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Performance Evaluation of a Direct-Indirect Two-Stage Evaporative Cooler: A Numerical Approach

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Abstract

This work presents a comprehensive numerical study of a direct-indirect two-stage evaporative cooling system. Utilizing advanced mathematical models and MATLAB simulations, the research evaluates the cooler's performance under varying environmental conditions, particularly different air inlet velocities. The model's accuracy is validated through empirical data, showing minimal errors. The study reveals that higher air velocities significantly enhance temperature reductions and humidity ratio changes, improving overall system effectiveness. Performance evaluations in regions like Rajshahi, Dhaka, and Khulna, Bangladesh, indicate the direct-indirect cooler outperforms traditional single-stage systems, achieving the effectiveness of wet bulb range of 98%-115% and dew point effectiveness of 60%-70%. The maximum temperature drop observed was 27.7C, depending on the weather conditions. The research underscores the cooler's suitability for hot and arid climates, suggesting its potential for sustainable building designs. Future recommendations include exploring diverse climates, integrating smart control systems, using renewable energy, and improving material science for greater efficiency and practicality.

Keywords: DI evaporative cooler, inlet air velocity, wet bulb effectiveness, dew point effectiveness.

1.0 Introduction

The direct-indirect (DI) evaporative cooler represents a significant advancement in cooling technology, offering enhanced efficiency compared to traditional single-stage systems. Its use of water as a cooling medium makes it an eco-friendly alternative to conventional air conditioning, reducing reliance on refrigerants and lowering carbon emissions. This multi-stage approach enables the cooler to achieve lower air temperatures than typical single-stage evaporative coolers, while also introducing less humidity into the internal environment in comparison to a single-stage system [1]. The indirect cooling stage is essential because its pre-cools the air without introducing moisture, enhancing the system's ability to lower indoor temperatures while managing humidity levels. This is especially advantageous in regions where high humidity can be unpleasant. Evaporative cooling has been extensively researched and various studies have highlighted its effectiveness and applications [2].

Many scholars have carried out a great deal of research on evaporative coolers. Aimiuwu et al. [3] observed that the temperature of water in a porous ceramic container is below the ambient temperature over the long run. In an evaporatively cooled chamber that was box-shaped, Taha et al. [4] noticed a temperature decrease. Musa et al. [5] investigated the response in transient conditions of an evaporative cooling chamber (ECC) to fluctuations in ambient relative humidity and temperature throughout both dry and rainy seasons. The typical leaf temperature of a cotton plant that is well-watered was determined by Upchurch et al. [6]. While, Prange [7] discovered that insects employ evaporative cooling as a mechanism for thermoregulatory issues. Kassem [8] undertook both theoretical and practical research to enhance the efficiency of water evaporation in the pad. Ghiabaklou et al. [9] suggested a system that capitalizes on the evaporative cooling action of water. With a 30% rise in the relative humidity of the incoming air, Ibrahim et al. [10] observed dry bulb temperature decreases of 6-8°C. The packing material utilized by Dai and Sumathy [11] was honeycomb paper, which served as the medium for cooling the air stream. Zalewski and Gryglaszewski [12] created a heat and mass transport model for an evaporative cooler. Meanwhile, Armbruster and Mitrovic [13] established relationships for the Nusselt number and Song et al. [14] determined that the cooling impact is considerably enhanced by evaporation. Jamil et al. [15] developed an energy-efficient indirect evaporative cooling system (ECS) with a heat transfer coefficient correlation matching experimental data (±5%), while Chea et al. [16] designed a solar module with ECS with experimental validation to forecast annual performance. Moreover, Kapilan et al. [17] reviewed a comprehensive evaporative cooling system, highlighting how combining direct and indirect ECS can reduce air temperature below its wet bulb temperature.

From the above discussion, several studies have been performed on single-stage evaporative coolers. As per the literature, there is little to no research on the direct-indirect two-stage system of evaporating coolers. Consequently, the target of this work is to conduct a numerical analysis of the direct-indirect two-stage cooling system of evaporative coolers through the use of MATLAB simulations. Moreover, the investigation of DI cooler performance in a wider range of climatic conditions and integration of the smart systems for controlling the unit in order to enhance the operations.

2.0 Methodology

Evaporative coolers are essential devices in hot climates for providing efficient and eco-friendly cooling solutions. A direct-indirect evaporative cooler operates on a continuous cycle to reduce the temperature of incoming air, enhancing indoor comfort. The primary purpose of this cooler is to maximize cooling effectiveness by combining direct and indirect evaporative cooling techniques. The basic operation of this system is illustrated in Fig. 1, where the cooling pads and heat exchangers play crucial roles in delivering the cooling effect. To achieve this goal, the heat transfer characteristics of the cooling pads and heat exchangers will be numerically investigated.

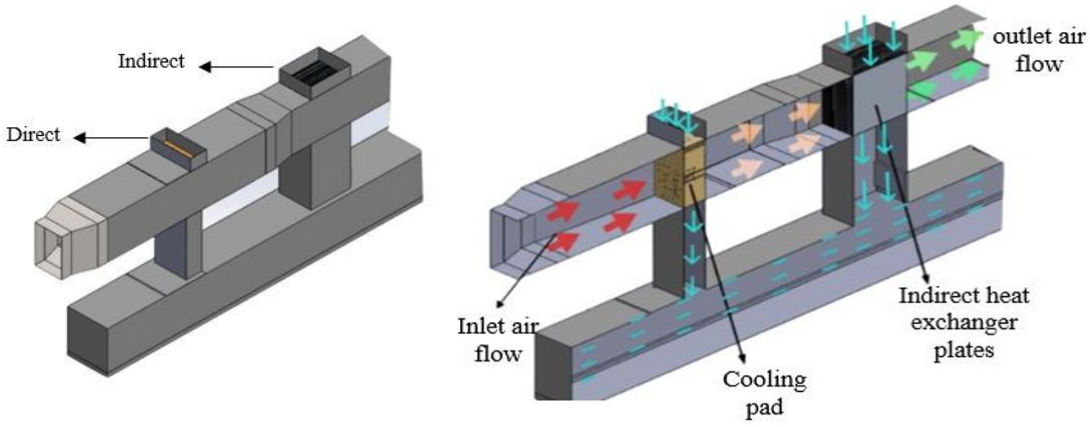


Fig. 1. (a) CAD model (b) cross-sectional view of DI evaporative cooler

2.1 Numerical model

This paper aims to derive the numerical model for predicting the temperature changes of the air within a DI evaporative cooler. The air and water properties are listed in Table 1. By discretizing the governing equations and applying numerical integration techniques, we can simulate the cooling performance under various operating conditions. The derived model will provide insights into the temperature distribution and help in designing more efficient and effective DI evaporative coolers.

Table 1. Air and water properties

Properties	Values	Properties	Values
Air specific heat capacity, J/kgK	1006	Water specific heat capacity, J/kgK	4181
Thermal conductivity, W/m^{2}K	0.0263	Water vapor specific heat capacity, J/kgK	1854
Density, kg/m^3	1.138	Latent heat of vaporization, J/kg	2.30×10^6

2.2 Governing equations

Differential equation of humidity ratio: $\frac{\partial \omega_a}{\partial x} = h_d \frac{\beta L_z L_y}{m_a} (\omega_{sw} - \omega_a)$ (1)

Differential equation of air temperature: $\frac{\partial T_a}{\partial x} = \frac{\beta L_z L_y (T_w - T_a)}{m_a (C_{pa} + \omega_a C_{pv})} \left(C_{pv} h_d (\omega_{sw} - \omega_a) + h_a \right)$ (2)

Differential equation of water temperature: $\frac{\partial T_w}{\partial y} = \frac{\beta L_z L_x}{C_w m_w} \left(h_d (\omega_a - \omega_{sw})(H_v - T_w C_w) + h_a (T_a - T_w) \right)$ (3)

$\frac{dT_p}{dx} = -\frac{U_Z a (T_p - T_W)}{m_P C_P}$ where, $U_Z = \frac{1}{\frac{1}{h_p} + \frac{\delta_{wall}}{k_{wall}} + \frac{1}{h_W}}$ (4)

The finite difference approach was implemented in this instance to resolve Eqs. (1), (2), (3), and (4). In order to achieve this objective, a code was created in the MATLAB program using the finite difference method. The heat and mass transfer coefficients were determined by inputting all the appropriate input data, such as the thermodynamic parameters of the water and air and also the boundary conditions, into the code. Initially, the computational nodes estimated the water temperature. Then, Eqs. (1), (2), and (4) were resolved in sequence. The water-falling film temperature at the computational nodes, particularly at the cooler outlet, was determined by solving Eq. (3) using the humidity ratio and air temperature data acquired from all computing nodes.

2.3 Validation

In an effort to ensure that the results were not contingent upon the computational nodes, the cooler output air temperature was computed for 100, 200, 300, 400, and 500 nodes which are presented in Table 2. To avoid repetitive figures, we have incorporated the temperature variation, air temperature profile, and humidity ratio profile only for the 400 nodes. Due to the consistency of the outlet temperature, which is illustrated in Fig. 2, 400 computing nodes were employed to do all calculations. Nevertheless, the evaporative cooler is estimated to have a pad heat transfer area to volume ratio (β) of 370, with dimensions of 0.152 m × 0.336 m × 0.61 m.

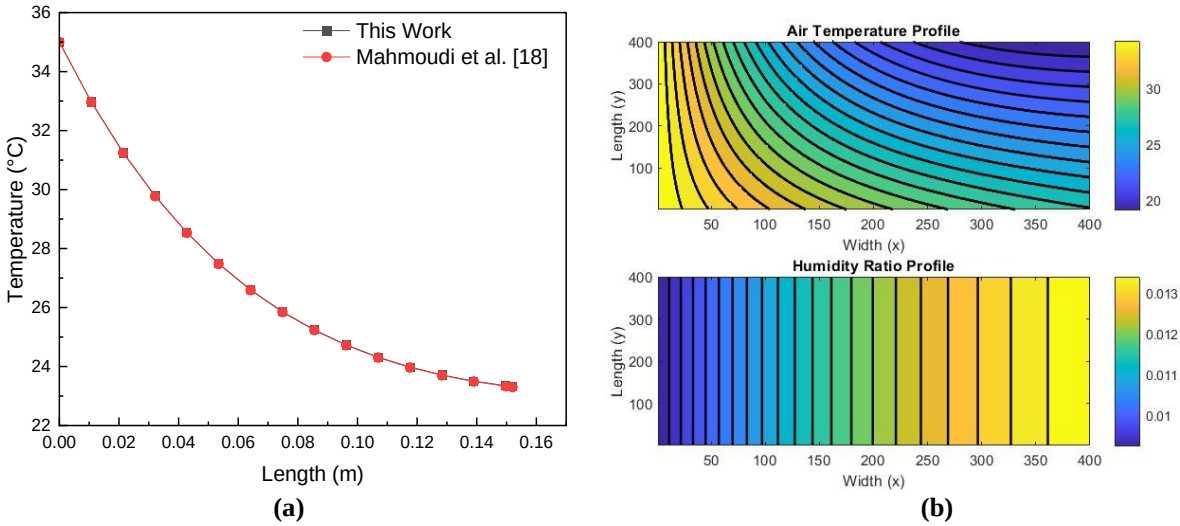


Fig. 2. Validation for a direct evaporative cooler of (a) temperature variation and (b) temperature and humidity ratio profile at 400 nodes

Table 2. Direct evaporative cooler validation for dimensions 0.152m×0.336m×0.610m

Number of nodes	Outlet temperature	Outlet temperature (Mahmoudi et al. [18])
100	23.2857	23.2912
200	23.2875	23.2965
300	23.2901	23.2982
400	23.316	23.2990
500	23.313	23.2995

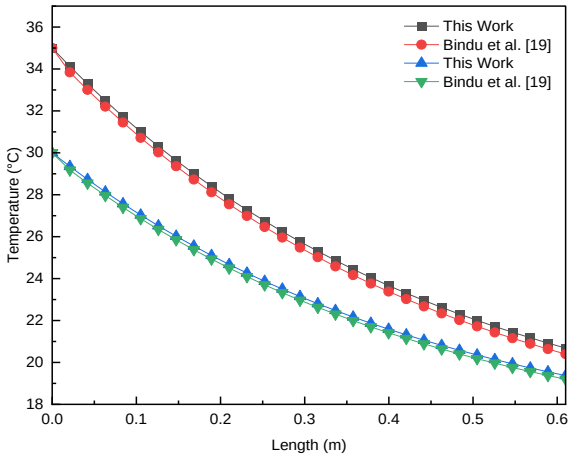


Fig. 3. Validation for indirect evaporative cooler [19].

The validation for an indirect evaporative cooler is illustrated in Fig.3. To validate the numerical model of the indirect evaporative cooler, specific parameters were employed to mirror real-world conditions. The setup involved 40 plates, each measuring 0.6 meters in length, 0.4 meters in width, and 0.4 mm in thickness. The primary air was introduced at a velocity of 1 m/s, while the secondary air moved through the cooler at 0.5 m/s. The channel

width between the plates was maintained at 4 mm to ensure proper airflow and heat exchange. The validation results demonstrated an error percentage of up to 1.36%, which is within an acceptable range for engineering applications. This level of accuracy underscores the reliability of the numerical model in predicting the performance of indirect evaporative coolers under specified conditions.

3.0 Results and discussion

Numerical simulation was performed to analyze the effectiveness of the DI evaporative cooler by changing the inlet air velocity, and outdoor air temperature and analyzed the cooling contribution of stages where the outdoor temperature was 45°C, 40°C, 35°C, and inlet velocity was 2 m/s. Again, the temperatures 30°C and 50°C were not analyzed as 30°C is near comfort temperature and 50°C is not suspected to be higher, according to Bangladesh. Moreover, the simulation was done by analyzing yearly data of different temperatures.

3.1 Impact of inlet air velocity

Fig. 4 shows the impact of inlet air velocity when outdoor temperatures at 45°C, 40°C, and 35°C, respectively. As air inlet velocity changes in a certain range, outlet temperature decreases because as the velocity increases, the mass flow rate of air also increases. Due to the increase in velocity higher air inlet velocity increases the rate at which air flows through the cooling media (e.g., wet pads).

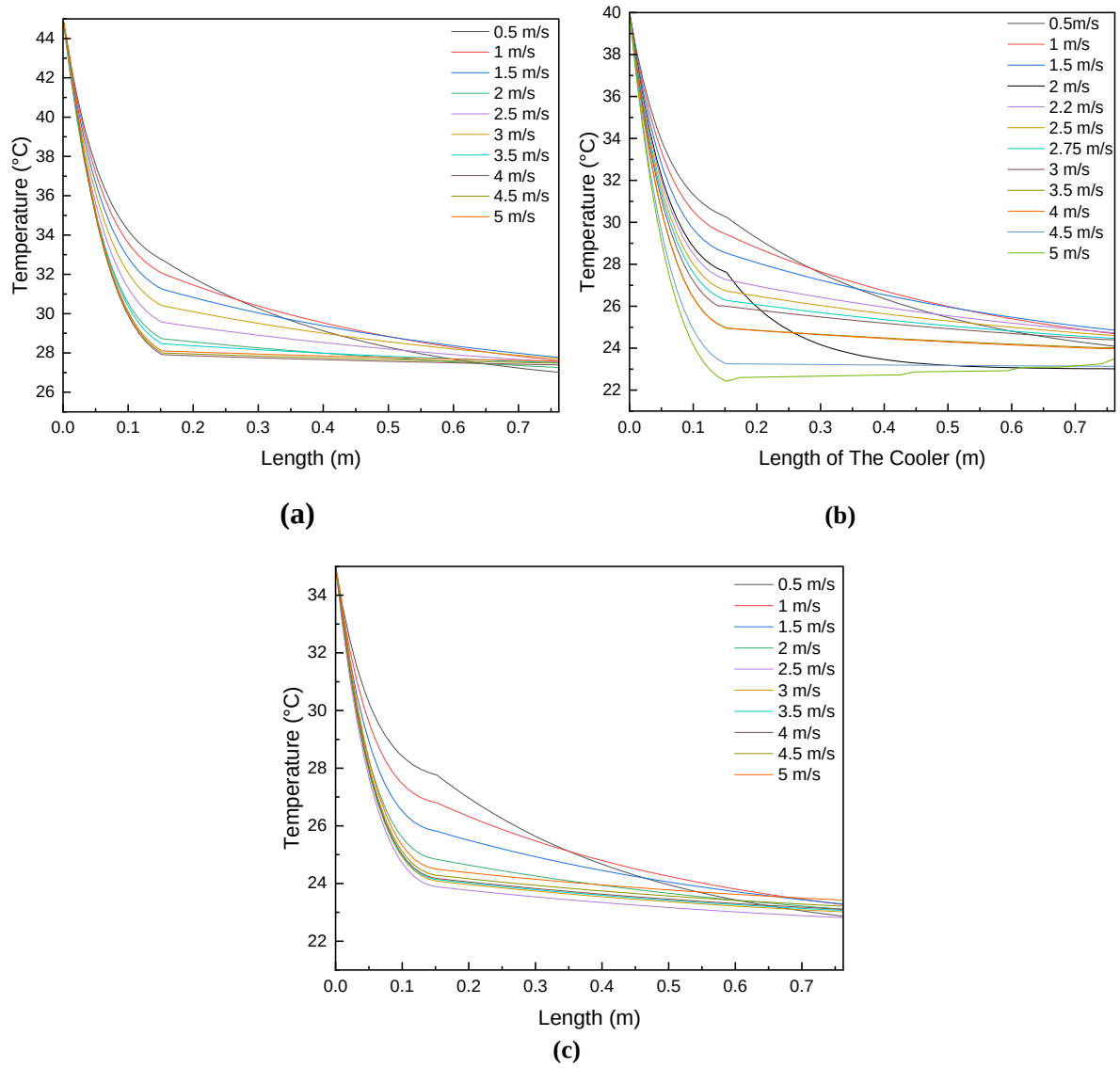


Fig. 4. Impact of inlet air velocity when outdoor temperature at: (a) 45°C (b) 40°C, and (c) 35°C

The faster flow of air allows for more effective heat transfer between the air and the water in the cooling media, resulting in a more efficient evaporation process. Table 3 shows that when outdoor temperature increases, the

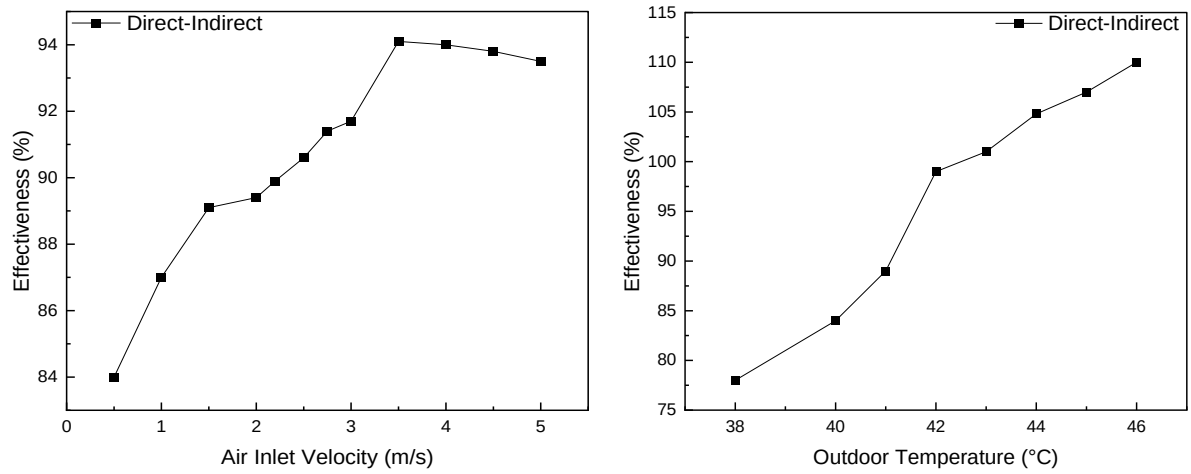
maximum temperature drop takes place at lower inlet velocity. At higher outdoor temperatures, the air can hold more moisture, thus allowing more water to evaporate and result in a greater cooling effect. Lower inlet velocity at high temperatures allows the air to stay in contact with the evaporative media longer and maximizes the cooling effect. At lower outdoor temperatures, the air is closer to saturation, reducing the cooling potential. A higher inlet velocity then be optimal to balance the residence time and the cooling effect. The output temperature was minimum at 1.5 m/s, 2 m/s and 2.5 m/s when the outdoor temperature was 45 ℃, 35℃, 30℃.

Table 3. Impact of air inlet velocity on output temperature

Velocity (m/s)	Output Temp. (℃)		
	Input air temp. (45℃)	Input air temp. (40℃)	Input air temp. (35℃)
0.5	27.01	24.09	22.86
1.0	27.60	24.68	23.27
1.5	27.77	24.85	23.28
2.0	27.25	23.01	23.10
2.5	27.52	24.59	22.81
3.0	27.71	24.40	23.01
3.5	27.50	24.02	23.06
4.0	27.39	23.98	23.10
4.5	27.48	23.13	23.21
5.0	27.58	23.50	23.42

3.2 Impact of air inlet velocity and outdoor temperature on effectiveness

Fig. 5 shows the variation of effectiveness with outdoor air at temperature of 40 °C and air velocity of 2 m/s, respectively. As the velocity was increased, effectiveness of direct-indirect evaporative cooler was increased. As the velocity was increased, it increased the mass flow rate which increased the evaporation rate. But in indirect evaporative cooler, effectiveness was decreased with the increase of velocity due to less time in contact. The DI evaporative cooler's effectiveness was maximum at velocity 2 m/s when the outdoor temperature is 40℃. As the outdoor temperature was increased, effectiveness of the evaporative cooler increased because of the temperature difference between water and outdoor temperature was increased.



(a) **(b)**

Fig. 5. Variation of effectiveness with outdoor air conditions at (a) temperature of 40 °C (b) velocity of 2 m/s

3.3 Performance of different evaporative coolers in hot weather

Fig. 6 shows the maximum temperature variation (Rajshahi) according to month for the proposed model along with outdoor air temperature as in hot weather. From March to August is considered as hot weather. Using different types of evaporative coolers, maximum temperature drop was possible by the use of a DI evaporative cooler. The temperature dropped down in the human comfort range which was quite satisfactory. In April, May, and June the temperature was above 44°C to 46°C. Using a direct evaporative cooler, the temperature was dropped down to 31.5°C-32°C. Using a direct-indirect (DI) evaporative cooler it was dropped down to 27.2°C-27.7°C. It is evident that the two-stage DI evaporative cooler is more effective than a single-stage direct evaporative cooling system.

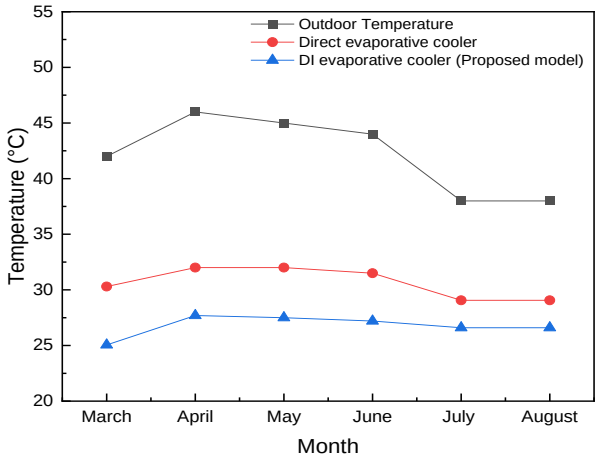


Fig. 6. Temperature variation according to month for the proposed model along with outdoor air temperature.

4. Conclusion

In this study, the numerical model developed and simulations conducted in MATLAB showed minimal percentage errors when compared to empirical data, validating the accuracy of the model. The specific findings are as:

- The DID cooler achieved a wet bulb effectiveness range of 98%-115% and a dew point effectiveness range of 60%-70%.
- The maximum temperature drop observed was 27.7C, depending on the weather conditions.
- The DI cooler not only achieves substantial temperature reductions but also manages to operate efficiently across various air inlet velocities and outdoor temperatures.

5. Acknowledgement

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