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EVAPORATION BEHAVIOR OF WATER DROPLETS FLOWING THROUGH AXIAL FLOW COMPRESSORS

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ABSTRACT

Evaporation of a water droplet flowing through an axial flow compressor is studied theoretically and experimentally. A droplet evaporation model applicable to a rapid change of ambient air temperature is proposed. Change in droplet diameter, evaporation rate and resultant temperature decrease of the ambient air are calculated with the effect of heat absorbed by the water in the droplet taken into consideration. The model has been verified by fundamental experiment using a high temperature wind tunnel. Cumulative evaporation profile along axial position is estimated by the model for the compressor of a heavy duty gas turbine and design considerations with respect to diameter of the atomized water droplet are made for practical application.

1. INTRODUCTION

Moisture air gas turbine (MAT) cycle has been recently put into commercial operations to augment power and efficiency of gas turbines especially for solution tactics of peak electricity demand in summer. Taking advantage of large evaporative latent heat of water, the cycle effectively cools intake as well as compressed air of the compressor by injecting finely atomized normal temperature water droplets at inlet to the compressor. Its principle and experimental verification were reported [4-6]. Walsh et al. reported a technology acquisition program for inlet fog boosting of gas turbines [7].

Cooling of the intake air (inlet air cooling effect) increases mass flow of the air introduced to the compressor, which results in the increase of gas turbine power output. Whereas, with the evaporation of the water in droplets flowing through the compressor, like coincident realization of both intercooling and steam injection, specific power may be also increased. Power boosting and compressor performance characteristics may be greatly affected by the water evaporation rate. Hence, the performance evaluation of the MAT cycle requires detailed analysis of the temperature profile along the air flow path resulting from the evaporation. There are few references available in this context. Hill derived an analytical expression for wet compression [1]. Cranfield university evaluated gas turbine performance with water injection and showed efficiency of the compressor is improved [3]. Successful cooling of compressed air by the evaporation of water droplets has been confirmed by a 15 MW axial flow load compressor [5]. Horlock indicated an effective increase in the specific heat within the compressor due to the latent heat requirement in evaporation [15]. Utamura et al. calculated ideal compression work of the air water mixture with phase change [4] and obtained numerical results that the work required to drive the compressor is reduced with the amount of water injected. These studies, however, assumed isentropic change of state, regardless of limited speed of heat and mass transfer rates, which might more or less lead to overestimation of the evaporation rate. Tsuchiya et al. calculated the evaporation time of water droplets in the air stream with constant temperature [2]. Evaporation models proposed so far mostly assumed thermal equilibrium in the droplet, i.e. all heat absorbed by water is discharged to the ambient air in an instance with evaporative latent heat [10-13]. Previous works with respect to water fogging to the compressor has been well reviewed by the references [7,8].

Focusing upon the evaporation rate in the axial flow compressor, the present study aims to establish an evaporation model of droplets flowing through the axial flow compressor where the water droplet is exposed to a rapid temperature change of the air in both space and time. Typically, during 10ms of the residence time of the air in the compressor, air temperature rise is more than 300 K where steady state mass and heat transfer may no longer be assumed. In this paper, the analytical model that treats transient evaporation phenomenon as a quasi steady state is proposed and applied to industrial gas turbines. Experiment to verify the model is also described.

2. THEORY

2.1 Physical Model

Following assumptions are made in the analysis:

①the water droplet is spherical shape during evaporation,

②air steam mixture is ideal gas,

③the air near the droplet surface is saturated at the temperature of the droplet surface,

④the water temperature in the droplet is spatially homogeneous,

⑤vapor pressure is negligibly small to a total pressure in the compressor,

⑥velocity difference between the droplet and air is sufficiently large,

⑦the temperature of the dry gas is constant within each stage,

⑧the temperature of steam evaporated instantly increases up to that of gas mixture, and

⑨evaporation rate is proportional to the total surface area of the droplets and the difference between the vapor pressure near droplet surface and ambient one.

Since the total amount of water droplets evaporated in the compressor is very small in the MAT cycle, i.e. the order of 1-2 % at most to air mass ratio, the change in the diameter of water droplets will essentially be independent of the total amount evaporated. According to the assumption ⑤, the change of water mass of a single droplet due to evaporation was considered in the model. Then, the total amount evaporated would be obtained by the change in mass of the droplet times the number of droplets. It is known the water droplet with the diameter of the order of 10μ m has spherical shape due to large surface tension effect.

CFD studies conducted by Utamura et al. [6] indicated that droplet 10-20μ m in diameter would follow the air flow path, running behind the air by about 40 m/s. Under this relative velocity, according to Tsuchiya [2], time required for a 15 μm droplet to complete its evaporation (evaporation time), ranges from 0.1-1 ms under the temperature differences of 1-10 K whereas a typical time of flight of air through a stage in industrial compressors is 1 ms. As above two values are of the same order, unlike previous works [2,11], thermal equilibrium of a water droplet can no longer be assumed. Thus, time dependent analysis is needed. Fig. 1 shows outline of the analytical model. With unknown variables r and T_w, following equations hold:

mass conservation of water :

$$dr/dt = - w/(\rho S) \tag{2-1}$$

energy conservation with heat absorption of the droplet taken into consideration :

$$Cp_w \, m \, dT_w/dt = Q - L w \tag{2-2}$$

where,

Q is heat conduction from the ambient air to the water in the droplet,

$$Q = 4 \pi r k G(T_{am} - T_w) \tag{2-3}$$

w is evaporation rate of the representative water droplet,

$$w = 4 \pi r a F(M/R)(P(T_w)/T_w - P_{am}/T_{am}) \tag{2-4}$$

in which F and G are forced convection correction factors [10,13] to the evaporation in stationary air without gravity,

F for mass transfer,

$$F = 1 + 0.275 R_{ed}^{0.5} S_c^{1/3} \tag{2-5}$$

G for heat transfer,

$$G = 1 + 0.275 R_{ed}^{0.5} P_r^{1/3} \tag{2-6}$$

Ambient vapor pressure P_{am} is calculated by using absolute humidity χ and gas phase total pressure. For many gases, the value of S_c is almost equal to that of P_r. So, essentially G = F , i.e. Lewis number(L_e) is unity. In our topics, the right hand sides of Eqs. (2-5) and (2-6) are calculated to be about 2-3 in the range of air temperatures 330-600 K and relative velocity of air to the water droplet is 40 m/s. From Eq.(2-2), in thermal equilibrium Q = Lw holds from which droplet temperature T_∞ is obtained. It is interesting to note that T_∞ is independent of droplet size provided L_e =1.

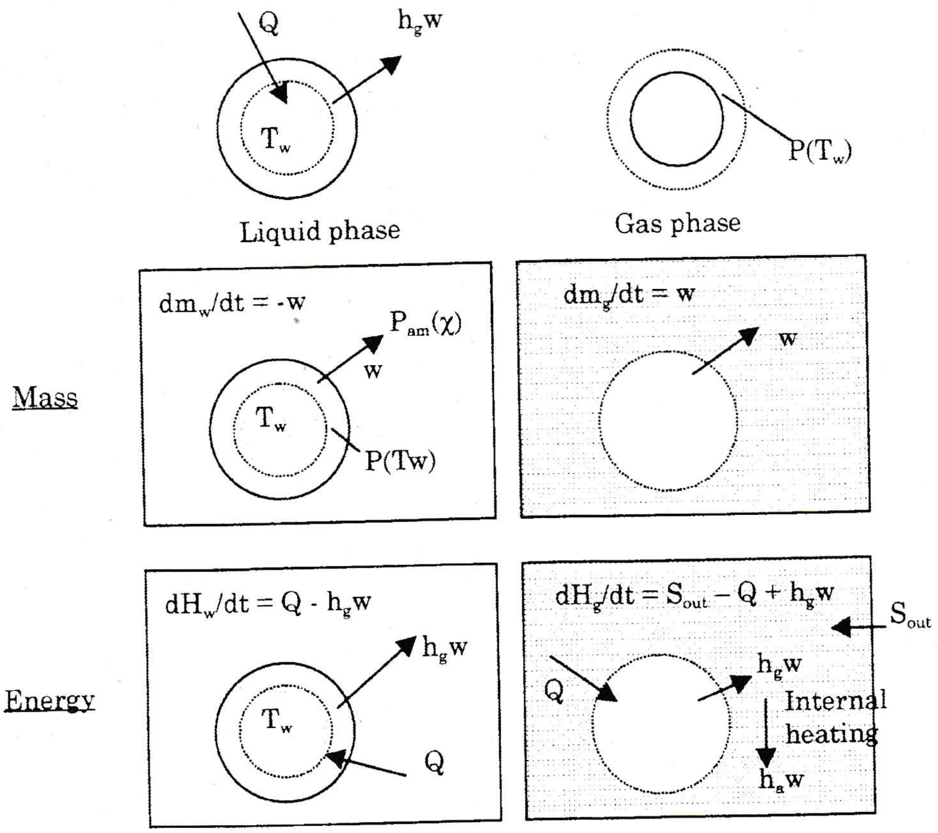

Fig.1 Analytical Model

2.2 Solution Procedure

The decrease of the dry air temperature due to water evaporation through the compressor may be expressed formally by

$$\Delta T_d = N \int (Q + C_{ps}(T_{am} - T_{wav})w)\, dt / (C_{pa} W_a + C_{ps} W_s) \tag{2-7}$$

which can be converted into a discretized form in terms of stage

$$\Delta T_d = \Sigma (N Q_{av} \Delta t_j / (C_{ps} W_a) + N w \Delta t_j (T_{am} - T_{wav}) / W_a) / (C_{pa} / C_{ps} + \chi) \tag{2-7'}$$

in which stage averaged heat conduction Q_{av} has to be expressed in a concrete form.

Because of very short residence time of droplet in the stage, we have to consider thermal non-equilibrium effect. Since Eqs. (2-1) and (2-2) are first order differential equations, we may write solutions of stage exit water temperature T_{we} and stage averaged one T_{wav} in an approximate form as

$$T_{we} = T_{\infty} - (T_{\infty} - T_{wi}) \exp(-\Delta t/\tau) \tag{2-8}$$

$$T_{wav} \equiv \int T_w\, dt / \Delta t = T_{\infty} + (T_{wi} - T_{we})(\tau / \Delta t) \tag{2-9}$$

where

$$\tau = \rho C_{pa} r^2 / (3G) / (k - aLMP(T_{\infty}) / (RT_{\infty}^2)) \tag{2.10}$$

with approximations $(TT_{\infty}) \Rightarrow T_{\infty}^2$ and $P(T) \Rightarrow P(T_{\infty})$ being made. According to Eq.(2-10), τ varies from 0.5 to 0.35 ms corresponding to T_{am} variations from 300 to 600 K . These are of the same order of the droplet residence time 1 ms in a stage. For example, if we take τ to be 0.5 ms, then from Eq.(2-8) droplet temperature rise in a stage would be 15 % smaller than that in the case of assuming thermal equilibrium. Therefore, it is understood that the existence of the left hand side of Eq.(2-2) is important. As a result, temperature rise of the water in droplets would be delayed in the compressor when we take the effect of heat absorption by the water droplet into consideration.

Then, from Eqs.(2-3) and (2-8), we can derive an expression for Q_{av} ,

$$Q_{av} = 4\pi r kG[(T_{am} - T_{\infty}) + (T_{\infty} - T_{wi})(1 - \exp(-\Delta t/\tau)(\tau/\Delta t))] \tag{2-11}$$

Putting Eq.(2-11) into Eq.(2-7)', we can calculate ΔT_d . T_{we} in Eq.(2-8) is regarded as T_{wi} of the next stage downstream and calculation proceeds stage by stage.

Integration of Eq.(2-1) with Eq.(2-4) into consideration gives the change of droplet radius squared within a stage

$$ri^2 - re^2 = 2\, aMG \Delta t / (\rho R) / (P_{am}/T_{am} - P(T_{wav})/T_{wav}) \tag{2-12}$$

Calculation procedures follow. Given stage temperatures of dry air flowing through compressor stages, the amount of temperature drop of the gas and change in droplet diameter were calculated consecutively stage by stage from the entry to the stage where evaporation completes.

2.3 Calculation Results

Taking an axial flow compressor of a 115 MW class stationary gas turbine as an example given in the reference [4], evaporation behavior of a water droplet was calculated from the inlet to the exit of the compressor with ambient conditions 308 K and 60 %R.H. The air at the inlet was assumed to be saturated (301K) by inlet air cooling effect.

A droplet velocity relative to the working fluid flow was assumed to be 40 m/s based on CFD

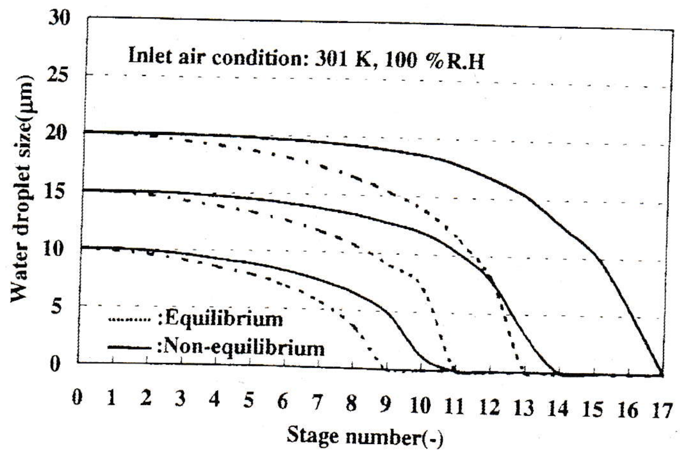

Fig.2 Droplet Diameter Profile

results [6]. The amount of liquid water introduced to compressor was 1 % to air mass ratio. Fig. 2 shows calculation results of droplet diameter change along axis of the compressor with initial droplet diameters 10, 15 and 20 μm as parameter. Calculations were made for cases with and without the assumption of thermal equilibrium. Evaporation delayed in larger droplet, which is marked more in thermal non-equilibrium model. The temperatures of the water droplet as well as gas are given in Fig.3. It is worthy of note that in smaller droplet gas temperature drop at mid-stage is larger, but smaller at exit of the compressor. The larger the initial droplet diameter , the higher grows the liquid temperature at the completion of evaporation. Fig.4 compares equilibrium model with that of non–equilibrium. The equilibrium model gave gas temperature almost independent of droplet size and higher gas temperature than in that of non-equilibrium. It is because heat is removed more in the non-equilibrium state of change where the droplet works as a heat sink. Fig. 5 shows temperature drop through the compressor as function of the initial droplet diameter. It is interesting to note there is an optimum droplet diameter around 22 μm that cools the air most effectively. All droplets below 22 μm were found to complete their evaporations within the compressor. Heat of dry air is removed by ① latent heat of evaporation, ② steam heating and ③ heat absorption by the water droplet. As the initial droplet size becomes larger, effects of ① and ② become smaller but larger the effect of ③ to the contrary.

Stage polytropic exponents were evaluated based on the above calculations.

$$n_j = \ln(P_{j+1}/P_j)/\ln(T_j P_{j+1}/(P_j T_j + 1)) \qquad (2\text{-}13)$$

The polytropic exponent had lower values while evaporation proceeds. It should be noted that the value ranges below the value of super heated steam 1.3 even though steam concentration is very low. This low value would tend to shift compression process from adiabatic to isothermal and result in less work to drive the compressor. After completion of evaporation, the polytropic exponent increases again to reach around that of dry air 1.4.

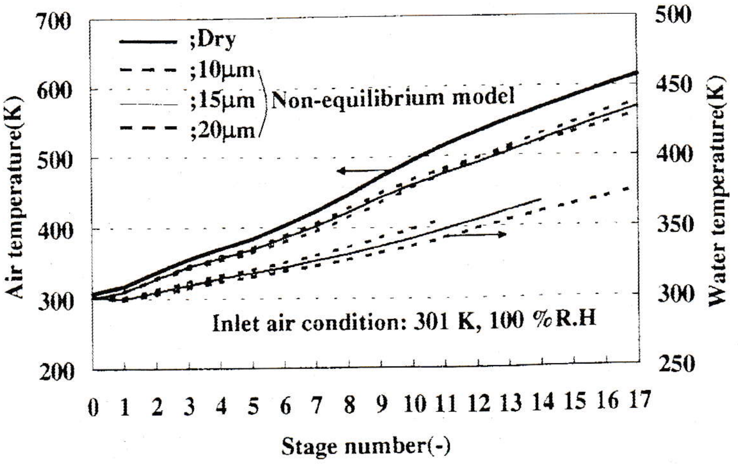

Fig.3 Temperatures of Water Droplet and Gas

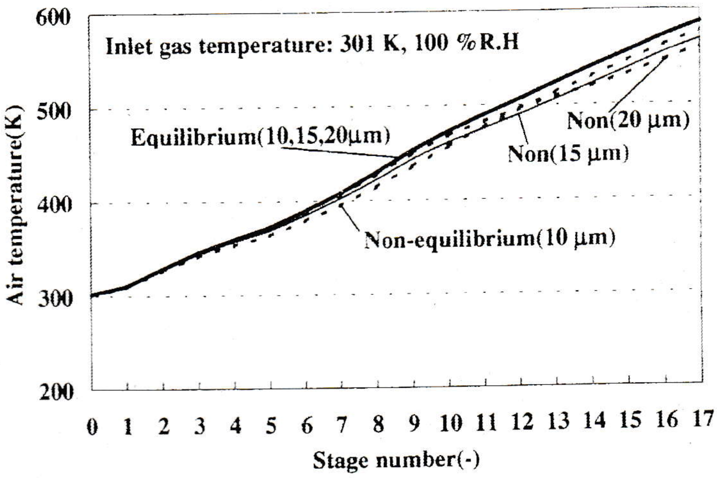

Fig.4 Air Temperature Profile

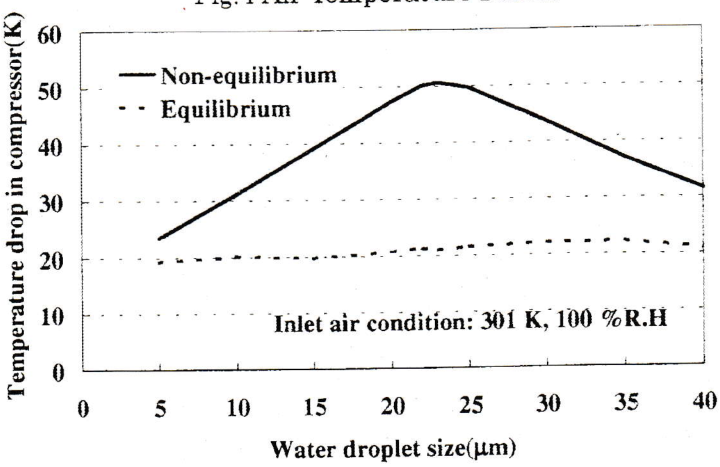

Fig.5 Temperature Drop Through the Compressor as Function of the Initial Water Droplet Size

3. EXPERIMENT

3.1 Sauter Mean Diameter

The accuracy of this calculation was verified by a controlled experiment. Definition of Sauter Mean Diameter(SMD) of water droplets with various diameter follows,

$$SMD = \Sigma(N_i{*}D_i^3)/\Sigma(N_i{*}D_i^2) \tag{3-1}$$

Where N_i is the number of droplets of ith band of droplet size. Physical meaning of the SMD is the average diameter under the condition that the total surface area of all droplets are conserved(equal to the surface area of a droplet with the SMD times the number of droplets) and is obtained experimentally. The SMD may be a good representative diameter for heat and mass transfer phenomena of droplets with distributed diameters because transfer area is conserved.

3.2 Experimental Verification

Experimental apparatus is shown in Fig.6. Hot air was introduced into a test section via a wind tunnel by a fan with heater. A spray nozzle was installed at the position of 600mm downward from the inlet of air. The two phase spray nozzle was an internal mixing type, to which water and air were supplied at the pressure of 0.39 MPa. The volume ratio of air to water was 1000 with the SMD of about 10 μm. The temperatures of air in the test section were measured at 14 points along the flow pass. The droplet sizes were measured at 50mm downward from the spray using a laser diffraction drop size analyzer. The accuracy of drop size measurement was examined using the standard particles with the diameters of 10.35 and 48.6 μm. The measuring error was within 4 %. The drop size distribution of the evaporated water was calculated by subtracting the drop size distribution measured at room temperature from that at higher temperature as shown in Fig.7. The volume decrease due to the water

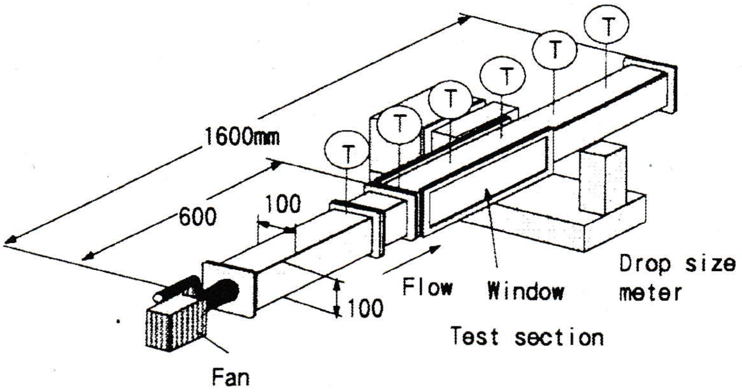

Fig.6 Experimental Apparatus

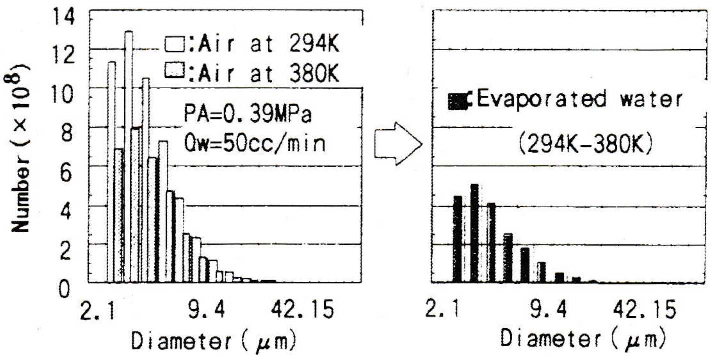

Fig. 7 Droplet Size of Evaporated Water

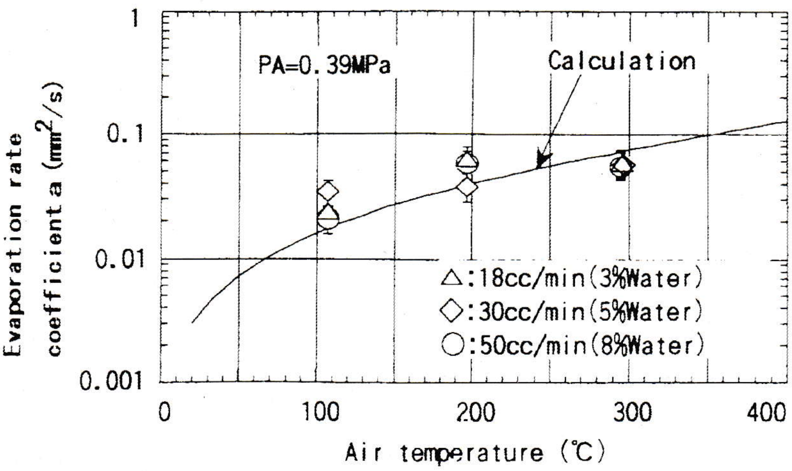

Fig.8 Comparison of Evaporation Rate Coefficient

evaporation could be expressed as follows using Eq. (2-1) and Eq.(2-4),

$$dV/dt = d(4\pi/3)\Sigma n_i r_i^3)/dt = 4\pi\Sigma n_i r_i (r_i dr_i/dt) = 4\pi\Sigma n_i r_i a\ F(M/R)(P(T_w)/T_w - P_{am}/T_{am})/\rho = 4\pi C\Sigma n_i r_i \quad (3\text{-}2)$$

where C is the evaporation rate coefficient and is assumed to be independent of the drop size within the experimental error when the drop is smaller than 10 μm. The air velocity was maximum at the center of the test section because of a jet from the nozzle. The average air velocity was 50m/s at the measuring point of the droplet size. As shown in Fig.8, the calculated evaporation rate coefficient was in good agreement with the data. So the evaporation model could be used under high temperature and high air velocity.

4. CONCLUSIONS

Evaporation behavior of water droplets was analyzed using the heat and mass transfer model with the effect of heat absorbed taken into consideration. Following conclusions were with drawn:

1) Above the droplet diameter of 5 μm, the heat absorption by the droplet i.e. non-equilibrium effect becomes significant.
2) Temperature drop of gas through compressor stages becomes maximum at water droplet initial diameter of 22 μm.
3) Suitable definition of droplet mean diameter was confirmed to be Sauter Mean diameter(SMD) with respect to cooling capacity.
4) The model were verified by the experiment using a high temperature wind tunnel.

NOMENCLATURE

a : molecular diffusivity of steam (m2/s)
Cp: specific heat (J/(kgK))
D: diameter of water droplet (m)
F: correction factor of mass transfer coefficient for forced convection (-)
G: correction factor of heat transfer for forced convection (-).
H: enthalpy (J)
h: specific enthalpy (kJ/kg)
k: thermal conductivity of gas mixture(practically dry air) (W/m/K)
L: evaporative latent heat of water (J/kg)
M : molecular weight of water (kg/mol)
m: mass of representative water droplet (kg)
N: the number of water droplets entering compressor at inlet 1/s)
nj: polytropic exponent at the jth stage (-)
P : saturated vapor pressure at Tw(Pa)
Pj: tatal gas pressure at the jth stage (Pa)
Pam: ambient vapor pressure(in the stream) (Pa)
Pr : Prandtl number (-)
Q: heat conduction to water droplet from ambient air (W)
Qav: average of Q in a stage(Eq.(2-11)) (W)
R : gas constant (J/(mol K))
Red: droplet Reynolds number (-)
r : water droplet radius (m)
S: surface area of a water droplet (m2)
Sc : Schmidt number (-)
S_{out}: external work (W)
Tam: dry air temperature in gas path (K)
Tj: gas temperature at the jth stage (K)
Tw: water droplet temperature (K)
Twi: water droplet temperature at stage entry (K)
Twe: water droplet temperature at stage exit (K)
Twav: average water temperature in a stage (Eq.(2-9)) (K)
T∞: droplet temperature in thermal equilibrium(K)
Δt: residence time of droplet in a stage (s)
ΔTd: temperature drop of air by evaporation (K)
t: time (s)
V: axial velocity of air (m/s)
w: evaporation rate of a representative water droplet (kg/s)

Wa: mass flow of dry air (kg/s)

Ws: mass flow of steam (kg/s)

x: axial position (m)

Greek letters

ρ : water density (kg/m3)

τ :time constant (s)

χ :absolute humidity (=Ws/Wa)(-

Suffices

a:air

am: ambient

av: average

d: drop

e: exit

g: gas

i: inlet

j: stage number

s: steam

w: water

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