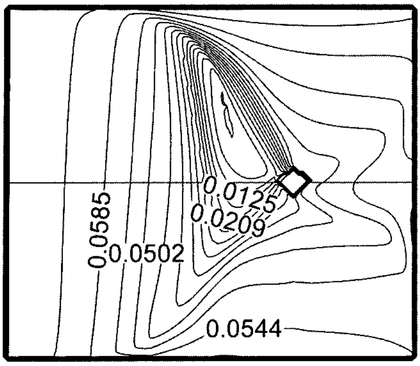


Fig. 9. Distribution of condensate rate resulted from the model considering the influence of flow direction (kg/m²).

influence of flow direction on flow resistance shown in Fig. 6(a). So it can be seen that the influence of flow direction on the performance of heat transfer is small.

5. Conclusion

The developed numerical simulation approach and program allowing for the influences of steam flow direction can successfully simulate main characteristics of steam flow and heat transfer of pannier-arrangement condenser.

Flow direction will influence the resistance performance in the lower tube bank area of the pannier-arrangement condenser, furthermore result in a transformation of steam flow and heat load distribution.

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Nomenclature

u : x velocity component ,m/s
v : y velocity component, m/s
w : total inundation rate, kg/s
c : condensation of the nth tube, kg/s
q : air concentration
Re : Renold number
Nu : Nuselt number
S_ϕ : source term

Greek Symbols

α_n : mean heat transfer coefficient, kg/m³
α_N : Nusselt heat transfer coefficient, kg/m³
ε : resistance coefficient
d : downward flow
D : sing phase
n : n^{th} tube
ϕ : independent variable
ρ : mixture density, kg/m³
Γ_ϕ : diffusion coefficient
N : Nuselt number
u : upward flow
v : kinemtic viscosity

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