The Computational Technique for Compressible Fluid Based on Steam Properties and Performance Improvements on Steam Turbines

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The computational technique for compressible fluid was developed. A finite volume method based on an approximate Riemann solver was adopted. The real gas effects of steam because of its widely changed state were taken into account using a steam table. The condensation by heat exchange with cooling water was formulated by two thermodynamic equations of state for pressure and energy. The spontaneous condensation in an adiabatic expansion was formulated according to non-equilibrium condensation theory. The computational technique was applied to the compressible fluid in the condenser combined with the turbine exhaust hood. Subcooling temperature from the annulus of the turbine blades to the condensate in the hot well was investigated. It was demonstrated that the subcooling temperature was reduced by improving the steam guide. Spontaneous condensation in the turbine stages was computed. The Wilson line which shows the maximum supercooling was predicted reliably.

1. Introduction

In steam turbines, steam pressure and temperature decrease greatly because of adiabatic expansion of steam. Expanded steam becomes wet in low pressure turbines, and finally it is condensed by heat exchange with cooling water in condensers. For computation of transonic and supersonic flows in these processes, it is necessary to consider real gas effects of steam, the state of which varies widely. Condensation especially plays an important role. For example, spontaneous condensation in the low pressure turbine affects the incident angle of flow through turbine blades. Condensation in the condenser dominates the turbine exhaust vacuum. Therefore, computation of the compressible fluid with condensation is important to improve the product design.

A computational technique for condensing flow was developed to improve the condenser design [1]. The condenser tube bundle was modeled as a porous media with drag and sink and advanced geometries to improve performance were investigated. However, the technique applicability is limited to condensers in which the flow velocity is relatively small, since the energy conservation law was approximated due to the large latent heat of the steam. Computation of compressible fluid in the condenser combined with the turbine exhaust hood is required for further improvement of the turbine exhaust vacuum.

In this paper, a computational technique for compressible fluid is developed for steam turbines by applying an approximate Riemann solver based on the theory of characteristics. The real gas effects of steam are taken into account using a steam table. The condensation is formulated by two equations of state for pressure and energy. The computational technique is applied to the compressible fluid in the condenser combined with the turbine exhaust hood.

2. Computational Techniques

2.1. Equations of State Condensation is characterized by decreases of both volume and energy. Under an isothermal state of steam (Fig.1), the pressure is held constant in the condensation process. On the basis of thermodynamics [2], the equation of state for pressure requires an equation of state for the internal energy given by

$$\left(\frac{\partial \varepsilon}{\partial V}\right)_T = T\left(\frac{\partial p}{\partial T}\right)_V - p \quad . \tag{1}$$

Consequently, internal energy is determined as Fig.1(b). Two equations of state for pressure and internal energy are transformed into one for pressure and temperature. Density and internal energy are chosen as independent variables of

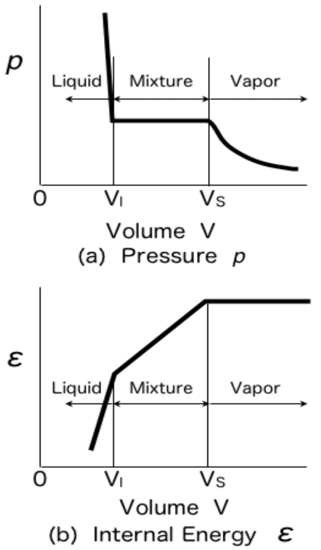

Fig. 1. Diagram of isothermal state.

transformed equations because these variables are conserved variables in an Euler equation.

Density in a computational grid is approximated with the averaged density of the mixture of gas and liquid although various forms of condensation are observed as in Fig. 2. A thermodynamic equilibrium between gas and liquid is assumed in a computational grid.

2.2 Euler Equation Two components of steam and non-condensable gas are assumed. Tube banks are approximated as porous media with drag and sink. The sink term represents condensation by heat exchange with cooling water. The Euler equation is expressed as followings.

$$Q_t + E_x + F_y + G_z = M \quad , \tag{2}$$

Here, $Q = (\rho_1, \rho_2, \rho u, \rho v, \rho w, e)$ is a conserved variable vector, where ρ_1 is the density of the mixture of gas and liquid of the steam component, ρ_2 is the density of non-condensable gas, $\rho = \rho_1 + \rho_2$ is the total density, and $\rho u, \rho v, \rho w$ are the momentum components in the directions of *x, y* and *z*. Energy per unit volume *e* is expressed as:

$$e = \rho(\varepsilon + (u^2 + v^2 + w^2)/2) \quad . \tag{3}$$

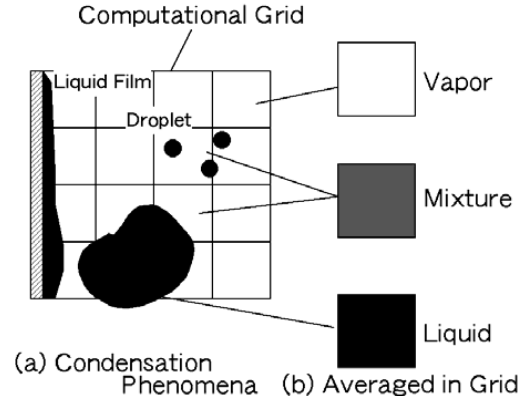

Fig. 2. Averaged density in computational grid.

E, F and *G* are flux vectors expressed as:

$$E = (\rho_1 u, \rho_2 u, p + \rho u^2, \rho uv, \rho uw, (e+p)u) \,,$$
$$F = (\rho_1 v, \rho_2 v, \rho vu, p + \rho v^2, \rho vw, (e+p)v) \,,$$
$$G = (\rho_1 w, \rho_2 w, \rho wu, \rho wv, p + \rho w^2, (e+p)w) \,.$$

Pressure p is obtained by summation of steam and non-condensable gas pressures assuming a small volume of liquid,

$$p = p_1 + p_2 \quad , \tag{4}$$

where p_1 is determined by the equation of state of steam, and p_2 is calculated as an ideal gas. In the same way, internal energy is expressed as:

$$\rho\varepsilon = \rho_1\varepsilon_1 + \rho_2\varepsilon_2 \tag{5}$$

and M describes the source term. Sink due to condensation by heat exchange and drag of tube banks are taken into consideration by

$$M = (-\dot{m}, 0, -\rho uq\xi_x/2, -\rho vq\xi_y/2, -\rho wq\xi_z/2, \\ -(\rho q/2)(\xi_x u^2 + \xi_y v^2 + \xi_z w^2) - \dot{m}\varepsilon) \,.$$

Here, $\dot{m}$ is condensation by heat exchange per unit volume, ξ_x, ξ_y, ξ_z are drag coefficients, and q is absolute velocity.

2.3 Heat Transfer Heat transfer between steam and cooling water is determined as follows.

$$q_f = \alpha(T - T_C) \quad , \tag{6}$$

Steam temperature T is given by the equation of state of steam. Cooling water temperature T_C is calculated by integrating heat flux from the inlet. The heat transfer coefficient α is evaluated using an empirical formula for condensation in tube banks[3].

2.4 Spatial Differencing In order to apply the computational technique to a complex geometry, an unstructured grid system is adopted. The computational volume is divided into tetra or prism cells. Variables are defined at vertexes of a cell according to a cell vertex scheme [4]. Equation (2) is integrated over a control volume at each vertex by subdividing the cell.

$$\frac{\partial}{\partial t}\int Q d\Omega + \int_{\partial\Omega} F_n dS = \int M d\Omega \quad , \qquad (7)$$

Here, $F_n = n_x E + n_y F + n_z G$ represents flux in the normal direction of the surface of the control volume.

A Riemann problem is assumed so that conserved variables Q take two values on the left and right sides of the surface of the control volume. Flux F_n is formulated by Roe's flux difference splitting [5].

$$F_n = \frac{1}{2}[F_n(Q_R) + F_n(Q_L) - |A|\Delta Q] \quad , \qquad (9)$$

Glaister's formulation[6] is applied to evaluate $|A|\Delta Q$. Second order difference scheme is used for high resolution.

2.5 Boundary Conditions At a wall surface, a slip condition is used because of an inviscid flow model. The normal velocity component is zero, and the tangential velocity components are extrapolated from the interior. The pressure is extrapolated by using the normal momentum equation.

At an inflow boundary, the inflow rate and the wetness of steam are fixed to give the heat load on the condenser. The pressure is extrapolated from the interior because the inflow is subsonic. The saturated density at the given pressure is used to calculate the velocity components normal to the inflow boundary under the conditions of the fixed inflow rate and the same stagnation pressure.

2.6 Spontaneous Condensation Next, non-equilibrium condensation in the low-pressure stage of steam turbines is modeled [7]. The nucleation rate by the modified Volmer-Frenkel model and the growth rate of the droplet's radius by the Young-Gyarmathy model [8] are coupled to solve the wetness mass fraction. The equation of

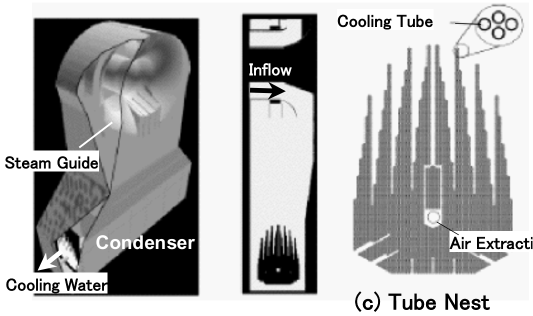

Fig. 3. Geometry of condenser combined with turbine exhaust hood.

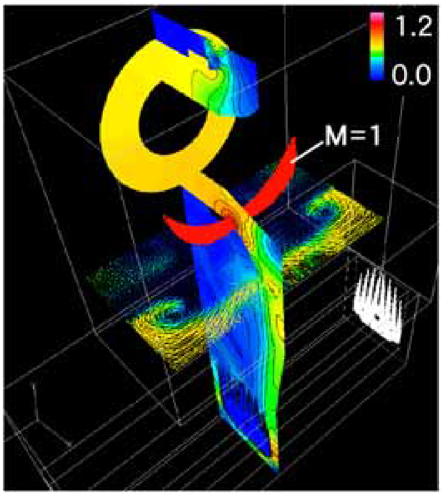

Fig.4. Distribution of Mach number(-).

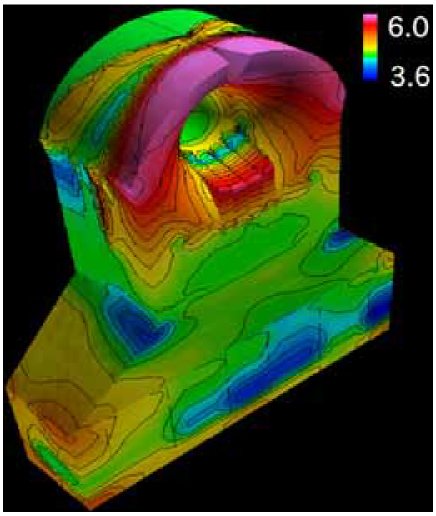

Fig.5. Distribution of pressure(kPa).

state for pressure is modified by the wetness mass fraction.

3. Results and Discussion

3.1. Condenser Combined with Turbine Exhaust Hood A compressible fluid is computed in the condenser combined with the turbine exhaust hood as shown in Fig. 3. The turbine exhaust hood was developed for a 52 inch turbine blade. The super balanced down flow type condenser [9] developed by Hitachi Ltd. is applied in order to decrease both pressure loss and stagnation of non-condensable gas in the tube nest.

Steam introduced in the lateral direction is diffused by a steam guide which decreases its velocity. The lateral flow is turned to the vertical direction and condensed in the tube nest. Finally, the mixture of remaining steam and non-condensable gas is extracted from the center of the tube nest.

Distribution of Mach number is shown in Fig.4. A supersonic region arises at the root of the steam guide. The flow through the annulus of the turbine blades expands in the radial direction because of obstruction by the the wall which is located on opposite side of the annulus. This movement accelerates the flow at the root of the steam guide and the supersonic region is formed. Pressure distribution on the casing wall is shown in Fig.5. A large pressure difference between the turbine exhaust hood and condenser is observed.

Figure 6 compares the distribution through the root of the steam guide at various cooling water temperatures T_C. It should be noted that the distribution changes significantly for a small change of cooling water temperature. At a low temperature of cooling water, the pressure drop by adiabatic expansion is recovered rapidly by a shock wave. Robustness is required to design the turbine exhaust hood.

The data plotted in Fig.7 demonstrate the effect of the advanced steam guide. Subcooling temperature, defined as the temperature decrease from the annulus of the turbine blades to the condensate in the hot well, is compared at various cooling water temperatures. Reduction of the subcooling temperature improves the heat rate of the steam turbine system.

3.2 Spontaneous Condensation in the Turbine Stages The last three stages of the low-pressure section are calculated. The wetness mass fraction contours and the distribution of the degree of supercooling,

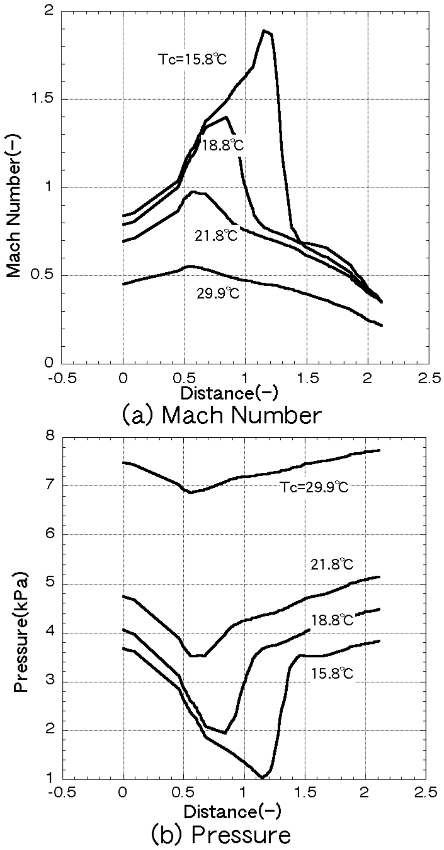

Fig. 6. Distributions through steam guide.

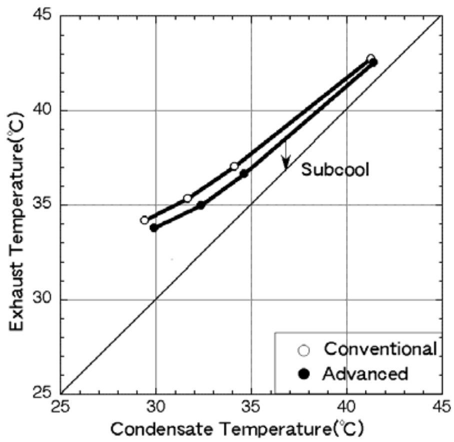

Fig. 7. Subcooling temperature decrease obtained by the advanced steam guide.

defined as the difference from the actual temperature to the corresponding saturation temperature of the steam pressure, are shown in Fig.8. The steam at the inlet is superheated and it crosses the saturation line at the nozzle throat of the first stage. However, condensation from steam to droplets does not occur because of the surface tension of the small droplets and the high speed flow. Therefore the steam is in a non-equilibrium state and condensation occurs suddenly when the degree of supercooling is about 30 K. It is non-equilibrium condensation with an entropy increase. This maximum supercooling line is called the Wilson line. Prediction of the Wilson line is important, because its position affects stage loads and water droplet nuclei are formed at the line. Then the droplet distributions are strongly influenced by the position of the Wilson line.

4. Conclusions

The computational technique for the compressible fluid was developed. A finite volume method based on an approximate Riemann solver was adopted. The real gas effects of steam because of its widely changed states were taken into account using a steam table. Condensation by heat exchange with cooling water was formulated by two thermodynamic equations of state for pressure and energy. The spontaneous condensation in adiabatic expansion was formulated according to the non-equilibrium condensation theory. The computational technique was applied to the compressible fluid in the condenser combined with the turbine exhaust hood. Subcooling temperature from the annulus of the turbine blades to the condensate in the hot well was investigated. It was demonstrated that the subcooling temperature was reduced by by using an advanced steam guide. Spontaneous condensation in the turbine stages was computed. The Wilson line which shows maximum supercooling was predicted reliably. Further improvement of the steam turbine system will be pursued by the presented technique.

References and Notes

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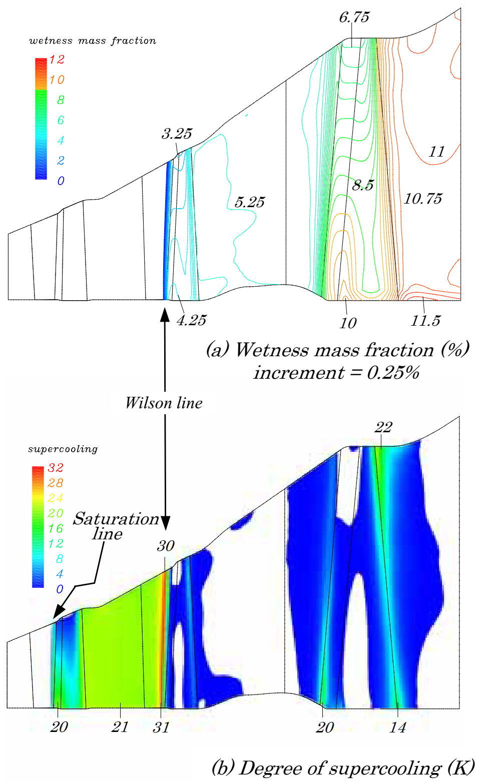

Fig.8. Wetness mass fraction contours and distribution of the degree of supercooling for the last three low pressure stages.