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Thermal performance analysis of heat exchanger for closed wet cooling tower using heat and mass transfer analogy †

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Abstract

In closed wet cooling towers, the heat transfer between the air and external tube surfaces can be composed of the sensible heat transfer and the latent heat transfer. The heat transfer coefficient can be obtained from the equation for external heat transfer of tube banks. According to experimental data, the mass transfer coefficient was affected by the air velocity and spray water flow rate. This study provides the correlation equation for mass transfer coefficient based on the analogy of the heat and mass transfer and the experimental data. The results from this correlation equation showed fairly good agreement with experimental data. The cooling capacity and thermal efficiency of the closed wet cooling tower were calculated from the correlation equation to analyze the performance of heat exchanger for the tower.

Keywords: Closed wet cooling tower; Heat and mass transfer analogy; Heat and mass transfer coefficients; Heat exchanger

1. Introduction

The use of cooling towers in the air conditioning systems of buildings is increasing. A cooling tower is a device that cools water using heat from evaporating water, and it is used to remove the heat generated from air conditioner, manufacturing and power plant facilities. Closed wet cooling towers are high in heat efficiency and low in levels of contamination and noise with a low risk of freezing, which makes it possible to run the system continuously year-round in buildings, hospitals and schools.

The most important coefficient used in models to analyse closed wet cooling towers is the mass transfer coefficient between the spray water interface of the external tube side and the air and the heat transfer coefficient between the tube surface and the spray water. Several researchers have presented correlations that were developed for different geometry and operating conditions.

Parker and Treybal [1] were the first researchers to present a detailed analysis of an evaporative cooler and provide experimental correlations that estimate the heat transfer coefficient between the spray water and the tubes and the mass transfer coefficient between the saturated air-water interface and air. Mizushina et al. [2] tested an evaporative cooler with three different tube diameters (12.7 mm, 19.05 mm and 40 mm) and presented the heat and mass transfer coefficients. In another research [3], they considered the effect in variation of spray water temperature inside the tower while ignoring the evaporation of the spray water. Niitsu et al. [4, 5] tested tube banks of plain and finned tubes and suggested a correlation between the heat and mass transfer coefficients and diameter and air mass velocity, also ignoring the effect of the spray water flow rate.

Hasan and Siren [6] presented a theoretical analysis of a closed wet cooling tower for the heat and mass transfer coefficients using the Lewis relation. Gan and Riffat [7] applied CFD code to predict the performance of a closed cooling tower by using a two-phase flow of air and water droplets on the outside tube. Fisenko et al. [8, 9] developed a mathematical model for predicting the performance of cooling towers and their results were validated by measured data.

Facao and Oliveira [10] compared simplified models with detailed models and noticed that simplified models based on an overall approach provide as good or even better results as those based on finite differences. Stabat and Marchio [11] presented a simplified model for closed wet cooling towers based on effectiveness models using a simplification of heat and mass balance equations.

The objective of this study is to analyze the performance of heat exchanger for the closed wet cooling tower. The experiments have been conducted using two heat exchangers that had different tube diameters and arrays. The heat transfer co-

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efficient could be calculated from the equation for external tube surfaces of tube banks. The mass transfer coefficient calculated from the heat and mass transfer analogy was compared with experimental data. The regulated correlation equations were obtained from the result of the comparison. The cooling capacity and thermal efficiency of the closed wet cooling tower were calculated from provided equation, and the performance of the tower were investigated.

2. Experimental apparatus and procedure

The experimental apparatus was designed and fabricated for the capacity of 1 RT; the schematic is shown in Fig. 1. The dimension of the device is W700 mm×D200 mm×H1600 mm. The heat exchanger inside the experimental closed wet cooling tower was made of copper. Two tube bundles were used for heat exchanger presented here: an 22 row by 11 column inline arrangement with diameter of 9.52 mm and tube pitch of 17 mm (heat exchanger 1) and a 8 row by 5 column inline arrangement with diameter of 25.4 mm and tube pitch of 40 mm (heat exchanger 2).

The calibrated t-type thermocouple was attached to the wall of each inlet and outlet of air and cooling water to measure the temperature. The relative humidity of each part of the test device was measured using the humidity sensor (Delta ohm, HD2008TC1-1). The cooling water was heated to the set inlet temperature (37℃) by the constant temperature water tank, and the air was appropriately controlled to satisfy the wet bulb temperature of 27℃. The spray water temperature was not controlled. When the flow of the cooling water, spray water and air reached steady state conditions, the temperatures, relative humidity and flow rates were measured with various experimental conditions.

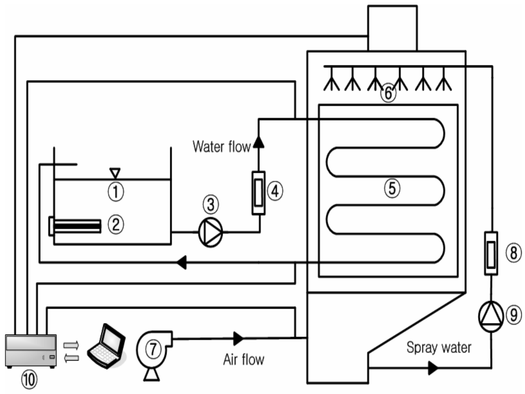
3. Heat and mass transfer coefficients

In the case of the closed wet cooling tower heat exchanger, the heat and mass transfer among the cooling water, spray water and the air is performed in a complex way as shown in Fig. 2; therefore, the existing normal heat exchanger design method cannot be applied. Even though only the heat transfer of the cooling water is considered to be inside the heat exchanger pipe, outside of the pipe, both the heat and mass transfers of the air and spray water need to be considered.

As numerous factors affect the transfer of heat and mass (e.g., pipe diameter, arrangement, air temperature, humidity, air supply method, spray water temperature, spray method, cooling water temperature, flow rate, etc.), the general experimental correlation equation is very difficult to obtain.

This study proposed the method of dividing the heat transfer rate external tube surface into the sensible heat transfer rate resulting from the temperature difference between the tube surface and the air, and the latent heat transfer rate that occurred from the evaporation of the spray water.

$$\dot{Q} = \dot{Q}_s + \dot{Q}_l \tag{1}$$



① constant temperature bath ② heater ③ cooling water circulation pump ④ cooling water flow meter ⑤ heat exchanger ⑥ spray nozzle ⑦ spray water flow meter ⑧ spray water circulation pump ⑨ fan ⑩ date acquisition

Fig. 1. Schematic of experimental apparatus.

Fig. 2. Flow scheme of heat exchanger for closed cooling tower.

The mass flow rate of the spray water is assumed small enough to be ignored. Thus, the heat transfer between the tube surface and the spray water was hypothesized to be included in the latent heat transfer of the external tube surface. The sensible heat transfer is calculated with the following equation.

$$\dot{Q}_s = h_o A F \Delta T_{lm} \tag{2}$$

$$\Delta T_{lm} = \frac{(T_{si} - T_{ao}) - (T_{so} - T_{ai})}{\ln\{(T_{si} - T_{ao})/(T_{so} - T_{ai})\}} \tag{3}$$

The latent heat transfer can be calculated from the evaporation rate and the latent heat of water as follows.

$$\dot{Q}_l = \dot{m}_e h_{fg} \tag{4}$$

$$\dot{m}_e = h_m A F \Delta\rho_{lm} \tag{5}$$

$$\Delta\rho_{lm} = \frac{(\rho_{sat,T_{so}} - \rho_{ai}) - (\rho_{sat,T_{si}} - \rho_{ao})}{\ln\{(\rho_{sat,T_{so}} - \rho_{ai})/(\rho_{sat,T_{si}} - \rho_{ao})\}} \tag{6}$$

The evaporation rate in Eq. (4) can be calculated from Eq. (5) using the mass transfer coefficient, and the log mean density difference can be calculated from Eq. (6) along the same concept as the log mean temperature difference in Eq. (3). The mass transfer coefficient can be obtained using the analogy of heat and mass transfer in Eq. (7).

$$Sh / Nu = \frac{(h_m D / D_{iff})}{(hD / k)} = (Sc / \Pr)^n \tag{7}$$

In Eq. (7), the mass diffusion coefficient, the thermal conductivity, the Schmidt number and the Prandtl number are properties of the fluid. When the heat transfer coefficient is identified, the mass transfer coefficient can be calculated from Eq. (7).

The equation used to calculate the heat transfer coefficient for tube banks can be expressed as follows, depending on the Reynolds number for air [12].

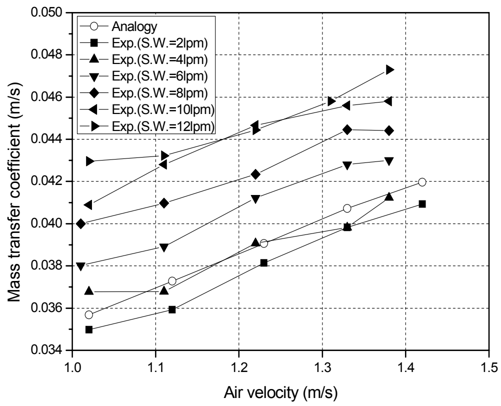
$$Nu = 0.52 \mathrm{Re}_a^{\ 0.5} \Pr^{0.36} (100 < \mathrm{Re}_a \le 1000) \tag{8}$$

$$Nu = 0.27 \mathrm{Re}_a^{\ 0.63} \Pr^{0.36} (1000 < \mathrm{Re}_a \le 2\times10^5) \tag{9}$$

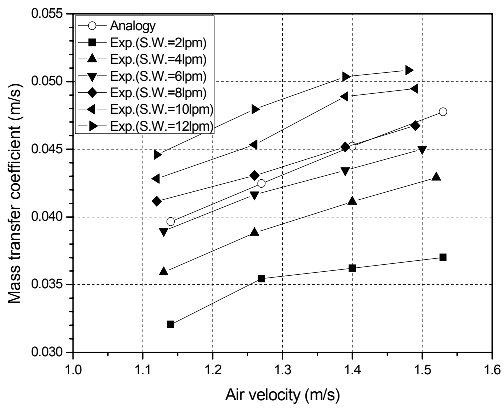
By using the analogy shown in Eq. (7), the mass transfer of the tube surface can be assumed as follows.

$$Sh_{ana} = 0.52 \mathrm{Re}_a^{\ 0.5} Sc^{0.36} (100 < \mathrm{Re}_a \le 1000) \tag{10}$$

$$Sh_{ana} = 0.27 \mathrm{Re}_a^{\ 0.63} Sc^{0.36} (1000 < \mathrm{Re}_a \le 2\times10^5) \tag{11}$$



(a) Heat exchanger 1 (D=9.52 mm)



(b) Heat exchanger 2 (D=25.4 mm)

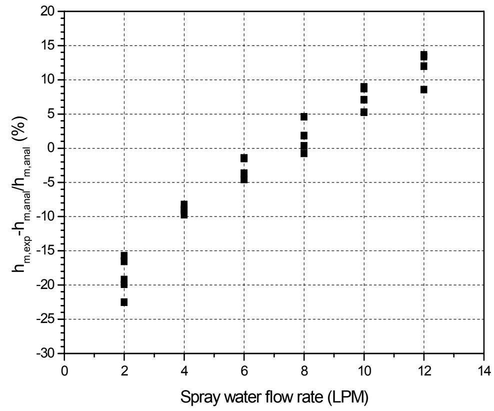
Fig. 3. Mass transfer coefficient with various velocities and spray water flow rates.

Fig. 3 shows the mass transfer coefficients calculated from the analogy and the experimental data.

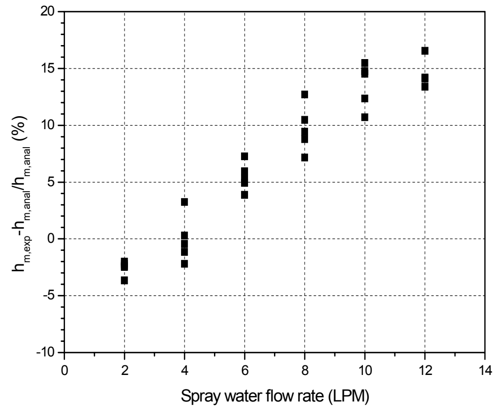
The mass transfer coefficient increased as the air velocity increased. Although the mass transfer coefficient from the analogy did not vary with the spray water flow rate, the experimental mass transfer coefficient increased with an increased spray water flow rate.

In the analogy, the tube surface was assumed sufficiently wet as a result of the spray water. However, when the spray water flow rate was low, the data calculated based on the analogy showed higher values than the experimental data. This tendency was more significant for heat exchanger 2 because tubes which have large diameter can be hardly sufficiently wet. And the analogy can be used on the assumption that tubes are completely wet.

Fig. 4 shows the differences between the analogy and the experiment. Here, the heat exchanger with the small diameter tube (heat exchanger 1) can be wet completely with a lower spray water flow rate than the heat exchanger with the large diameter tube (heat exchanger 2). In fact, the heat transfer between the tube surface and the spray water increased when the spray water flow rate was increased after the tube was sufficiently wet. When the spray water flow rate increases, the

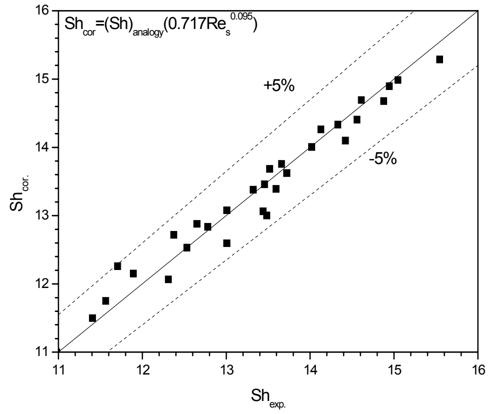


(a) Heat exchanger 1 (D=9.52 mm)

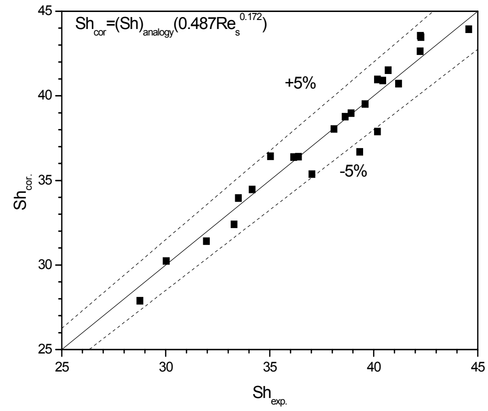


(b) Heat exchanger 2 (D=25.4 mm)

Fig. 4. Comparison of mass transfer coefficient.



(a) Heat exchanger 1 (D=9.52 mm)



(b) Heat exchanger 2 (D=25.4 mm)

Fig. 5. Comparison of Sherwood number.

mass transfer coefficient increases until tubes are wet completely and the heat transfer rate continuously increases after wet. In this study, heat transfer was included in the mass transfer, according to the hypothesis. Therefore, the mass transfer coefficient showed a continuous increase when the spray water flow rate was increased after the tube was sufficiently wet.

Even though the mass transfer coefficient was influenced by the flow rate of the spray water, this factor was not considered in the analogy. Thus, the following correlation equation was proposed.

$$Sh_{cor} = Sh_{ana}(C\,\mathrm{Re}_s^{\ n}) \tag{12}$$

Here, the constant and exponent of Eq. (12) differ with the tube arrays and spray characteristics, which, can be determined by the experimental data.

Fig. 5 shows the comparison of the Sherwood number between the calculated data from the correlation equation (Eq. (13), (14)) and the experimental data. As seen in the figure, the results from both heat exchangers were well aligned within the range of ±5%.

$$Sh_{cor} = Sh_{ana}(0.717\,\mathrm{Re}_s^{\ 0.095}) : Exchanger1 \tag{13}$$

$$Sh_{cor} = Sh_{ana}(0.487\,\mathrm{Re}_s^{\ 0.172}) : Exchanger2 \tag{14}$$

4. Thermal performance of heat exchanger

In order to evaluate the performance of closed wet cooling towers, calculations using the previous-mentioned equations were conducted with the following designed conditions for a 1 RT closed wet cooling tower using heat exchanger 1. The designed conditions consist of an air velocity of 1.5 m/s, air wet bulb temperature of 27℃, spray water flow rate of 6 LPM, cooling water flow rate of 13 LPM and cooling water temperature of 37℃.

The heat transfer rate in the closed wet cooling tower can be calculated from Eq. (15) below. Here, the mass transfer coefficient can be calculated using the correlation Eq. (12), and the evaporation rate in Eq. (5) can be expressed as the following Eq. (16).

$$\begin{aligned}\dot{Q} &= \dot{m}_w C_{pw}(T_{wi} - T_{wo}) \\ &= \dot{m}_a(i_{ao} - i_{ai}) \\ &= h_o AF\Delta T_{lm} + h_m AF\Delta\rho_{lm} h_{fg}\end{aligned} \tag{15}$$

$$\dot{m}_e = h_m AF\Delta\rho_{lm} = \dot{m}_a(W_{ao} - W_{ai}) \tag{16}$$

When the area of the heat exchanger, the inlet condition of the cooling water and the inlet condition of the air are given, the cooling capacity and the thermal efficiency of the closed wet cooling tower can be calculated from Eqs. (13)-(16).

The thermal efficiency for closed wet cooling tower defined as follow.

$$\varepsilon = \frac{T_{wi} - T_{wo}}{T_{wi} - T_{wb}} \tag{17}$$

As seen in Fig. 6, the performance parameters are determined by the flow rate of air. As described previously, the heat and mass transfer coefficients are affected by the air and spray water flow rate. Therefore, the cooling capacity and the efficiency of the cooling tower increases with an increase in air flow rate.

Air wet bulb temperature has an influence on the cooling capacity and the efficiency which is shown in Fig. 7. The latent heat transfer between the air and tube surfaces is determined by the density difference between them. Thus, when the wet bulb temperature of the air increases, the cooling capacity is significantly reduced. However, the efficiency of the tower increases slightly with the increasing wet bulb temperature. This increase is linear.

The effect of the spray water flow rate can be seen in Fig. 8. An increase in spray flow rate slightly increases the cooling capacity and the efficiency as the heat and mass transfer coefficients are affected by the spray rate. However, the effect of the spray water flow rate is relatively small compared to the

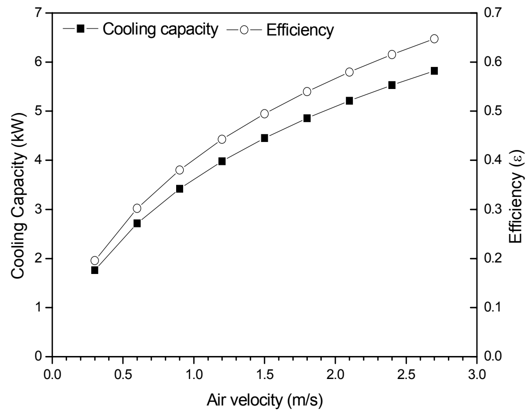


Fig. 6. Influence of air velocity in cooling capacity and efficiency.

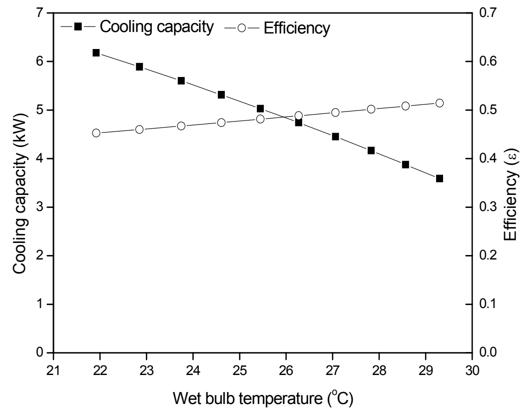


Fig. 7. Influence of air wet bulb temperature in cooling capacity and efficiency.

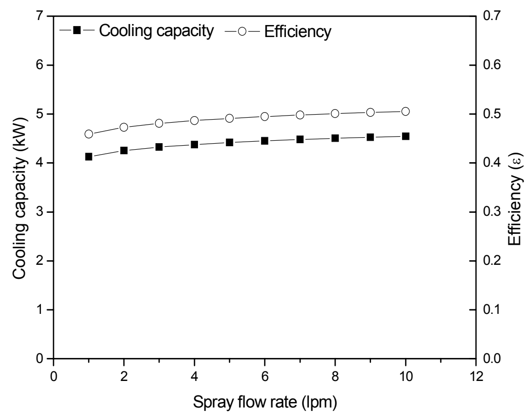


Fig. 8. Influence of spray flow rate in cooling capacity and efficiency.

effect of the air flow rate.

The tower`s cooling capacity and efficiency while varying the cooling water inlet temperature are shown in Fig. 9. When the temperature and density differences between the tube surface and the air increase according to the increase in water inlet temperature, the heat and mass transfer coefficients increase. Thus, the cooling capacity and the efficiency increase. However, as can be seen in the figure, the increase in the efficiency is not significant compared to the increase in the cooling capacity.

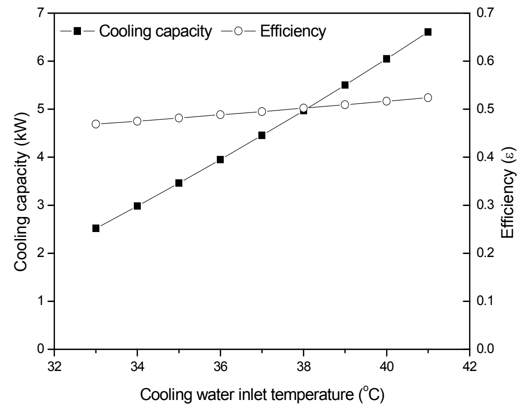


Fig. 9. Influence of cooling water inlet temperature in cooling capacity and efficiency.

5. Conclusion

The heat transfer rate of the external tube surface of the heat exchanger for a closed wet cooling tower can be divided into sensible and latent heat transfer rates. These in turn are expressed by heat and mass transfer coefficients. According to the results of the experimental data, the mass transfer coefficient shows an increase as the air velocity and spray water flow rate increase. Based on the analogy of the heat and mass transfer, the mass transfer coefficient can be estimated from the increase in the air Reynolds number. Considering the effect of the spray water flow rate seen in the experimental data, the correlation equation to estimate the latent heat transfer coefficient can be obtained. The result from the correlation equation shows accuracy within 5%. Using this equation, the performance of the closed wet cooling tower were investigated. The cooling capacity increases when the air velocity, spray water flow rate, and cooling water inlet temperature increase. Furthermore, the tower`s thermal efficiency increases with increase in the air velocity, inlet wet bulb temperature of the air, spray water flow rate, and cooling water inlet temperature.

Acknowledgment

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Nomenclature

A : Area of heat exchanger
C_{pw} : Specific heat of cooling water
D : Diameter of tube
D_{iff} : Mass diffusion coefficient
F : Correction factor

h_m : Mass transfer coefficient
h_{fg} : Vaporization heat
h_o : Heat transfer coefficient for external tube
i_{ai} : Enthalpy at the air inlet
i_{ao} : Enthalpy at the air outlet
k : Thermal conductivity
$\dot{m}_a$: Mass flow rate of the air
$\dot{m}_e$: Evaporation rate
$\dot{m}_w$: Mass flow rate of the cooling water
Nu : Nusselt number
Pr : Prandtl number
$\dot{Q}$: Total heat transfer rate
$\dot{Q}_l$: Latent heat transfer rate
$\dot{Q}_s$: Sensible heat transfer rate
Re_a : Reynolds number for the air
Re_s : Reynolds number for the spray water
Sc : Schmidt number
Sh : Sherwood number
Sh_{ana} : Sherwood number from analogy
Sh_{cor} : Sherwood number from correlation equation
ΔT_{lm} : Log mean temperature difference
T_{si} : Temperature at surface of cooling water inlet
T_{so} : Temperature at surface of cooling water outlet
T_{ai} : Temperature at the air inlet
T_{ao} : Temperature at the air inlet
T_{wb} : Wet bulb temperature at the air inlet
T_{wi} : Temperature at the cooling water inlet
T_{wo} : Temperature at the cooling water outlet
W_{ai} : Humidity ratio at the air inlet
W_{ai} : Humidity ratio at the air outlet

Greek symbols

ε : Thermal efficiency for the cooling tower
$\Delta\rho$: Log mean density difference
$\rho_{sat,Tsi}$: Saturation density at surface of the cooling water inlet
$\rho_{sat,Tso}$: Saturation density at surface of the cooling water outlet
ρ_{ai} : Density at the air inlet
ρ_{ao} : Density at the air outlet

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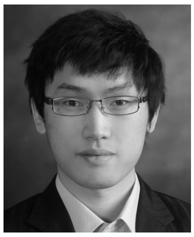

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