Distribution of heat transfer coefficient in the vertical tube of falling film evaporator treating saline wastewater based on micro flow and experimental verification

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ABSTRACT

It is still one of the significant solutions to treat saline wastewater with thermal desalination technology, especially falling film evaporators. To improve the performance of the falling film evaporator, a numerical study on the gas–liquid two-phase flow characteristics of saline wastewater in the vertical pipe was conducted using the VOF model. The results showed that the inlet velocity of the saline wastewater increased under the same operating conditions, resulting in the thickening of the liquid film and the increase of the average convective heat transfer coefficient. Increasing the inlet temperature of the working liquid reduced the temperature difference, which led to a decrease of the average convective heat transfer coefficient. In addition, as the inlet concentration of the working liquid increased, the film flow rate and the average convective heat transfer coefficient first decreased and then increased slightly. The experimental results verified the accuracy of the numerical simulation, and the average error was 9.27%.

Key words: convective heat transfer coefficient, falling film evaporation, numerical simulation, saline wastewater

HIGHLIGHTS

- The study of fluid microflow combined with heat transfer was proposed.
- The distribution of heat transfer coefficient based on numerical simulation.
- The distribution of heat transfer coefficient is correctly verified by experiments.
- The effects of the various operating parameters on the heat and mass transfer processes are studied.

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1. INTRODUCTION

Desalination of saline water (e.g. seawater and saline wastewater) is beneficial to alleviate the shortage of fresh water and the pollution caused by the discharge randomly of saline wastewater (Liang et al. 2013; Son et al. 2020). Currently, the desalination methods, including biological process (Xue et al. 2017; Fu et al. 2021), ultrafiltration, microfiltration, nanofiltration, reverse osmosis, and thermal process (Zhao et al. 2011), are widely studied. Among these methods, the thermal process is one of the significant approaches (Xue et al. 2019; Liu et al. 2020). The thermal technology accounts for 29% of the global desalination market share, whose market share even achieves 70% in the Gulf Cooperation Council (GCC) countries (Al-Mutaz & Wazeer 2014). The thermal process, also known as the evaporation method, is divided into multi-effect desalination, multi-stage flash, and mechanical vapor recompression. Because of the simple pretreatment, comparatively long equipment life, and concise working principle (Dahmardeh et al. 2019), these technologies are widely used. As the main equipment of these technologies, the heat transfer performance of the falling film evaporator has a direct relationship with the entire evaporation system.

Falling film evaporators are mainly divided into vertical tube falling film evaporators and horizontal tube falling film evaporators. Vertical falling film evaporators are more popular due to their obvious advantages such as small footprint, high heat transfer coefficient, and more stable performance (Kouhikamali et al. 2014). Therefore, in order to get the heat transfer law of falling film evaporators and improve energy efficiency, many researchers have conducted systematic research. The structural parameters and operating parameters have the most direct influence on the heat transfer performance of the falling film evaporator. Song et al. (2019) studied the effects of feed flow rate, evaporation temperature, temperature difference, and juice soluble solid content on the heat transfer coefficient of a vertical falling film evaporator. They used the Bayesian tree Gaussian process model to identify that the most sensitive variable affecting the heat transfer coefficient is evaporation temperature. The results of Shi et al. (2010) showed that working pressure had no effect on heat and mass transfer. Besides, they also obtained the relation between dimensionless heat transfer coefficient and the inlet Reynolds number. Wan et al. (2016) investigated the effects of inlet flow rate, wall structure, working fluid composition, and heat flux density on heat and mass transfer and proposed the nonuniformity of local heat transfer coefficient distribution on the wall. Matsekh & Pavlenko (2007) found that when the aluminum alloy plate was used as the heating wall, the local convective heat transfer coefficient first increased and then decreased from the upper part to the lower part of the wall.
The above research studies were all carried out at the macro level. However, it was also necessary to study the microscopic flow in the falling film evaporation tube (such as the flow rate and temperature of the liquid film). Liu et al. (2020) obtained the diameter and length of the evaporation tube with the best heat transfer efficiency by analyzing the axial and radial velocity, temperature, and liquid volume fraction of the liquid film. Wang et al. (2020) used the VOF model to perform numerical simulations to obtain the influence of tube length, tube diameter, and annular membrane spacing on film distribution and film fluctuations. Fang et al. (2019) developed mathematical models of flow in pipes to acquire the film thickness, liquid volume, velocity distribution, and temperature distribution of condensate at different axial positions outside the tube. And the accuracy of the model through experiments gradually was verified.

Most of the previous studies mainly focused on one side but did not combine the two aspects. In fact, the influence of structure parameters and operation parameters has complex effects on the internal flow field and heat transfer coefficient of falling film evaporation tube, and there is an inevitable relationship between the two factors. In this paper, the method of computational fluid dynamics (CFD) is used to study the relationship between the motion of the microfluid in the falling film evaporation tube and the heat transfer coefficient (Wen et al. 2020). The distributions of velocity, temperature, thickness, and heat transfer coefficient on the liquid film are studied. The influence of different variables on these parameters is integrated. The flow state of the liquid film and the relationship among the changes of heat transfer coefficients are obtained. The distribution and flow law of liquid film in the vertical tube and the variation of local convective heat transfer coefficient with tube length are studied and verified by experiments.

2. NUMERICAL SIMULATION METHOD

2.1. Mathematical model of flow

With the development of CFD, numerical simulation technology has been applied to analyze and predict the heat and mass transfer performance of vertical evaporation tubes. The falling film evaporation process in a vertical tube belongs to the category of stratified free surface flow of gas–liquid two-phase flow, in which the capture of the gas–liquid interface is an important part of the entire process. Therefore, in this paper, the VOF model is used to numerically simulate the falling film process in the vertical tube.

In the VOF model, the volume fraction of fluid phase \( \alpha \) is defined, and in each control volume, the sum of volume fraction \( \alpha_b \) of different phases is 1:

\[
\sum_{k=1}^{n} \alpha_b = 1
\]  

(1)

where \( n = 2 \), \( b = L \) (liquid phase) or \( G \) (gas phase), and the sum of the two-volume fractions is equal to 1. The density \( \rho \) and viscosity \( \mu \) in each control body can be given by the following two equations:

\[
\rho = \alpha_L \rho_L + (1 - \alpha_L) \rho_G \quad (2)
\]

\[
\mu = \alpha_L \mu_L + (1 - \alpha_L) \mu_G \quad (3)
\]

Then the continuity equation can be described as follows:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{u}) = S_t
\]  

(4)

where \( \rho \) is the density of the fluid, \( t \) is the time, and \( \vec{u} \) is the velocity vector. Since there is no phase transition in the cold state film-forming process, the mass source term \( S_t \) is zero. However, in the falling film process involving evaporation, since mass transfer occurs between the gas and liquid phases, the mass transfer is the mass source term \( S_t \).

The two phases in each control body use the same velocity field. And the momentum equation can be written as follows:

\[
\frac{\partial}{\partial t} (\rho \vec{u}) + \nabla \cdot (\rho \vec{u} \vec{u}) = -\nabla p + \nabla \cdot [\mu (\nabla \vec{u} + \nabla \vec{u}^T)] + \rho \vec{g} + F
\]  

(5)
where $p$ is the pressure, $\mu$ is the viscosity, $T$ is the temperature, $\bar{g}$ is the acceleration of gravity, and $F$ is the volume force source term.

The energy equation can be written as follows:

$$\frac{\partial}{\partial t} (\rho E) + \nabla \cdot (\rho \vec{u}(E + p)) = \nabla \cdot (k_{\text{eff}} \nabla T) + Q$$

(6)

where $E$ is the energy, $k_{\text{eff}}$ is the effective heat transfer coefficient, and $Q$ is the source term of fluid.

According to the conditions of the working fluid and the characteristics of large flow falling film in the tube, the RNG $k-\varepsilon$ model is selected as the turbulence model. And the turbulence viscosity $\mu_t$ is related to turbulent energy $k$ and turbulent dissipation rate $\varepsilon$:

$$\mu_t = C_\mu \rho \frac{k}{\varepsilon}$$

(7)

The turbulent energy equation $k$ is as follows:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho u_i k)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \alpha_k \mu_{\text{eff}} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S$$

(8)

where $\alpha_k$ is the turbulent Prandtl number, $\mu_{\text{eff}}$ is the effective viscosity, $G_k$ is the turbulent kinetic energy generated by the laminar velocity, $G_b$ is the turbulent kinetic energy generated by buoyancy, $\varepsilon$ is the dissipation rate of turbulent kinetic energy, and $Y_M$ is the wave generated by the transition diffusion in compressible turbulence.

The turbulent dissipation rate equation $\varepsilon$ is as follows:

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho u_i \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_j} \left[ \left( \alpha_\varepsilon \mu_{\text{eff}} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} \left( G_k + C_{3\varepsilon} G_b \right) - C_{2\varepsilon} \frac{\rho \varepsilon^2}{k} - R$$

(9)

where $C_{1\varepsilon}$, $C_{2\varepsilon}$, and $C_{3\varepsilon}$ are the constants, and $R$ is a user-defined parameter.

### 2.2. Mathematical model of heat transfer

The convective heat transfer coefficient of the liquid film can be obtained by Newton’s cooling formula:

$$h = \frac{q_w}{\Delta T} = \frac{q_w}{T_w - T_m}$$

(10)

where $T_w$ is the wall temperature, $T_m$ is the average temperature of the liquid film, and $q_w$ is the heat flux density. The heat flux is obtained by the following formula:

$$q_w = \frac{Q_L}{\pi D_l L_1}$$

(11)

where $Q_L$ is the heat flow rate.

### 2.3. Physical model and boundary conditions

To study the heat and mass transfer effect of falling film evaporation tubes, the evaporation tube was modeled by 3D modeling software. The structural parameters are shown in Table 1.

The ‘velocity inlet’ boundary condition is adopted at the inlet, and the ‘outflow’ boundary condition is used at the outlet. The heating wall of the evaporation tube is defined as constant temperature, no-slip, no penetration, and zero thickness, and the other wall surfaces are defined as no-slip, no penetration, and insulation. The ‘Standard Wall Function’ is adopted in the wall part. The VOF model is selected as a multiphase flow model, the RNG $k-\varepsilon$ model is selected as a turbulence model, ‘SIMPLE’ is set as pressure velocity coupling, and ‘QUICK’ is set as
discretization scheme of volume fraction and momentum. ‘PRESTO!’ is used in the discretization scheme of pressure.

2.4. Problem assumptions
To better explain this problem, some assumptions have been made to simplify the model (Wang et al. 2020):
(1) The pressure on the cross-section of the evaporation tube is uniform.
(2) Thermodynamic equilibrium is maintained in the tube.
(3) NaCl solution is an incompressible Newtonian fluid.
(4) The effect of the velocity of the vapor on the falling film is negligible.

2.5. Independence detection
The physical structure and meshing of the evaporation tube are shown in Figure 1. To capture the boundary features and improve the accuracy of mesh generation, the local refinement method is used to refine the mesh of the heating wall. Besides, the number of meshes has a significant influence on the simulation results. To verify the independence of the number of meshes, seven mesh groups with a total number of meshes ranging from 94,986 to 1,628,483 are used, and the film thicknesses at the axial of 500 mm are compared. Figure 2 shows the effect of the number of meshes on the liquid film thickness. The theoretical calculation is based on Mudawar

| Physical parameters | Symbol | Value (mm) |
|---------------------|--------|------------|
| The upper diameter of the distributor | D₂ | 30 |
| The length of the evaporation tube | L₁ | 1,000 |
| The inner diameter of the evaporation tube | D₁ | 40 |
| The height of the distributor | H | 23 |
| The wall thickness of the evaporation tube | δ₁ | 2 |
| Annular gap | L₂ | 1.5 |

Figure 1 | Physical model and meshing.
& Lyu’s (1991) theoretical value. And the formula of liquid film thickness $\delta$ is as follows:

$$\begin{align*}
\text{Re} \leq 1,600, & \quad \delta = \left(\frac{3\mu^2}{\rho \gamma g} \right)^{\frac{1}{5}} \left(\frac{R_e}{4}\right)^{\frac{1}{5}} \\
\text{Re} > 1,600, & \quad \delta = 0.302 \left(\frac{3\mu^2}{\rho \gamma g} \right)^{\frac{1}{5}} \left(\frac{R_e}{4}\right)^{\frac{1}{5}}
\end{align*}$$

(12) (13)

where $\delta$ is the liquid film thickness, and $R_e$ is the Reynolds number.

As shown in Figure 2, the calculation results increase with the increase of the meshing number. But, when exceeds 500,000, the results show only a slight difference. To save time and computational resources, a total of 505,216 meshing numbers is selected.

### 3. RESULTS AND DISCUSSION

#### 3.1. Physical parameters

Before analyzing the rules of mass and heat transfer, it is very important to gain the physical parameters in the evaporation process. Therefore, according to Sharqawy et al. (2011), the four parameters of saline wastewater (e.g. density, specific heat capacity, thermal conductivity, and viscosity) are fitted by polynomials and shown in Figure 3.

As shown in Figure 3(a), the density of the saline wastewater decreases as the temperature increases. In Figure 3(b), it can be seen that the specific heat capacity of the saline wastewater increases with the increase of temperature. It can be seen from Figure 3(c) that the thermal conductivity of the saline wastewater increases with the increase of temperature. As plotted in Figure 3(d), the viscosity decreases rapidly with the increasing temperature. What’s more, with the more concentration of saline wastewater, the density and viscosity of the saline wastewater are higher, but the specific heat capacity and thermal conductivity are both lower.

#### 3.2. Analysis of heat transfer performance in the vertical tube

Based on the boundary layer theory, the film thickness (Li et al. 2018), temperature distribution, and flow rate of the liquid film (30% volume fraction of liquid was used as the boundary between liquid film and vapor) in the evaporation tube were obtained by numerical simulation.

#### 3.2.1. Effect of inlet velocity on heat and mass transfer in tube

As shown in Figure 4, the distribution of the axial and radial velocities in the longitudinal direction of the tube was shown. The results indicated that the radial velocity was basically zero along the axial direction of the tube, except for slight fluctuations at the inlet due to the liquid distributor of the evaporation tube.
Figure 4 also shows that the axial velocity increased rapidly due to the effect of gravity at the inlet (0–100 mm), then basically kept at about 0.9 m/s (100–800 mm), and decreased at the outlet (800–1,000 mm). Additionally, the axial velocities in the different inlet velocities were slightly increased in the same range (0–100 mm). Taking the axial velocities of 0.1 and 0.175 m/s at 100 mm as an example, the axial velocities were 0.875 and 0.9 m/s, respectively.

The temperature distribution in the evaporation tube was directly related to its heat transfer performance. Figure 5 shows the temperature distribution under different conditions. Combined with Figure 5(a) and 5(b), it could be seen that the temperature of the saline wastewater increased slowly after entering the evaporation tube and reached the maximum value of about 380 K at the outlet. Additionally, as depicted in Figure 5(c),
the temperature close to the wall surface dropped sharply, which indicated that the heat transfer process was mainly concentrated in the liquid film region near the wall. Along the radial direction, the range of temperature change in the region, where vapor existed, was not large. Besides, according to Figure 5(b) and 5(c), the temperature of the fluid decreased due to the higher inlet velocity of the saline wastewater. The main reason was that the increase of inlet velocity led to the increase of saline wastewater in the tube, and it took a longer heating time to reach the same temperature.

Except for the temperature distribution, the liquid film thickness in the evaporation tube was also closely related to heat and mass transfer (Gao et al. 2018). The variation of liquid film thickness along the tube and the distribution of liquid volume fraction at different positions are shown in Figure 6.

As shown in Figure 6(a), the thickness of the liquid film decreased sharply after the saline wastewater entering the evaporation tube. The main reason was that the liquid just flowed out of the gap of the distributor and accumulated locally. Subsequently, due to the effect of gravity and inertia, the liquid film presented a relatively stable state in the process of flowing down. Then, the liquid film thickened gradually in the range of 100–800 mm, which is due to the appearance of bubbles near the wall. In the range of 800–1,000 mm from the inlet, the evaporation process was very severe, resulting in a decrease of the film thickness.

Besides, as also shown in Figure 6(a), the increase of inlet velocity led to the thickening of the liquid film. Moreover, it can be seen in Figure 6(b) that the increase of inlet velocity led to the increase of liquid volume fraction in the vapor region of the tube. The main reason was that the heat transfer coefficient in the tube is higher, so the evaporation efficiency increased.

3.2.2. Effect of inlet temperature on heat and mass transfer in tube

Figure 7 shows that the increase of inlet temperature led to a slight increase of the axial velocity in the range of 100–790 mm. The main reason was that the increase of temperature decreased the viscosity of saline wastewater, hence, reduced the resistance of liquid flow, which would increase the film flow rate.
It could be found from Figure 8(b) and 8(c) that the increase of the inlet temperature of the saline wastewater made the temperature in the overall tube increase. Then, the saline wastewater had the largest temperature difference at the inlet and increased to the same temperature at the outlet.

According to Figure 9(a) and 9(b), increasing the inlet temperature had little effect on the liquid film thickness in the range of 0–600 mm but had an obvious effect in the range of 600–1,000 mm. When the inlet temperature of the saline wastewater increased to 333 K, the liquid volume fraction at the outlet was less than 30%. Besides, when the inlet temperature increased to 343 K, the liquid film disappeared at 930 mm from the inlet. This phenomenon is due to the increase in the evaporation rate.

Figure 7 | Axial/radial velocity distribution at different inlet temperatures.

Figure 8 | Temperature distribution in evaporation tube. (a) Temperature distribution within 2 mm close to the wall when the inlet temperature is 323 K. (b) Axial temperature distribution at 1 mm near the wall under different inlet temperatures. (c) Radial temperature distribution at 500 mm under different inlet temperatures.

Figure 9 | Distribution of liquid film thickness and liquid volume fraction. (a) Distribution of liquid film thickness with tube length. (b) The liquid volume fraction at the axial distance of 100/600 mm from the inlet at different inlet temperatures.
3.2.3. Effect of inlet concentration on heat and mass transfer in tube

Figure 10 shows the variation in the axial/radial velocities at different inlet concentrations. The inlet concentration of the saline wastewater increased from 2 to 6%, which caused the axial velocity to decrease in the range of 40–1,000 mm from the inlet. This could be explained by a greater viscosity.

Combining Figure 11(b) and 11(c), it could be seen that the inlet concentration of the saline wastewater increased from 2 to 6%, resulting in an increase in the temperature of the liquid film in the range of 200–700 mm. This was mainly due to the increase in heating time caused by the decrease in film flow rate. In addition, in the range of 300–900 mm from the inlet, when the inlet concentration increased to 8%, the temperature of the liquid film was slightly lower than the temperature of 2%. The main reason was that the thermal conductivity of saline wastewater with an inlet concentration of 8% is lower than that of 2%.

It could be seen from Figure 12(a) and 12(b) that when the inlet concentration of the saline wastewater increased from 2 to 6%, the thickness of the liquid film changed little but more stable. The main reason was that the viscosity of saline wastewater had increased. However, when the salt concentration increased to 8%, the thickness of the liquid film exceeded the thickness when the salt concentration was 2 and 6%.

3.3. Convective heat transfer coefficient

3.3.1. Effect of inlet velocity on the convective heat transfer coefficient in tube

Under the condition of the inlet temperature of the saline wastewater of 313.15 K and the inlet concentration of the saline wastewater of 2%, the influence of inlet velocity on convective heat transfer coefficient is presented in Figure 13(a). The convective heat transfer coefficient first increased, then decreased, and reached the minimum at the outlet, which changed in the range of about 2,500–12,500 W/m² K. This phenomenon was due to the film flow rate, which was small at the inlet. The film flow rate in the middle section of the evaporation tube was relatively large, which led to a large local convective heat transfer coefficient. However, due to the increase in

![Figure 10](http://iwaponline.com/jwrd/article-pdf/doi/10.2166/wrd.2021.014/904336/jwrd2021014.pdf)

**Figure 10** | Axial/radial velocity distribution at different inlet concentrations.

![Figure 11](http://iwaponline.com/jwrd/article-pdf/doi/10.2166/wrd.2021.014/904336/jwrd2021014.pdf)

**Figure 11** | Temperature distribution in evaporation tube. (a) Temperature distribution within 2 mm close to the wall when the inlet concentration is 4%. (b) Axial temperature distribution at 1 mm near the wall under different inlet concentrations. (c) Radial temperature distribution at 500 mm under different inlet concentrations.
thermal resistance caused by the increase in the volume fraction of gas in the liquid film, the closer to the outlet of the tube, the smaller the local convective heat transfer coefficient.

In addition, the increase of the inlet velocity led to the overall increase of the local convective heat transfer coefficient in the tube. And the maximum value appeared closer to the outlet: when the inlet velocity was 0.175 m/s, the average convective heat transfer coefficient reached 9,313 W/m² K, which was undoubtedly beneficial to heat transfer. This was due, on the one hand, to the wettability of the evaporation surface which increased with the increase of liquid film thickness and, on the other hand, to the decrease in the temperature of the liquid film which led to an increase in the heat flux.

The effect of the inlet temperature on the convective heat transfer coefficient during the evaporation processes is shown in Figure 13(b) under the condition of the inlet velocity of 0.1 m/s and the inlet concentration of 2%. It could be seen from Figure 13(b) that the convective heat transfer coefficient increased (0–350 mm) and then decreased (350–1,000 mm) with the increase of the inlet temperature of the saline wastewater. The main reason was that in the range of 0–350 mm, the increase in temperature reduced the viscosity of saline wastewater, thereby increasing the film flow rate, which enhanced heat transfer. In addition, in the range of 350–1,000 mm, as the temperature of the saline wastewater increased, the temperature difference as the driving force for heat transfer decreased.

Figure 13(c) shows the effect of the inlet concentration on the convective heat transfer coefficient in the tube. The other fixed operating parameters were the inlet temperature of 313.15 K and the inlet velocity of 0.1 m/s. As shown in Figure 13(c), when the inlet concentration increased from 2 to 8%, the convective heat transfer coefficient decreased. It was worth noting that in the range of 520–1,000 mm, the convective heat transfer coefficient at an inlet concentration of 8% was slightly higher than that at 6%. The main reason was that, on the one hand, the film flow rate was the largest when the inlet concentration was 2%. On the other hand, in the range of 0–520 mm, when the inlet concentration was 8%, the thickness of the liquid film had a greater influence on the heat transfer effect, while in the range of 520–1,000 mm, the film flow rate had a greater impact on the heat transfer effect.
4. EXPERIMENTAL SETUP AND RESULTS

The experimental device of falling film evaporation is shown in Figure 14. The saline wastewater flowed downward under the action of gravity and is distributed on the inner wall of a vertical glass tube with a diameter of 40 mm and a length of 1,000 mm. It took a certain time for the falling film evaporation process to be stable. To ensure the accuracy of the experimental data, the pre-experiment was carried out before recording the data. And all the equipment was adjusted to a reasonable range of application.

The measured data were used to calculate the local convective heat transfer coefficient, which would be used to compare with the simulated value. It can be seen from Figure 15 that the trend of the simulated value was the same as the experimental value. And the average error was 9.27%.

5. CONCLUSIONS

In this paper, the VOF model was used to simulate the flow of saline wastewater in the vertical falling film evaporation tube under certain structural parameters and different operating parameters. The main conclusions were as follows:

1. The inlet velocity of the saline wastewater had a small effect on the film flow rate. However, its increase led to the decrease of temperature in the whole tube and the increase of liquid film thickness and convective heat transfer coefficient.
2. With the increase of the inlet temperature of the saline wastewater, the film flow rate and the temperature of the whole tube increased. Besides, the biggest convective heat transfer coefficient was obtained at low inlet temperature.
3. With the increase of the inlet concentration of the saline wastewater, the film flow rate first decreased and then increased, but the temperature in the overall tube increased first and then decreased. In addition, at relatively
low concentration, the change in concentration has little effect on the average convective heat transfer coefficient.

4. The average error between the experimental and simulated results was 9.27%.

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CONFLICT OF INTEREST

The authors declare no competing financial interest.

DATA AVAILABILITY STATEMENT

All relevant data are included in the paper or its Supplementary Information.

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