Numerical analysis of non-equilibrium steam condensing flows in various Laval nozzles and cascades

Chang Hyun Kim\textsuperscript{a}, Jae Hyeon Park\textsuperscript{a}, Dong Il Kim\textsuperscript{b} and Je Hyun Baek\textsuperscript{a}

\textsuperscript{a}Department of Mechanical Engineering, Pohang University of Science and Technology, Pohang, Republic of Korea; \textsuperscript{b}Thermal & Fluid Research Team, Doosan Heavy Industries & Construction, Yongin, Republic of Korea

ABSTRACT

When steam is used in fluid machinery, phase transition can occur that affects not only the flow fields but also machine performance. Therefore, to achieve an accurate prediction of steam condensing flow using computational fluid dynamics (CFD), phase-transition phenomena should be considered and a non-equilibrium wet-steam model is required. Such a model is implemented in this study using the in-house code T-Flow, and the flow fields – including phase-transition phenomena – in various Laval nozzles are examined. The results for multi-phase flows can be obtained in relatively short time by using mixture assumption and an inner-iteration method. The calculated results reflect the characteristics of the condensing flows well and are comparable with those obtained experimentally. Also, it was found that the superheating level of incoming steam can explain the tendency of condensation in the nozzles considered in a simple way. In addition, steam condensing flows in the blade cascades were simulated. As a result, the predicted blade loading agreed well with the experimental data and the superheating level at inlet was responsible for the condensation trend not only in the nozzles but also in the cascades. In future work, the characteristics of steam condensing flow in a steam turbine where complex flows and phase transition occur can be investigated using the presented model.

ARTICLE HISTORY

Received 26 April 2016
Accepted 28 November 2016

KEYWORDS

CFD; Moore nozzle; Moses and Stein nozzle; non-equilibrium wet-steam model; steam condensing flow; White cascade

1. Introduction

When steam is used as a working fluid in fluid machinery, phase transition can occur in the internal flow field. In addition, the temperature and pressure can locally increase due to the latent heat released from the vapor-to-liquid phase transition (condensation shock). For example, droplets produced in the low-pressure stages of a steam turbine affect not only the internal flow fields but also the turbine structures, thus degrading turbine performance in a variety of ways, including efficiency. Thus, to simulate steam flow precisely with computational fluid dynamics (CFD), phase-transition phenomena need to be considered. In real cases in which flow expansion is drastic, phase transition does not occur at the equilibrium. Therefore, to consider non-equilibrium condensation in simulating steam condensing flow, a non-equilibrium wet-steam model is needed.

Several studies have been conducted to simulate steam condensing flow in Laval nozzles, cascades and steam turbines. Dykas and Wroblewski (2011) studied wet-steam flow in arc Laval nozzles using single- and two-fluid models, comparing the two. Zhu et al. (2012) simulated steam condensing flow in Barschdorff nozzle and Bakhter cascade using quadrature method of moments. Halama (2012) and Halama and Fort (2013) studied inviscid, laminar flows in the cascade and transonic flow in nozzle and turbine by considering non-equilibrium phase transition. Chandler, White, and Young (2014) analyzed unsteady low pressure (LP) turbine flows with non-equilibrium wet-steam calculations. Grubel et al. (2014) and Schatz et al. (2014) conducted experiments and simulations with LP turbines, and compared them in detail. However, wet-steam flow remains still difficult to predict accurately, because it includes non-equilibrium phase-transition phenomena of which the physics are not fully understood. Also, high computational cost is one of the difficulties in simulating multi-phase flow. Therefore, existing models should be refined and further studies are required to obtain results that agree well with experimental data at a lower computational cost.

In this study, a non-equilibrium wet-steam model was implemented using the self-developed in-house code T-Flow, and steam condensing flows in a Moore nozzle B, a Moses and Stein nozzle and a White cascade were simulated. The results of the multi-phase flows can be obtained in a relatively short time using mixture...
assumption and an inner-iteration method. The distributions of the flow and the wet-steam parameters were compared with experimental data, and the characteristics of the steam condensing flows were investigated. In addition, the superheating level of incoming steam was used to predict the tendency of condensation in the nozzles and cascades.

2. Theoretical backgrounds

2.1. T-Flow

T-Flow was first developed in the 1990s to calculate 3D flow fields, and has been evolving since. It is designed to predict the internal flow in turbomachinery and has been used and validated in various studies on tip leakage flow, clocking effects, and stator/rotor interaction in centrifugal, axial compressors (Baek & Choi, 2007; Choi, Park, & Baek, 2005; Park, Chung, & Baek, 2003) and axial turbines (Park & Baek, 2002; Park, Choi, & Baek, 2006). T-Flow uses compressible Reynolds-averaged Navier–Stokes (RANS) equations as governing equations:

\[ \frac{\partial Q}{\partial t} + \frac{\partial E}{\partial x} + \frac{\partial F}{\partial y} + \frac{\partial G}{\partial z} = \frac{\partial E_v}{\partial x} + \frac{\partial F_v}{\partial y} + \frac{\partial G_v}{\partial z} \tag{1} \]

where

\[ Q = [\rho, \rho u, \rho v, \rho w, e]^T, \]
\[ E = [\rho u, \rho u^2 + p, \rho uv, \rho uw, u(e + p)]^T, \]
\[ F = [\rho v, \rho w, \rho v^2 + p, \rho vw, v(e + p)]^T, \]
\[ G = [\rho w, \rho uw, \rho vw, \rho w^2 + p, w(e + p)]^T, \]
\[ E_v = [0, \tau_{xx}, \tau_{xy}, \tau_{xz}, u\tau_{xx} + v\tau_{xy} + w\tau_{xz} - q_s]^T, \]
\[ F_v = [0, \tau_{yx}, \tau_{yy}, \tau_{yz}, u\tau_{yx} + v\tau_{yy} + w\tau_{yz} - q_p]^T, \]
\[ G_v = [0, \tau_{zx}, \tau_{zy}, \tau_{zz}, u\tau_{zx} + v\tau_{zy} + w\tau_{zz} - q_s]^T. \]

Here, the vector \( Q \) consists of conservative variables, \( E, F, \) and \( G \) are inviscid flux vectors, \( E_v, F_v, \) and \( G_v \) are viscous flux vectors, \( x, y, \) and \( z \) are the three directions in Cartesian coordinates, \( u, v, \) and \( w \) are the velocity components in each direction, \( T \) is the transpose of a matrix, \( t \) is the time, \( e \) is the total internal energy, \( q \) is the heat flux, and \( \tau \) is the Reynolds stress. The governing equations are discretized using a finite-volume method in space and a Euler implicit time-marching scheme is used. Flux difference splitting (as suggested by Roe, 1981) and flux vector splitting (as suggested by Van Leer, 1982) along with an upwind scheme and Monotonic Upstream-Centered Scheme for Conservation Laws are used for inviscid flux terms. The central difference scheme is used for viscous flux terms. A local time-stepping method based on the Courant-Friedrichs-Lewy number and a message-passing interface (MPI) are used for rapid calculation. The governing equations and discretization methods used in T-Flow are described in detail in Choi (2008).

2.2. The non-equilibrium wet-steam model

Condensation is a phase-transition phenomenon in steam flow that frequently occurs in non-equilibrium states when drastic expansion takes place. In this case, the boundary of the transition from vapor to liquid is not a saturation line but rather a Wilson line at which supercooled vapor starts to condensate. To consider this kind of unusual phase transition in calculations, a non-equilibrium wet-steam model is required. The wet-steam model is composed of two equations (ANSYS Fluent) related with wet-steam parameters, so that they are implemented on T-Flow. By using this model, droplet generation from non-equilibrium phase transition can be demonstrated:

\[ \frac{\partial (\rho \beta)}{\partial t} + \frac{\partial (\rho \beta u)}{\partial x} + \frac{\partial (\rho \beta v)}{\partial y} + \frac{\partial (\rho \beta w)}{\partial z} = \Gamma, \tag{2} \]
\[ \frac{\partial (\rho \eta)}{\partial t} + \frac{\partial (\rho \eta u)}{\partial x} + \frac{\partial (\rho \eta v)}{\partial y} + \frac{\partial (\rho \eta w)}{\partial z} = \rho I. \tag{3} \]

where \( \beta \) is the liquid mass fraction (wetness) and \( \eta \) is the number of droplets per unit volume (droplet number). The source term of Equation (2) is expressed in Equation (4), which is related to newly-created droplets and the growth of extant droplets. The radius of droplets newly created by nucleation is represented as the critical radius \( r_s \), which is determined by the pressure and surface tension of water (Equation (5)). The droplet growth rate in Equation (6) is determined by the temperature difference between the phases. The source term of Equation (3) is expressed in Equation (7), which is the number of newly-created droplets per unit time and unit volume (nucleation rate); \( \theta \) in the nucleation rate (Equation (8)) is a non-isothermal correction factor. The droplet size can be determined from the calculated wetness and droplet number values (Equation (9)). (ANSYS Fluent):

\[ \Gamma = \frac{4}{3} \pi \rho_d r_s^3 \left( 4 \pi \rho_d \eta \pi^2 \frac{dr}{dt} \right) \tag{4} \]
\[ r_s = \frac{2 \sigma}{\rho_d RT_c \ln S} \left( \frac{P}{P_{sat}(T_c)} \right), \tag{5} \]
\[ \frac{dr}{dt} = \frac{P}{h_v \rho_d \sqrt{2 \pi RT_c}} \left( \gamma + 1 \right) \frac{1}{2 \gamma} C_p(T_d - T_c), \tag{6} \]
\[ I = \frac{q_c}{(1 + \theta)} \left( \frac{\rho_d}{\rho_i} \right)^{\frac{2 \sigma}{\rho_d}} \sqrt{\frac{2 \pi K_b T_c}{M_i^2 \sigma}} \exp \left( -\frac{4 \pi r_s^2 \sigma}{3 K_b T_c} \right), \tag{7} \]
\[ \theta = \frac{2 (\gamma - 1)}{\gamma + 1} \left( \frac{h_v}{RT_c} \right) \left( \frac{h_w}{RT_c} - 0.5 \right). \tag{8} \]
respectively (Gerber, 2008). The amount of inter-phase transfer in mass and momentum between the phases are assumed to be zero, because of no external gain or loss; if there is a loss in vapor, the equivalent gain in liquid occurs. In contrast, the latent heat released from the condensation should work as a source for representing phase transition, so the net amount of inter-phase transfer in energy is not zero and can be represented with Equation (10) – i.e., the product of the mass transfer rate and the latent heat. This source term is essential for simulating steam condensing flow in T-Flow, which uses ideal gas law as the equation of state:

\[ \frac{3\alpha d}{r} \cdot [q_c + q_d] = \dot{m} h_{iw}. \]  

(10)

where \( \frac{3\alpha d}{r} \) is the surface area density of the droplets, and \( q_c \) and \( q_d \) mean that the heat flux has transferred to the vapor and liquid phases from the released latent heat, respectively (Gerber, 2008). The amount of heat flux to each phase is determined by droplet size as follows:

(1) \( 2r \leq 0.1 \) nm:

\[ q_c = 0, \quad q_d = 0. \]

(2) \( 0.1 \text{ nm} < 2r \leq 1 \mu \text{m}:

\[ q_c = \frac{k_c}{2r} N u_c (T_{sat} - T_c), \quad \text{where} \]

\[ N u_c = \frac{2}{1 + 3.18 K r}, \quad q_d = 0. \]

(3) \( 2r > 1 \mu \text{m}:

\[ q_c = \frac{k_c}{2r} N u_c (T_{sat} - T_c), \quad \text{where} \]

\[ N u_c = 2 + 0.6 \text{Re}_d^{1/2} \text{Pr}_{d}^{1/3}, \]

\[ q_d = \frac{k_d}{2r} N u_d (T_{sat} - T_d), \quad \text{where} \ N u_d = 6. \]

The latent heat and conductivity of vapor and liquid used in the above equations were calculated from user-defined functions of temperature by fitting the original values of International Association for the Properties of Water and Steam-97 (Wagner & Kretzschmar, 2007). To get \( q_c, q_d \) and the droplet growth rate, the temperatures of the vapor and liquid phases (\( T_c \) and \( T_d \)) are required. The calculations with mixture assumption can just give the temperature of a mixture. As in the research of Gerber (2008) and the publication of Moore and Sieverding (1976), at a droplet diameter of \( \leq 1 \mu \text{m} \) (usually a reasonable assumption), the temperature of the liquid phase can be assumed from the saturation temperature and droplet radius (Equation (11)). In the case of low wetness, the temperature of the vapor is not significantly different from the temperature of the mixture, so the temperatures of both phases can be determined individually:

\[ T_d = T_{sat} - (T_{sat} - T_c) \frac{r_s}{r}. \]  

(11)

After calculating conservative variables by solving Equation (1) in matrix form, the density, velocity and internal total energy of a mixture can be obtained. In T-Flow, the pressure is used to derive from the total energy and fluid velocity based on the ideal gas law. The pressure can change due to the formation of water droplets, so the approximation of pressure in wet-steam flows is redefined using Equation (12) (Mei & Guha, 2006). Additionally, the accuracy of calculated flow variables can be increased by changing the equation of state via the Virial equation or IAPWS-97, so further studies on the effects of the equations of states used are needed. The equation is as follows:

\[ p = \left( \gamma - 1 \right) \left( 1 - \beta \right) \frac{1}{1 + \beta (\gamma - 1)} \left[ e - \frac{1}{2} \rho V^2 + \rho \beta h_{iw} \right]. \]  

(12)

To reduce the calculation time, the inner-iteration method was also implemented. When RANS equations (Equation (1)) and wet-steam equations (Equations (2) and (3)) are solved simultaneously in a single matrix, the accuracy is slightly increased, but it is a time-consuming job because a small time-step size is required for wet-steam calculation. However, when the equations are solved individually as in turbulence models (the tracking method) with different time-step sizes, the computational cost and time for multi-phase calculations is significantly reduced. Due to the difference in time-step sizes, wet-steam calculation must be iterated to match up with the time marching of the RANS calculations. As a result, the computational time for 1000 iterations in the case of a Laval nozzle was considerably reduced from 190 s to 32 s by adopting an inner-iteration method.

### 3. CFD modeling and numerical results

To validate the non-equilibrium wet-steam model in T-Flow, the multi-phase flow in a Moore nozzle C (Moore,
Walters, Crane, & Davidson, 1973) was simulated in a previous study (Kim et al., 2015). The results show that droplets were produced by non-equilibrium phase transition and that condensation shock took place in the middle of the nozzle. The estimated pressure and droplet size were consistent with the experimental data. The wet-steam model used in this study is improved by adopting real water properties (e.g., viscosity and conductivity of water) in the wet-steam calculations, and is expected to predict the multi-phase flow more accurately.

In the following sections, steam condensing flows in various Laval nozzles and cascades are studied and their results are compared with experimental data. The computational grids and experimental data of a Moore nozzle B and Moses and Stein nozzles are taken from International Wet Stem Modelling Project (Starzmann et al., 2016).

3.1. Moore nozzle B (Moore et al., 1973)

In the case of the Moore nozzle B, not only do the distributions of the pressure and droplet radius help with understanding the physics of the steam condensing process, but the contours of the flow fields are also useful.

The Moore nozzle B in Figure 1 slightly differs from the nozzle C in Kim et al. (2015); it has different throat and outlet sizes, and has a slightly higher expansion rate (2300 s\(^{-1}\)). The computational grid consists of 120,000 (400 \(\times\) 101 \(\times\) 3) nodes in a 3D structured format (hexahedral). The maximum \(y^+\) value with the grid is 0.54, which is fine enough to simulate the nozzle flow. This grid size (\(y^+ < 1\)) is required to obtain reliable results near the wall boundary without using the wall model (Salim & Cheah, 2009) and can provide grid-independent results. Therefore, similar grid sizes were chosen in the following cases of this study. For the boundary conditions, the total pressure and total temperature conditions were imposed at the inlet in the normal direction, and the static pressure condition was used at the outlet, while no-slip and adiabatic conditions were used at the walls and a symmetric condition was used on the planes in the thickness direction. The inlet does not provide any incoming droplets and the details of the used boundary conditions are given in Table 1. A one-equation Spalart–Allmaras turbulence model for compressible flows (Deck, Duveau, d’Espiney, & Guillen, 2002) was used to simulate turbulence flow, and the result of the single-phase flow was used as the initial value of the multi-phase calculation for numerical stability.

In the general cases without phase transition, the velocity increased and the pressure decreased consistently throughout the nozzle. However, it was revealed that the velocity was lower in the multi-phase case than in the single-phase case, and the velocity even decreased locally in the downstream (Kim et al., 2015). Similar phenomena occurred and are captured in Figure 2(a). The pressure and temperature increased locally in the middle of the nozzle. Thus, it can be assumed that supercooled vapor reached the Wilson point after the nozzle throat and influenced the main flow field. The place where the pressure and temperature increased is located right after the nucleation zone (Figure 2(b)); this increase in pressure and temperature due to droplet generation is called condensation shock. After condensation, droplet size and wetness started to increase. The difference between the vapor and liquid temperature is expected to affect the droplet growth. At the end of the nozzle, about 5% of total water mass was liquid.

The supercooling level (\(T_{\text{sat}} - T_c\); Figure 3) is an important parameter in non-equilibrium phase transitions. The supercooling level reached about 30 K at the middle of the nozzle (after the nozzle throat) but decreased after the droplet formation started. The pressure and droplet radius distributions along the nozzle centerline (Figure 4(a)) can be compared with those of the experiments in Moore et al. (1973). In the single-phase simulation, as mentioned before, the pressure probably decreased consistently throughout the nozzle. However, the pressure obtained in the experiments and calculations increased locally after the nozzle throat due

Table 1. Boundary conditions of the Moore nozzle B.

| Moore nozzle B | Value |
|----------------|-------|
| Inlet total pressure (\(p_o\), kPa) | 25.00 |
| Inlet total temperature (\(T_o\), K) | 358.10 |
| Wetness (inlet) | 0.00 |
| Droplet number (inlet) | 0.00 |

Figure 1. The computational grid for the Moore nozzle B.
Figure 2. Distributions in the Moore nozzle B for (a) velocity, pressure and temperature and (b) wetness, droplet number, droplet radius and nucleation rate.

Figure 3. Supercooling level distribution of the Moore nozzle B.

to the formation of droplets. Compared to the experimental results, the multi-phase calculation with T-Flow predicted a higher pressure, but the pressure curve shapes are similar. The overestimation of downstream pressure in the calculation might come from overestimated intensive condensation shock, which can be modified by adopting another equation of state or surface tension correction factor in the future.

Different from the result of nozzle C (Kim et al., 2015), T-Flow underestimated the droplet size along the nozzle at around half the size of those in the experiments. The droplet size is strongly affected by the equation of state in the wet-steam model, as in Halama and Fort (2013) and Grube et al. (2014). However, the droplet radius has been underestimated in many numerical studies (e.g., Dykas & Wroblewski, 2012; Yang & Shen, 2009) and measurement uncertainty should be taken into account due to its small size. For wet-steam parameters (nucleation rate, wetness and supercooling level) there is no available experiment data. However, their distributions along the nozzle centerline can explain the relationships among the wet-steam parameters.

In Figure 4(b), a tremendous amount of nucleation takes place at \( x = 0.095 \) m and wetness starts to increase rapidly. This starting point of increasing wetness coincides with the starting point of increasing pressure and the point where the droplets are shown to grow rapidly in Figure 4(a). The supercooling level in Figure 4(c) has a maximum value about 33.6 K at \( x = 0.095 \) m where the maximum value of the nucleation rate is located. After nucleation, the supercooling level declines to almost zero, which means that the generation of water droplets alleviated the non-equilibrium state of the steam.

To compare the results quantitatively, the pressure, droplet radius, nucleation rate, wetness and supercooling level distributions along the nozzle centerline are shown for the other nozzles. The contours of pressure and temperature are not included because the physics of phase
transition are not significantly different from those of the Moore nozzle.

3.2. Moses and Stein nozzle (cases 252 and 257; Moses & Stein, 1978)

The Moses and Stein nozzle (Figure 5) is a relatively small nozzle compared to the Moore nozzle. It has a large inlet but a small outlet, and its expansion ratio is much higher (9000 s⁻¹) than that of Moore nozzle. This nozzle has been used to validate wet-steam code in several studies (Grubel et al., 2014; Zhu et al., 2012) and has been tested at higher pressures than Moore nozzles. Numerous experiments were run on Moses and Stein nozzles by Moses and Stein (1978), and cases 252 and 257 were simulated in the present study and then compared to the experimental results of Moses and Stein. Case 257 exhibits a slightly higher inlet total pressure and temperature than case 252, but both cases were tested in the same nozzle geometry. Therefore, cases 252 and 257 share a computational grid composed of 95,000 (350 × 91 × 3) nodes (hexahedral). The maximum $y^+$ values with the grid are 0.18 and 0.20 for case 252 and case 257, respectively. Similar settings to those used for the Moore nozzle were used for the boundary conditions (Table 2) and a Spalart–Allmaras turbulence model for compressible flows was adopted (Deck et al., 2002). Single-phase calculations were performed before the wet-steam calculation for initialization.

Figure 6 shows the results for case 252. It can be seen from Figure 6(a) that T-Flow predicted the pressure increase (condensation shock) well. The location of the pressure increase coincides with that in the experiment with a little difference in intensity. The predicted droplets are smaller than the measured ones. The nucleation rate and wetness distribution shown in Figure 6(b) explain the location of the condensation shock, as in the case of Moore nozzle. However, different from the case of Moore nozzle, incoming steam was superheated with a negative supercooling level and the supercooling level reached its maximum value in the nucleated region (42.3 K at $x = 0.036$ m; Figure 6(c)).

Figure 7 shows the results for case 257. It can be seen from Figure 7(a) that T-Flow captured the pressure increase (condensation shock), but the location and intensity are different from those in experiment ($\Delta x = 0.005$ m). T-Flow predicted the pressure increase earlier and with a higher intensity. However, the pressure in the downstream is consistent with the results of the experiment when comparing with the other nozzle.

**Table 2.** Boundary conditions of the Moses and Stein nozzle (252/257).

|                | Case 252 | Case 257 |
|----------------|----------|----------|
| Inlet total pressure ($p_0$, kPa) | 40.05    | 67.66    |
| Inlet total temperature ($T_0$, K) | 374.30   | 376.70   |
| Wetness (inlet)               | 0.00     | 0.00     |
| Droplet number (inlet)        | 0.00     | 0.00     |

**Figure 5.** The computational grid for the Moses and Stein nozzle.
cases. The droplet size was also underestimated for case 257. The nucleation and wetness are distributed similarly, as in the other cases (Figure 7(b)). The supercooling level also reached maximum value in the nucleated region (35.7 K at $x = 0.019$ m; Figure 7(c)). However, the intensity of the nucleation and wetness at the outlet differ from those of case 252. In case 257, the pressure increased more by condensation shock and the steam contained higher wetness at the outlet. On the other hand, intensive nucleation took place in case 252. Therefore, it is clarified that the condensation shock, nucleation rate and wetness are closely related but cannot be interpreted in a simple way.

To compare the nozzles simulated here, some guidelines are needed due to the differences in nozzle geometry. Moore nozzles and Moses and Stein nozzles have totally different shapes and sizes; the convergence and divergence of the passage are drastic in the Moses and Stein nozzle. The nozzles not only have different scales but also different expansion rates based on the throat area. However, the expansion rates based on the pressure change can be derived as in Equation (13) (Leyzerovich, 2005) and are compared along the nozzles in Figure 8:

$$
\dot{p} = \frac{1}{p} \frac{\partial p}{\partial t} = \frac{1}{p} \frac{\partial p}{\partial x} \frac{\partial x}{\partial t} = \frac{u}{p} \frac{\partial p}{\partial x}.
$$

(13)

This would be a meaningful approach if the locations of the condensation shock could be predicted from the pressure change in single-phase calculations. Therefore, the results of both the multi-phase and single-phase calculations are shown. In the Moore nozzle, the location of the pressure increase in the multi-phase calculation coincides with the maximum point of the expansion rate in the single-phase calculation. However, this assumption cannot be used universally, because the Moses and Stein nozzles have no maximum point of expansion rate in the single-phase calculations, near the location of the condensation shock in the multi-phase calculations.

Another guideline that can be compared regardless of nozzle shape and size is the superheating level (which is opposite to the supercooling level). As shown in Figures 4, 6 and 7, the supercooling level distribution can be obtained after the multi-phase calculations. Though the profile of the supercooling level is hard to estimate from only the boundary conditions, the characteristics
of the condensation in the nozzle might be altered by the inlet condition. Therefore, it is suggested that the superheating level can be computed based on the condition of the incoming steam (Table 3) in order to infer the tendencies of the condensation in the nozzles. Case 257 in the Moses and Stein nozzle has the lowest superheating level based on the inlet condition; this means that the incoming steam was close to the saturation line and then the fluid was prone to becoming condensed. The superheating level then increases from the Moore nozzle B to the Moses and Stein nozzle in case 252. A reasonable assumption is that a lower superheating level is connected to the active generation of droplets. As a result, the predicted wetness values in the calculations show the same trend as the superheating levels; case 257 demonstrates a higher liquid mass fraction at the outlet (6.5%) than with the other nozzles. In addition, the maximum supercooling level in case 257 is the lower than that in case 252. The intensity of the condensation shock is also greater and the location of the pressure increase is closer to the inlet in case 257 than in case 252, which has a higher superheating level at the inlet. However, this suggestion that the superheating level of incoming steam can explain the tendency of condensation regardless of nozzle shape should be evaluated further by testing more Laval nozzles.

### 3.3. White cascade (White, 1992)

Not only the Laval nozzles but also blade cascades have been widely tested to validate or evaluate the wet-steam calculations. Blade cascades have more a complicated geometry than nozzles, and calculating flows in cascades can be said of preliminary work to simulate flows in steam turbines. The calculation domain and blade shape are shown in Figure 9. The computational grid was generated using ICEM CFD v14.0 in structured format (hexahedral) and consists of 650,000 nodes. The maximum $y+$
Table 4. Boundary conditions of the White cascade (H1/L1/W1).

| Moses and Stein nozzle | H1  | L1  | W1  |
|------------------------|-----|-----|-----|
| Inlet total pressure ($p_o$, kPa) | 43.70 | 40.30 | 41.90 |
| Inlet total temperature ($T_o$, K) | 378.00 | 354.00 | 350.00 |
| Wetness (inlet) | 0.00 | 0.00 | 0.00 |
| Droplet number (inlet) | 0.00 | 0.00 | 0.00 |

values with the grid are 0.41, 0.42, and 0.45 for the high-superheat condition (H1), low-superheat condition (L1), and wet-inlet condition (W1). Settings similar to those used in the nozzle cases were again used for the boundary conditions (Table 4). However, to simulate the flow in a single passage, a periodic condition was used on the planes in the circumferential direction. The three cases (H1, L1, and W1) were tested by changing the inlet and outlet conditions on the same computational grid. Single-phase calculations were used as the initial values of the multi-phase calculations. Again, a Spalart–Allmaras turbulence model for compressible flows was adopted (Deck et al., 2002).

In White (1992), the pressure on the blade surface and the wetness at the traverse plane were measured, but only the surface pressure was used in the comparison due to the unspecified traverse plane location. In the present study, to examine the flow physics in the cascade and make comparisons with that of the nozzle flows, the distributions of velocity, pressure, wetness and supercooling level in the H1 case are given in Figure 10.

The flow patterns in the cascade are similar to those in the nozzles; the flow accelerates and the pressure decreases as the flow moves downstream. The mach number reaches unity at the throat region, and the pressure seems to locally increase after the throat, where the supercooling level reaches its maximum value. The nucleation starts at that location and the droplets grow so that the wetness varies along the downstream of the cascade. However, the wetness approaches zero near the walls, as in the nozzle cases. These characteristics can also be observed in Yamamoto (2000) and Gerber (2008). Similar trends were achieved in the L1 and W1 cases, but there are some differences in the pressure profiles.

The pressure profiles are plotted in Figure 11. The simulated results are in good agreement with the experimental ones on the pressure surfaces because condensation rarely occurs in the high-pressure region with low velocity. On the other hand, condensation occurred on the suction surfaces and affected the pressure profiles. An increase in pressure (condensation shock) was captured in all test cases, but the trends are different. In the H1 case, the superheated steam was imposed at the inlet, so the condensation shock was located far downstream. The locations where the pressure increased are slightly different between the experiment and the calculation, with the latter predicting the location a little way further downstream than the experiment. Also, the calculation underestimated the pressure in the downstream with a small discrepancy. In the L1 case, the location of condensation shock was closer to the inlet than in the H1 case due to the low superheated condition. The simulation captured the starting point of the pressure increase but underestimated the intensity. The pressure in the downstream was also underestimated. In the case of W1, the condensation shock occurred at the closest location to the inlet due to the wet conditions. The calculation predicted the location a little bit upstream compared to the experimental result, but the pressure profile in the downstream is consistent with that of the experiment.

To evaluate the guidelines suggested in the nozzle section, the superheating level based on inlet conditions and the maximum values of the nucleation rate, the

Figure 10. Distributions in the White cascade (H1) for (a) velocity, (b) pressure, (c) wetness, and (d) supercooling level.
Figure 11. Pressure distributions along the blade surfaces for the White cascade for (a) H1, (b) L1, and (c) W1.

Table 5. Superheating level based on the inlet conditions and the maximum values of the nucleation rate, the wetness, and the supercooling level (H1/L1/W1).

| Moses and Stein nozzle | H1   | L1   | W1   |
|------------------------|------|------|------|
| Superheating level (K) | 26.85| 4.81 | -0.13|
| Nucleation rate (×10²²/m³s) | 50.00 | 6.00 | 0.60 |
| Wetness (%)            | 4.10 | 7.60 | 8.20 |
| Supercooling level (K) | 39.50| 35.70| 33.00|

wetness, and the supercooling level in the cascades are listed in Table 5. As has already been shown in the case of the nozzles, a higher superheating level involves more supercooling so that more intensive nucleation occurs. In addition, the lower superheating level connected to high wetness because the condition of the incoming steam is close to the saturation line. Therefore, the superheating level at the inlet explains the tendencies of the nucleation, supercooling and developed wetness in the cases considered in this study.

Through the calculation of the steam condensing flow in a Moore nozzle, a Moses and Stein nozzle and a White cascade, the characteristics of wet-steam flow (e.g., condensation shocks) were captured quite well. In comparison with the experiments, there are some discrepancies in the pressure and the droplet radius (max. Δ(p/p₀) <8% in the L1 case). However, the predicted locations and intensities of the condensation shock in the calculations are not significantly different from those of the experiments. Discrepancies between the experiments and the calculations might come from wet-steam modeling, the equation of state used or some other factors. The predicted droplet sizes are always smaller than in the experiments, so the properties used and the measured data should be checked and improved for better accuracy.

T-Flow is a density-based code meaning that IAPWS-97 – which can provide accurate steam properties across wide ranges including in metastable regions – cannot be applied in a simple way without using a pre-conditioning technique. Resultingly, the multi-phase calculations of the nozzles and cascades in this study can be carried out in a relatively short time using a mixture assumption with an ideal gas law and an inner-iteration method. However, it is certain that the ideal gas law is not sufficient to guarantee high accuracy or reliability in relation to wet-steam predictions. Therefore, these modifications of adopting a new equation of state – especially IAPWS-97 with pre-conditioned code – are required to improve the accuracy in the long term, although the computational cost might be considerably increased as a result.

In addition, a number of researchers have attempted to construct a universal wet-steam model which can be used without changing the empirical factors. For example, the surface tension provided by IAPWS is valid for a flat surface, though the surface tension used in a wet-steam model should be considered in relation to spherical surfaces (Fakhari, 2006). Therefore, changing or classifying the parameters that are used is also an effective way of improving the accuracy. Furthermore, considering poly-dispersed droplets (Bohn, Surken, & Kreitmeier, 2003) and hetero-geneous nucleation might also be good approach for obtaining comparable data.

As mentioned before, the flow in the nozzles and cascades were usually simulated to prepare the calculations of more complex geometries – particularly the last stages of steam turbines. Not only the steam condensation but also rotating effects should be considered in steam turbines, so further studies are required and the models will need modifying along the lines of those presented in this study in order to adopt this model in general cases and retain high accuracy.

4. Conclusion
To consider the phase transition of steam condensing flow, a non-equilibrium wet-steam model was implemented in T-flow and the flow of various Laval nozzles
and cascades was analyzed. The results of multi-phase flows can be obtained in a relatively short time using mixture assumption and an inner-iteration method. The numerical results were compared with the experimental data and the following conclusions were obtained:

1. In a Moore nozzle B, the non-equilibrium phase transition occurs with characteristic condensation shock and shows quite good agreement with the experiment in terms of the pressure and droplet radius. In the cases of the Moses and Stein nozzles, the steam condensing flows were predicted with reasonable accuracy using the implemented wet-steam model.

2. The superheating level of the incoming steam can explain the tendency of condensation in the nozzles considered. Supplying low superheated steam involves high wetness in the flow field. On the other hand, high superheated steam at the inlet is connected to a high supercooling level before the condensation so that more intensive nucleation occurs.

3. In the case of the White cascade, the trends of condensation are similar to those in nozzles. The condensation shock was captured on the suction surfaces and the wetness varied along the downstream. The pressure distribution along the blade alters with changes in the inlet conditions and matched well with the experimental data. In addition, the trends of wetness, supercooling level and nucleation can be explained by the condition of the steam at inlet, as in the nozzle cases.

Discrepancies in the pressure and the droplet radius existed (max. $\Delta (p/p_o) <8\%$), but they are not significantly different in terms of equation of state used and measurement uncertainty. The implemented model predicted the experimental results with reasonable and consistent accuracy and is expected to bring further accurate results in various cases including steam turbines when it is further improved in the future.

**Nomenclature**

- $\alpha$: Volume fraction
- $\beta$: Water mass faction, wetness
- $\gamma$: Specific heat ratio ($C_p/C_v$)
- $\Gamma$: Mass generation rate
- $\eta$: Number density of droplet
- $\theta$: Non-isothermal correction factor
- $\pi$: Circular constant (pi)
- $\rho$: Density (w/o subscript, mixture)
- $\sigma$: Surface tension
- $C$: Speed of sound
- $h_{lv}$: Enthalpy of vaporization
- $I$: Nucleation rate
- $K_b$: Boltzmann constant
- $Kn$: Knudsen number
- $k$: Conductivity
- $M_m$: Water molecular mass
- $Nu$: Nusselt number
- $p$: Pressure (w/o subscript, static)
- $Pr$: Prandtl number
- $R$: Gas constant
- $r$: Droplet radius
- $Re$: Reynolds number
- $T$: Temperature (w/o subscript, mixture)
- $V$: Fluid velocity

**Subscript**

- $c$: Continuous phase, steam
- $d$: Dispersed phase, water
- $p$: Isobaric
- $sat$: Saturated
- $o$: Inlet, total
- $v$: Isochoric
- $^*$: Critical

**Acknowledgements**

The computational grids and experimental data of the Moore nozzle B and Moses and Stein nozzles were provided through taking part in the University of Cambridge’s International Wet Steam Modelling Project (IWSMP; http://wet-steam-project.eng.cam.ac.uk).

**Disclosure statement**

No potential conflict of interest was reported by the authors.

**Funding**

This work was supported by Doosan Heavy Industries and Construction.

**References**

ANSYS Fluent (14.0) [Computer software]. Location: ANSYS Inc.

Baek, J.-H., & Choi, M.-S. (2007). Effects of the inlet boundary layer on the flow in an axial compressor. *The KSME Journal of Fluid Machinery*, 10(5), 83–88. doi:10.5293/KFMA.2007.10.5.083

Bohn, D. E., Surken, N., & Kreitmeier, F. (2003). Nucleation phenomena in a multi-stage low pressure steam turbine. *Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy*, 217(4), 453–460. doi:10.1243/095765003322315513

Chandler, K., White, A., & Young, J. (2014). Non-equilibrium wet-steam calculations of unsteady low-pressure turbine
flows. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 228(2), 143–152. doi:10.1177/0957650913511802

Choi, M.-S. (2008). Effects of inlet boundary layer thickness on the loss characteristics and the rotating stall in an axial compressor (Doctoral dissertation). POSTECH, Republic of Korea.

Choi, M.-S., Park, J.-Y., & Baek, J.-Y. (2005). Effects of the inlet boundary layer thickness on the flow in an axial compressor (1) – Hub corner stall and Tip leakage flow –. Transactions of the Korean Society of Mechanical Engineers B, 29(8), 948–955. doi:10.3795/KSME.B.2005.29.8.948

Deck, S., Duveau, P., d’Espiney, P., & Guillen, P. (2002). Development and application of Spalart–Allmaras one equation turbulence model to three-dimensional supersonic complex configurations. Aerospace Science and Technology, 6(3), 171–183. doi:10.1016/S1270-9638(02)01148-3

Dykas, S., & Wroblewski, W. (2011). Single- and two-fluid models for steam condensing flow modeling. International Journal of Multiphase Flow, 37(9), 1245–1253. doi:10.1016/j.ijmultiphaseflow.2011.05.008

Dykas, S., & Wroblewski, W. (2012). Numerical modelling of steam condensing flow in low and high-pressure nozzles. International Journal of Heat and Mass Transfer, 55(21), 6191–6199. doi:10.1016/j.ijheatmasstransfer.2012.06.041

Fakhari, K. (2006, January). Development of a two-phase eulerian/lagrangian algorithm for condensing steam flow. 44th AIAA aerospace sciences meeting and exhibit. Reno, NV: AIAA. doi:10.2514/6.2006-597

Gerber, A. G. (2008). Inhomogeneous multifluid model for prediction of nonequilibrium phase transition and droplet dynamics. Journal of Fluids Engineering, 130(3), 031402, 1–11. doi:10.1115/1.2844580

Grubel, M., Starzmann, J., Schatza, M., Eberle, T., Vogt, D. M., & Sieverding, F. (2014). Two-phase flow modeling and measurements in low-pressure turbines – part 1: Numerical validation of wet steam models and turbine modelling. Proceedings of ASME Turbo Expo 2014, GT2014-25244, 1–13. ASME, Dusseldorf, Germany. doi:10.1115/GT2014-25244

Halama, J. (2012). Transonic flow of wet steam – numerical simulation. Acta Polytechnica, 52(6), 124–130.

Halama, J., & Fort, J. (2013). Numerical simulation of transonic flow of wet steam in nozzles and turbines. Computing, 95(1), 303–318. doi:10.1007/s00607-013-0292-6

Kim, C.-H., Park, J.-H., Ko, D.-G., Kim, D.-I., Kim, Y.-S., & Baek, J.-H. (2015). Analysis on steam condensing flow using non-equilibrium wet-steam model. Journal of Computational Fluids Engineering, 20(3), 1–7. doi:10.6112/kscfe.2015.20.3.01

Leyzerovich, A. (2005). Wet-steam turbines for nuclear power plants. PennWell Books.

Mei, Y., & Guha, A. (2006). Modification of the upwind schemes for the computation of condensing two-phase flows. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 220(7), 809–814. doi:10.1243/09576509JPE146

Moore, M. J., & Sieverding, C. H. (1976). Two-phase steam flow in turbines and separators: Theory, instrumentation, engineering. Hemisphere Publishing Corporation.

Moore, M. J., Walters, P. T., Crane, R. I., & Davidson, B. J. (1973). Predicting the fog-drop size in wet-steam turbines. Wet Steam, 4, 101–109.

Moses, C. A., & Stein, G. D. (1978). On the growth of steam droplets formed in a laval nozzle using both static pressure and light scattering measurements. Journal of Fluids Engineering, 100(3), 311–322. doi:10.1115/1.3448672

Park, J.-Y., & Baek, J.-H. (2002). Three-dimensional flow analysis under stator/rotor interaction in one-stage turbine using the parallel programming method. Proceedings of the KSME 2002 Spring Annual Meeting, 1500–1505. KSME, Pyeongchang, Republic of Korea.

Park, J.-I., Choi, M.-S., & Baek, J.-H. (2006). Numerical study on the clocking effect in a 1.5 stage axial turbine. Journal of Computational Fluids Engineering, 11(4), 1–8.

Park, J.-Y., Chung, H.-T., & Baek, J.-H. (2003). Tip leakage flow on the transonic compressor rotor. Transactions of the Korean Society of Mechanical Engineers B, 27(1), 84–94. doi:10.3795/KSME.B.2003.27.10.084

Roe, P. L. (1981). Approximate riemann solvers, parameter vectors, and difference schemes. Journal of Computational Physics, 43(2), 357–372. doi:10.1016/0021-9991(81)90128-5

Salim, S. M., & Cheah, S. C. (2009). Wall y+ strategy for dealing with wall-bounded turbulent flows. Proceedings of the International MultiConference of Engineers and Computer Scientists, 1–6. IAENG, Hong Kong.

Schatz, M., Eberle, T., Grubel, M., Starzmann, J., Vogt, D. M., & Surken, N. (2014). Two-phase flow modeling and measurements in low-pressure turbines – part 2: Turbine wetness measurement and comparison to cfd-predictions. Proceedings of ASME turbo expo 2014, GT2014-25245, 1–12. ASME, Dusseldorf, Germany. doi:10.1115/GT2014-25245

Starzmann, J., Hughes, F. R., White, A. J., Halama, J., Hric, V., Kolovratnik, M., . . . Li, L. (2016, September). Results of the international wet steam modelling project. Wet Steam Conference, Czech Technical University in Prague, Prague, Czech Republic.

Van Leer, B. (1982). Flux vector splitting for the euler equations. Lecture Notes in Physics, 170, 501–512.

Wagner, W., & Kretzschmar, H. J. (2007). International steam tables-properties of water and steam based on the industrial formulation IAPWS-IF97: Tables, algorithms, diagrams, and CD-ROM electronic steam tables—all of the equations of IAPWS-IF97 including a complete set of supplementary backward equations for fast calculations of heat cycles, boilers, and steam turbines. Springer Science & Business Media.

White, A. J. (1992). Condensation in steam turbine cascades (Doctoral dissertation). University of Cambridge, UK.

Yamamoto, S. (2000, September). Numerical study of nonequilibrium condensation in unsteady transonic viscous flow problem. Barcelona: ECCOMAS.

Yang, Y., & Shen, S. (2009). Numerical simulation on nonequilibrium spontaneous condensation in supersonic steam flow. International Communications in Heat and Mass Transfer, 36(9), 902–907. doi:10.1016/j.icheatmasstransfer.2009.06.001

Zhu, X., Lin, Z., Yuan, X., Tejima, T., Niizeki, Y., & Shibukawa, N. (2012). Non-equilibrium condensing flow modeling in nozzle and turbine cascade. International Journal of Gas Turbine, Propulsion and Power Systems, 4(3), 9–16.