

## A SIMULATED STUDY ON THE EFFECT OF WATER TEMPERATURE ON COOLING EFFICIENCY OF WATER MIST FIRE EXTINGUISHERS

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### ABSTRACT

In this paper, the effect of initial water temperature on the cooling performance of a water mist fire suppression system has been investigated. A zero-dimensional analytical model was first developed to study the thermal and dynamic behavior of a single water droplet. The developed model was validated against available experimental data in the literature. The developed model was further extended to simulate the transient heating, evaporation, break-up, and liquid penetration length of a hollow-cone water spray. Results indicated that increasing the initial temperature of the water spray before injection resulted in a decrease in the initial size distribution of the spray droplets. It was further found that the evaporation time of the injected droplets decreased by about 11% and the cooling power of the pre-heated water mist system enhanced by 12% in exchange for increasing the initial temperature of the water spray by 10°C. It was concluded that pre-heating the sprayed water droplets would likely improve the cooling efficiency of the water mist system. It could be also inferred that at equal cooling power, less water was consumed by the pre-heated spray compared to conventional water mist systems.

**Keywords:** *Cooling Efficiency, Droplet Evaporation, Spray Formation, Water Mist System*

### INTRODUCTION

Over the last decades, water spray systems have been utilized in a variety of applications within the areas of fire suppression, air conditioning, heat exchangers, and manufacturing processes [1-5]. Water mist systems are one of the recent high-pressure water spray systems that are widely used in fire suppression applications. In this system, water is sprayed under high pressure (600–4000 kPa) to generate ultra-fine water droplets (10-500 μm). The significant interests in using water mist systems are due to their high cooling capacity and less environmental impacts in comparison with other conventional fire suppression methods that are commonly based on using hydrogenated hydrocarbons. Due to the fine droplets generated by water mist systems, it is expected that the injected water droplets evaporate entirely during the operation of the water mist systems in hot ambient. Therefore, the detrimental effects and damages of the remaining water droplets in sensitive compartments such as libraries and heritage museums are eliminated; contrary to the other water-based fire suppression systems (sprinkler) which yield relatively large droplets. The water mist systems suppress the fire through three main factors: direct cooling of the fire, lessening the thermal radiation and oxygen displacement [6]. These factors are mainly prompted by the evaporation of the water droplets. Since the evaporation of the water droplets takes place in a relatively short time after injection, investigating the time-varying thermal and dynamic behavior of fine droplets within the spray through experimentation requires complex and costly optical techniques. Due to the limitations of such experimental studies, several numerical investigations have been performed to simulate the water mist fire extinguishing mechanisms [7-9]. Some primary studies have been accomplished to predict the physical phenomena arising in fire suppression by water mist systems in real scale compartments [10-12]. Yang et al. [10] and Kim and Ryou [11] developed and validated numerical models to obtain the temperature, oxygen concentration, and extinguishing time of water mist systems in a room fire. Adiga et al. [12] designed a numerically total flooding fire test for an ultra-fine water mist in a 28 m<sup>3</sup> compartment. Alongside these practical studies, some parametric investigations on fire suppression mechanisms have been done in small test scales [13,14].

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Santangelo [13] experimentally investigated the characteristics of a solid-cone water mist system, such as drop-size distribution, initial velocity, and cone angle of the water mist spray by using a laser-diffraction-based instrument and Particle Image Velocimetry (PIV) technique. Colin et al. [14] studied the detailed dynamics, thermal, and radiative behavior of a water spray.

The study and improvement of water mist performance due to its significant advantages is still of great interest to many researchers. In recent years, numerous studies have focused on the enhancement of the cooling efficiency of water mist systems. Researches have shown that the efficiency of the water mist systems can be considerably influenced by the injection conditions at the nozzle exit [15]. Using additives and chemicals [16,17], changing the characteristics of nozzle [18,19] and increasing the backpressure of the nozzle [20] are among the most important ways proposed to reduce the droplet mean diameter and change the initial size distribution at the nozzle exit.

In addition to the above-mentioned factors, the initial droplet size distribution is largely dependent on the thermophysical properties of the water. Since these properties are often temperature dependent, the initial water temperature seems to have a decisive influence on the cooling performance of water mist systems. In the present study, the effect of water preheating on the evaporation rate and cooling efficiency of water mist systems will be investigated. The main objectives of the current work can be summarized as (1) develop an analytical model to study the dynamics and evaporative behavior of a single water droplet and extend the proposed model to a generalized hollow-cone water mist spray, (2) study the effect of pre-heating the water spray on the size distribution and evaporation time of the injected droplets, and (3) investigate the qualitative behavior of the evaporative and cooling efficiency of the water mist system with a particular focus on water usage of these systems.

## MATHEMATICAL MODEL

The Eulerian-Lagrangian approach is usually used to study the dynamics and thermal behavior of spray flows [21, 22]. According to this method, the continuum gas phase was studied under the Eulerian description; while the dispersed liquid particles in the flow field could be modeled by the Lagrangian description in which individual particles were tracked in the flow [21]. Generally, coupling the two aforementioned descriptions to monitor the mutual interactions of the continuum gas flow and the particles required complex mathematical and numerical modeling. To simplify the modeling, it can be assumed that the surrounding gas flow is quiescent, especially in the modeling of the dynamics and thermal behavior of water sprays in hot environments where the water spray is sufficiently far from the fire plume [23]. In this case, the impact of the fire plume on the liquid particles can be assumed to be negligible; thereby, the Eulerian modeling can be neglected from the calculations. As a result of this assumption, the Lagrangian method can be solely considered to model the in-flight liquid droplets. The conservation equations for droplets in the Lagrangian framework are as follows.

### *Momentum conservation*

The momentum equation for a droplet of mass  $m_d$  can be described as [24]:

$$\frac{d(m_d \vec{V}_d)}{dt} = \frac{1}{2} C_D \rho_a A_d |\vec{V}_a - \vec{V}_d| (\vec{V}_a - \vec{V}_d) + m_d \vec{g} \quad (1)$$

$C_D$  represents the drag coefficient of a spherical evaporating droplet and can be expressed as [25]:

$$C_D = \frac{C_{D0}}{(1 + B_m)^\zeta} \quad (2)$$

where  $C_{D0}$  is the drag coefficient of a non-evaporating droplet, and can be determined by [24]:

$$C_{D0} = \begin{cases} \frac{24}{Re} \left( 1 + \frac{1}{6} Re_D^{\frac{2}{3}} \right) & ; Re_D \leq 1000 \\ 0.424 & ; Re_D > 1000 \end{cases} \quad (3)$$

where,

$$Re_D = \frac{V_{rel}D}{\nu_m} \quad (4)$$

$V_{rel}$  is the relative velocity between the droplet and the surrounding air, and  $\nu_m$  is the kinematic viscosity of the gas mixture at the vicinity of the droplet surface.

In Eq. (2),  $B_m$  is the Spalding number and represents the non-dimensional mass transfer force for evaporation of a moving droplet and can be expressed as [24]:

$$B_m = \frac{Y_m - Y_g}{1 - Y_m} \quad (5)$$

where  $Y_m$  and  $Y_g$  are the mass fraction of water vapor at the vicinity of the droplet surface and the surrounding hot air, respectively. According to the one-third law, the water vapor mass fraction at the vicinity of the droplet surface may be obtained by [26]:

$$Y_m = \frac{1}{3}Y_g + \frac{2}{3}Y_s \quad (6)$$

where  $Y_s$  represents the water vapor mass fraction at the droplet surface. Assuming equilibrium and using Raoult's law,  $Y_s$  can be calculated as [26]:

$$Y_s = \frac{P_{v,D}}{P_{env}} \cdot \frac{M_w}{M_{mix}} \quad (7)$$

$P_{v,D}$  is the saturated vapor pressure at the droplet temperature.  $M_w$  and  $M_{mix}$  are the molecular masses of water and gas mixture at the droplet surface. In Eq. (2),  $\zeta$  can be expressed as [24]:

$$\zeta = \begin{cases} 1 & ; B_m < 0.78 \\ 0.75 & ; B_m \geq 0.78 \end{cases} \quad (8)$$

### Conservation of mass

The evaporation rate of a droplet can be expressed by [25]:

$$\frac{d(m_d)}{dt} = -Sh(\pi D)\rho_m D_{v,a} \ln(1 + B_m) \quad (9)$$

Considering the assumption of the unity Lewis ratio [23], the Eq. (8) can be reformulated as:

$$\frac{d(m_d)}{dt} = -Sh(\pi D) \left( \frac{k_m}{C_{pm}} \right) \ln(1 + B_m) \quad (10)$$

Sh represents the Sherwood number which is the ratio of the convective mass transfer to the diffusive mass transfer. Sherwood number for a water droplet can be defined as [24]:

$$Sh = 2 + 0.6Re^{0.5}Sc^{0.33} \quad (11)$$

where  $Sc$  is Schmidt numbers of the moving droplet.

$$Sc = \frac{\nu_m}{D_{v,a}} \quad (12)$$

### Conservation of Energy

The thermal energy that is absorbed by the droplet from the hot surrounding air contributes to raising the temperature and vaporizing the water droplet. It has been shown that in a short time after injection, the temperature of the water droplet reaches the thermal equilibrium with the surrounding hot air; henceforth, the temperature of the droplet remains constant [24]. The equilibrium temperature can be determined by solving the energy balance between the droplet and the hot environment. The energy balance over a water droplet can be expressed by [23]:

$$m_d C_{pl} \frac{d(T_d)}{dt} = A_s h (T_a - T_d) + h_{fg} \frac{d(m_d)}{dt} \quad (13)$$

where  $h$  is the convective heat transfer coefficient of the moving water droplet. It is common to express the heat transfer coefficient as a function of the Nusslet number,

$$h = \frac{Nu k_m}{D} \quad (14)$$

The Nusselt number for a moving droplet can be expressed as [25]:

$$Nu = 2 + 0.6Re^{0.5}Pr^{0.33} \quad (15)$$

where  $Pr$  is the Prandtl number of the moving droplet.

$$Pr = \frac{\nu_m}{\alpha_m} \quad (16)$$

Since water mist droplets are very small, the inside temperature of the water droplets is assumed uniform [27]. It should be noted that the radiant heat exchange of the droplets with the surrounding environment has been ignored in the calculations [23,28].

### Initial droplet size distribution

It is common to define the size distribution of the water droplet at the nozzle exit as the initial condition when a spray is being described. To this end, the corresponding flow pattern of the water spray needs to be defined as well. It is already reported that the injected sprays in water mist systems are generally formed by pressure-swirl atomizers [29]. Therefore, it is expected that the flow pattern of the water mist sprays follows the hollow-cone nozzle injectors. Han, et al. [30] showed that the initial size distribution of the injected droplets by a hollow-cone water spray can be modeled by the Rosin-Rammler density function. Thus, to extend the mathematical model of the thermal and dynamic behavior of a water droplet to form a hollow-cone water spray, the Rosin-Rammler density function was employed to model the initial size distribution of the injected droplets. According to the Rosin-Rammler density function, the initial size distribution of the droplets can be expressed by [19]:

$$F(\psi) = \left( \frac{\psi^3}{X^3 \Gamma(1 + \frac{3}{q})} \right) \frac{q \psi^{q-1}}{X^q} \exp \left[ - \left( \frac{\psi}{X} \right)^q \right] \quad (17)$$

where  $\Gamma$  is the complete Gamma function and  $q$  is the dispersion coefficient. It has been reported that  $q=1.5$  can be used in modeling the hollow-cone water sprays [19]. In Eq. (17),  $X$  is the characteristic mean diameter of Rosin-Rammler PDF and can be defined as:

$$X = \frac{SMD}{\Gamma\left(1 + \frac{1}{q}\right)} \quad (18)$$

where SMD is the so-called ‘‘Sauter Mean Diameter’’, which is defined as the diameter of a representative droplet which holds the same volume to surface area ratio as the ratio of the sum of the entire droplets’ volume to the sum of all droplet surface area within the water spray. The relevant semi-empirical correlation for SMD in Eq. (18) for a hollow-cone water spray is [26]:

$$SMD = 2.25\sigma^{0.25}\mu_l^{0.25}\dot{m}_l^{0.25}\Delta P^{-0.5}\rho_a^{-0.25} \quad (19)$$

In Eq. (17),  $\psi$  is the random droplet diameter in the range of 0 to  $2X$ .

### *The secondary breakup of droplets*

The water droplets may become unstable under the action of aerodynamic forces induced by their motion in the continuous phase. According to Hsiang and Faeth model [31], breakup takes place if

$$We = \frac{\rho_a V_{rel} D}{\sigma} > 12 \quad (20)$$

and, the droplet diameter changes due to secondary breakup can be calculated from [31]:

$$\frac{d(D)}{dt} = -\frac{D - D_s}{\tau} \quad (21)$$

where,  $D_s$  is stable droplet diameter which is given by [31]:

$$D_s = 6.2D \left(\frac{\rho_l}{\rho_a}\right)^{0.25} \sqrt{\frac{\mu_l}{\rho_l D V_{rel}}} \quad (22)$$

In Eq. (21),  $\tau$  is characteristic breakup time which can be obtained by [31]

$$\tau = \frac{5}{\left(1 - \frac{Oh}{7}\right)} \frac{D}{V_{rel}} \sqrt{\frac{\rho_l}{\rho_a}} \quad (23)$$

where,

$$Oh = \frac{\mu_l}{(\rho_l D \sigma)^{0.5}} \quad (24)$$

### *Thermo-Physical Properties of water and vapor-air mixture*

The approximation of thermo-physical properties of water is summarized in an experimental study by Kukulka et al. [32] and Kincaid and Longley [33]. The thermo-physical properties of air and water such as viscosity, specific heat capacity, and thermal conductivity were calculated at the vapor-air film temperature. To calculate the aforementioned thermo-physical properties of the vapor-air mixture, the mass fraction of water vapor at the vicinity of the droplet surface was utilized as [34]:

$$\mu_m = Y_m \mu_v + (1 - Y_m) \mu_a \quad (25)$$

$$C_{p,m} = Y_m C_{p,v} + (1 - Y_m) C_{p,a} \quad (26)$$

$$K_m = Y_m K_v + (1 - Y_m) K_a \quad (27)$$

## SOLUTION PROCEDURE

In this section, a step by step procedure has been proposed for simulating droplet injection and spray modeling based on Lagrangian coordinate.

- 1- Considering several random diameters for the droplets injected at each time step. The selected diameters should be in the range of 0 to  $2X$ , where  $X$  is the mean diameter which can be obtained from Eq. (18).
- 2- Calculating the presence probability of droplets with any selected diameter, based on the Rosin-Rammler distribution function:  $P(D_i) = \frac{f(D_i)}{\sum f(D_i)}$
- 3- Calculating the number of drops corresponding to each diameter:  $N(D_i) = \frac{(\delta m)P(D_i)}{\frac{\pi D_i^3}{\rho \cdot 6}}$
- 4- Calculating the injection velocity at the nozzle exit. The velocity magnitude is calculated according to the pressure difference at the nozzle exit and the velocity vector direction is randomly selected for each droplet.
- 5- Calculating the droplets' velocities at  $t_0 + \delta t$  by Eq. (1)
- 6- Calculating the droplets' positions at  $t_0 + \delta t$  by integrating Eq. (1)
- 7- Calculating the new diameter of the droplets after evaporation and breakup
- 8- Calculating the droplets' temperatures after heat exchange with the surrounding environment

This procedure is repeated at every time step. To ensure that the calculations were independent of the time step, the time step was iteratively decreased, until the results converged at a time step of  $10^{-6}$  s.

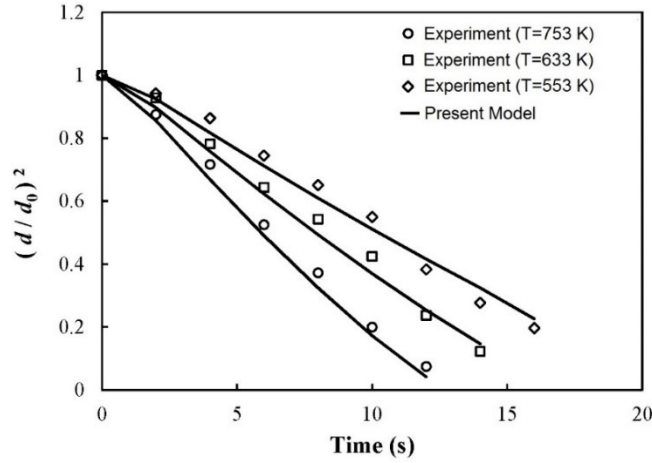
## VALIDATION

To validate the present mathematical model, the predicted non-dimensional diameter changes of a single water droplet was compared to that measured data by Guang et al. [35]. According to the reported experimental data in [35], a single water droplet with an initial diameter of 1.7 mm was exposed to a hot air ambient at different temperatures of 553, 633, and 753 K and the constant pressure of 0.1 MPa. As shown in Fig. 1, the result of the present model for evaporation of a single water droplet was in good agreement with the reported experimental data.

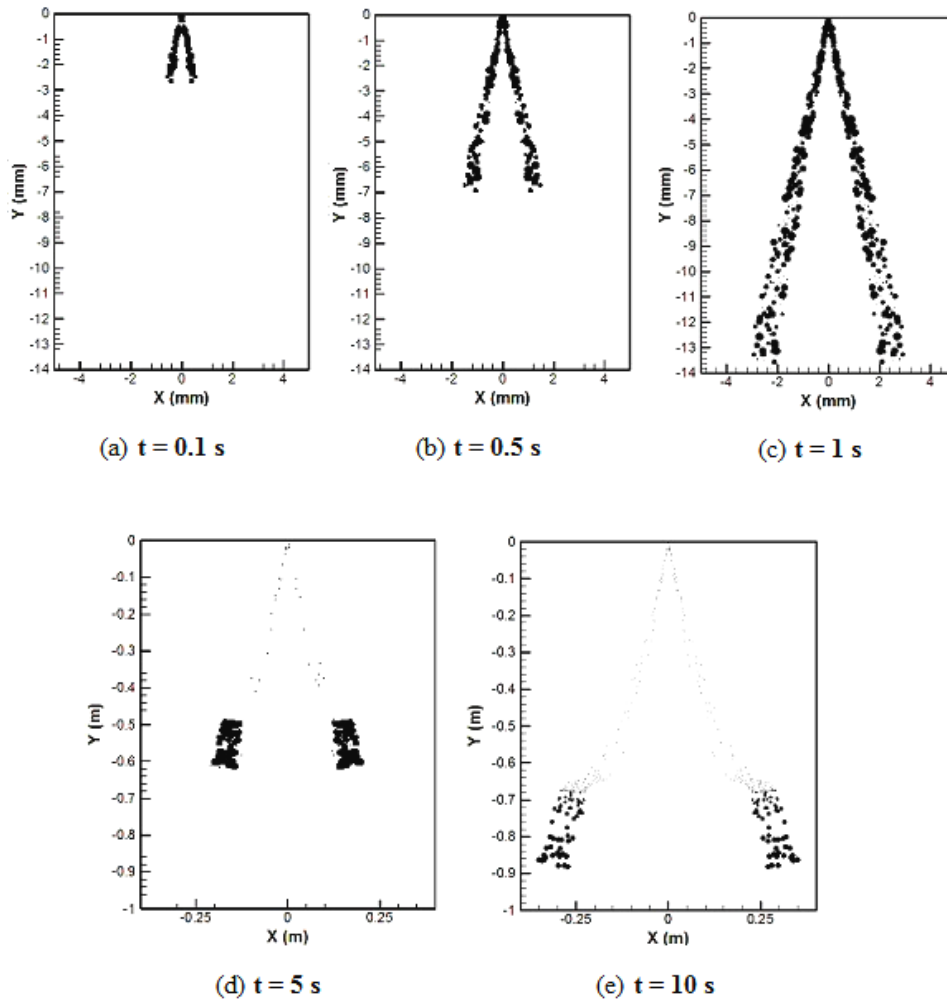
## RESULTS AND DISCUSSION

The developed analytical model for a single water droplet was extended to a hollow-cone water spray by incorporating the Rosin-Rammler distribution function for calculating the droplet sizes at the nozzle exit. At first, a preliminary evaluation was conducted to have a qualitative overview of the dynamic and thermal behavior of the modeled water mist spray. In this regard, the pressure difference of the nozzle was considered to be 10 MPa to generate fine droplets [1]. The nozzle diameter was considered as 2 mm and the mass rate of injected water was calculated as 0.45 kg/s. Water was injected into the hot air with a pressure of 0.1 MPa and a temperature of 400 K.

Figure 2 shows the dynamic and thermal behavior of a two-dimensional hollow-cone water spray after injection at five different times. As seen in Fig. 2, small droplets evaporate entirely at the shorter distance from the nozzle exit, while larger droplets penetrate further into the hot ambient. The dimensions of the scale bars are different in Fig. 2.



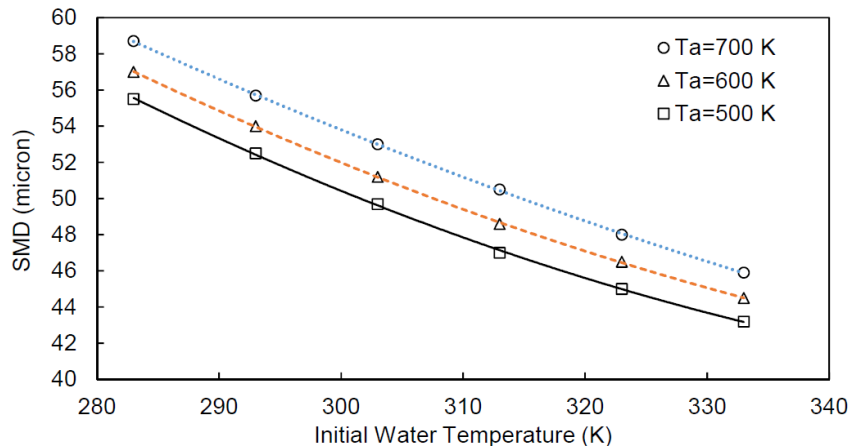
**Figure 1.** The comparison of the results of the present model for the transient variation of the non-dimensional diameter of a single water droplet with the experimental results reported by Guang et al. [35] at  $P_a = 0.1$  MPa at various air temperatures



**Figure 2.** Formation of the water spray: penetration and evaporation of the droplets in the air with a temperature of  $T_a = 400$  K and pressure of  $P_a = 0.1$  MPa at different times after injection start

Since the initial size of the droplets and their statistical size distribution at the nozzle exit has a very effective role on the spray cooling performance [1], the influence of the initial water temperature on the *SMD* of droplets at the vicinity of the nozzle exit has been studied in the following.

The variation of the *SMD* of a water spray with the initial temperature of the droplets at different ambient temperatures (500, 600, and 700 K) is shown in Fig. 3. For all ambient temperatures, the *SMD* of the water spray decreased almost linearly with increasing the initial temperature of droplets. This was likely since increasing the temperature of the droplets would decrease the viscosity and surface tension of water. As a result, the aerodynamic forces on the droplets due to their relatively high velocity at the nozzle exit would likely be predominantly higher than the surface tension forces within the droplet. Therefore, the tendency of water to primarily atomize into smaller droplets increased, resulting in smaller *SMD* of the water spray.



**Figure 3.** The variation of the initial *SMD* at the nozzle exit with the initial temperature of droplets at various air temperatures

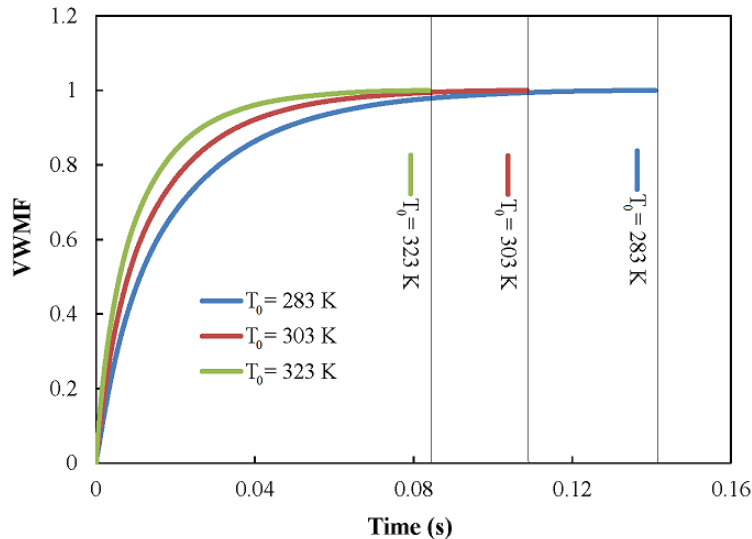
In the following, the effect of the initial water temperature on the evaporative rate of the water spray has been investigated. To this end, a non-dimensional parameter of “Vaporized Water Mass Fraction (*VWMF*)” was defined as the mass ratio of vaporized droplets to total injected ones. This parameter demonstrates the percentage of the evaporated water droplet after being sprayed into the hot ambient.

Figure 4 shows the variation of *VWMF* of the water mist spray in hot ambient air (700 K) from the starting moment of the injection (*VWMF*=0) until the time that water droplets evaporate entirely (*VWMF*=1) for initial temperatures of the water droplets of 283, 303, and 323 K. As observed in Fig. 4, droplets with higher initial temperature evaporated quicker than those with lower initial temperature. In other words, the so-called “pre-heated” droplets spent less time to evaporate entirely. This might be related to the initial size distribution of the sprayed water droplets. Spraying water with higher initial temperature generated droplets with a smaller diameter. As the diameter of the droplet decreased, the surface area to volume ratio of the water droplets increased. Increasing the surface to volume ratio resulted in an increase in the convective heat transfer between the hot ambient and the droplets and as a result, volumetric heating and evaporation rate of the droplet increased. Therefore, the evaporation time of the water spray decreased with increasing the initial temperature of the droplets. Moreover, droplets with higher temperatures possessed a greater value of Weber number. This was because the surface tension of the water decreased as the temperature increased [37]. As the Weber number of the droplet increased, their tendency to approach the critical Weber number and eventually undergo the secondary break-up increased. As a result of the secondary breakup, even much smaller droplets would be generated. Concerning the aforementioned justifications, the evaporation time of the water spray would decrease with increasing the initial temperature of the droplets.

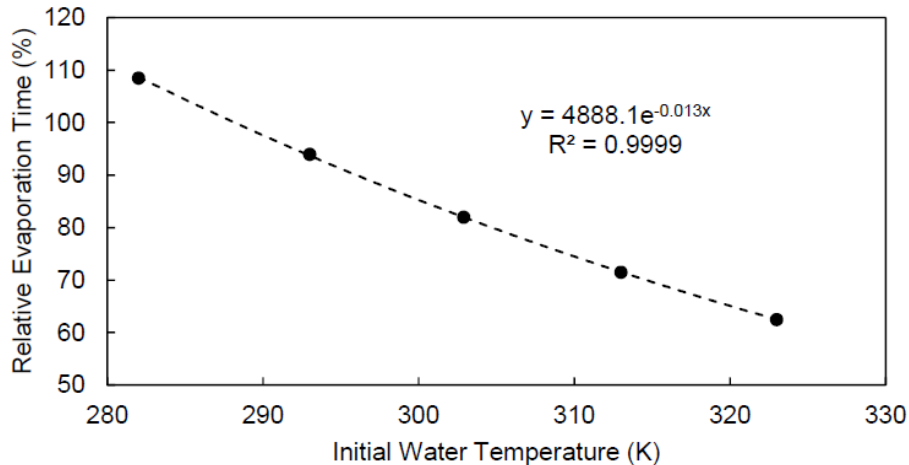
To have a quantitative comparison, a new parameter as “relative evaporation time” was defined as the ratio of the evaporation time of the entire water spray for different initial temperatures (293 - 323 K) to that with the reference initial temperature (283 K). Figure 5 shows the percentage of the relative evaporation time for a water mist spray with



different initial temperatures. As shown in this figure, the relative evaporation time decreased by increasing the initial temperature of the water droplets.



**Figure 4.** The transient variation of  $VWMF$  of the water mist spray for the various initial temperature of droplets

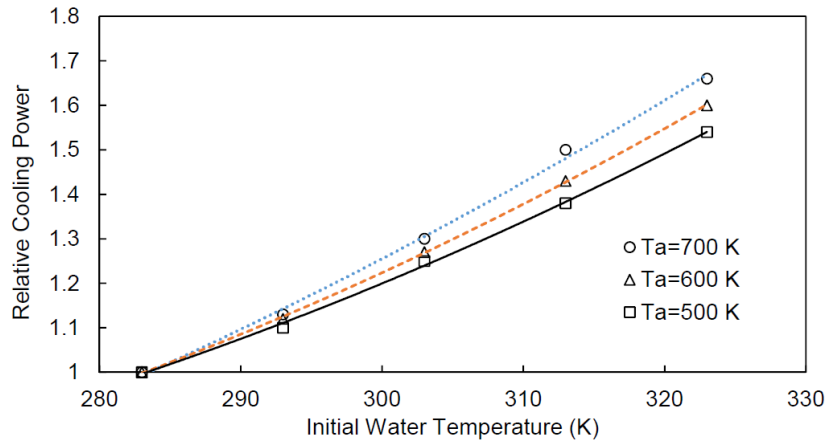


**Figure 5.** The relative evaporation time versus the initial temperatures of injected water droplets

It was observed that increasing the initial temperature of the spray by 10°C would result in the reduction of the evaporation time by approximately 11.5%.

In the following, the effect of the initial temperature of the sprayed water droplets on the cooling efficiency of the water mist system is investigated. To this end, it is necessary to compare the cooling performance of a preheated spray with that of non-preheated spray at a reference temperature. Accordingly, the non-dimensional “relative cooling power” was defined as the ratio of the cooling power of the preheated water mist system to the reference cooling power. In this regard, the cooling power of the water mist system was defined as the ratio of the total (sensible and latent) absorbed heat from hot ambient air to total evaporation time of the water spray. This definition took into account both evaporation of the water droplets and the rate of the heat extraction from the hot ambient through sensible heating of the droplets; therefore, it can be a good indicator for analyzing and comparing the cooling performance of the water mist system under different situations. In the present research, the cooling power of a water spray with an initial temperature of 283 K was considered to be the reference cooling power.

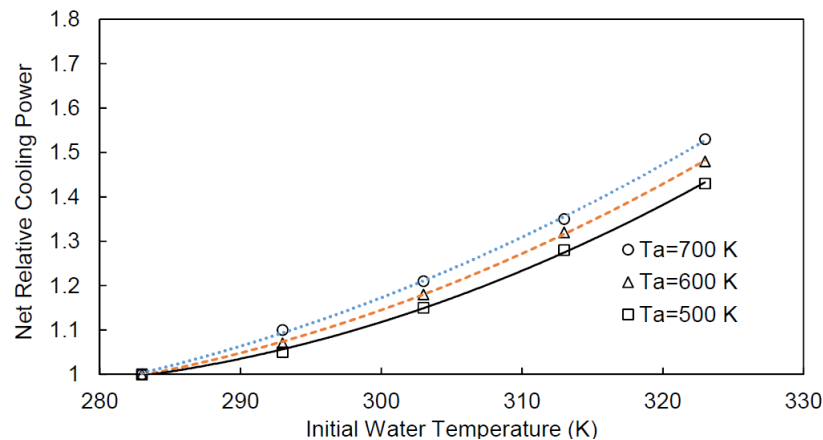
The variations of the relative cooling power of a water mist system with the initial temperature of the droplets that were sprayed into a hot ambient with different temperature is shown in Fig. 6. As shown in this figure, the estimated cooling power of the water mist system increased with the initial temperature of the water spray, which suggested that the cooling efficiency of the water mist systems could be enhanced by pre-heating the injected water spray. On average, it was observed that increasing the initial temperature of the water spray by 10°C resulted in the improvement of the cooling power of the water mist system by 12.5%. Through a qualitative analysis, it could be also inferred that at equal cooling power, the water consumption of the pre-heated water mist system reduced significantly.



**Figure 6.** The relative cooling power for various initial temperatures of droplets at various air temperatures

Moreover, it was further found that approximately 93% of the enhancement of the cooling power of the water mist system was due to the increase in the evaporation rate of the water droplets. In other words, the effect of the sensible heating of the droplets which were exposed to the hot ambient on the cooling efficiency of the water mist system was negligible, compared to the significant contribution of the evaporation of the droplets.

It should be noted that the required power for preheating has not been taken into account in the results presented in Fig. 6. If the power required for water preheating is also included in the calculations, the net cooling power will as be shown in Fig. 7. Although the relative cooling power values have slightly decreased by considering the power required for preheating, as can be seen in Fig. 7, water preheating still has a significant effect on improving the evaporative performance of water mist spray.



**Figure 7.** The net relative cooling power for various initial temperatures of droplets at various air temperatures

## CONCLUSION

The primary purpose of the current study was to study the effect of the pre-heating of the sprayed water droplets on the water mist cooling efficiency. To this end, an analytical approach was developed to model the evaporation rate, penetration, velocity, and break-up of a single droplet that was exposed to a hot ambient. The prediction of the analytical model was validated against experimental data in the literature. By using the Rosin-Rammler probability density function for the initial condition, the developed model for a single water droplet was extended to a hollow-cone water mist spray. The effect of the pre-heating the water droplet on the primary atomization of the water mist system was investigated and it was found that increasing the initial temperature of the droplets resulted in the smaller initial size distribution of the water droplets. Reducing the initial droplets' diameter can decrease their drag coefficient and thus increase the relative velocity between the droplets and the surrounding environment. Besides, decreasing the initial droplet diameter will increase the spray evaporation rate. It was further found that the evaporation time of the entire water spray reduced by 11.5% in exchange for increasing the initial temperature of the droplets by 10°C. The effect of the pre-heating on the cooling power of the water mist system was studied and it was observed that the cooling efficiency of the water mist fire suppression system increased as much as approximately 12.5% per 10°C increase in the initial temperature of a water droplet. It could be also inferred that the water usage of the pre-heated water mist system reduced significantly. Moreover, it was found that the cooling due to the sensible heating of the droplet is negligible compared to the evaporative cooling of the droplet.

As a complementary work, an experimental study is required to study the interaction of preheated water mist spray with a fire plume. Also, investigation of the effect of using additives and chemicals to improve the cooling performance of preheated water mist systems may be a good idea for the extension of the current study.

## NOMENCLATURE

$A$	Surface area, $m^2$
$B_m$	Spalding number, (-)
$C_D$	Drag coefficient of droplets, (-)
$C_{D0}$	Reference drag coefficient, (-)
$C_p$	Specific heat capacity, (J/kgK)
$d$	Droplet diameter, (m)
$D_{v,a}$	Binary diffusion coefficient of water vapor and air, ( $m^2/s$ )
$F$	Density function
$g$	Gravitational acceleration, ( $m/s^2$ )
$h$	Convective heat transfer coefficient, ( $W/m^2K$ )
$h_{fg}$	Latent heat, (J/kg)
$k$	Thermal conductivity, ( $W/mK$ )
$m_d$	Droplet mass, (kg)
$\dot{m}$	Mass flow rate, (kg/s)
Nu	Nusslet number, (-)
Oh	Ohnesorge number, (-)
$P$	Pressure, (kPa)
$P(D)$	Presence probability of droplet with diameter of D
Pr	Prandtl number, (-)
$q$	Dispersion coefficient
Re	Reynolds number, (-)
Sc	Schmidt number, (-)
Sh	Sherwood number, (-)
$SMD$	Sauter mean diameter, (m)
$t$	Time, (s)
$T$	Temperature, (K)

$V$	Velocity, (m/s)
$We$	Weber number, (-)
$X$	Mean diameter, (m)
$Y$	Mass fraction of water vapor, (-)

Greek symbols

$\alpha$	Thermal diffusivity, (m <sup>2</sup> /s)
$\Delta$	Difference
$\Gamma$	Gamma function
$\mu$	Dynamic viscosity, (kg/ms)
$\nu$	Kinematic viscosity, (m <sup>2</sup> /s)
$\rho$	Density, (kg/m <sup>3</sup> )
$\sigma$	Surface tension, (N/m)
$\psi$	Random droplet diameter, (m)

Subscripts

0	Initial value
$a$	Air
$cr$	Critical
$d$	Droplet
$g$	Gas phase
$l$	Liquid phase
$m$	Mixture
$p$	Projected area
$rel$	Relative value
$s$	Droplet surface
$v, a$	Vapor-air

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