

Heat transfer enhancement of a translating plate impinged by dual-slot nozzles with integrated multiple jets: A comparative study of single- and double-row configurations

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ABSTRACT

The thermal-fluid-structural behavior of a translating aluminum plate in an air-cushion furnace equipped with dual-slot nozzles integrated with multiple jets is numerically investigated, focusing on the effect of the jet-row number. Three configurations are considered: dual-slot without multiple jets (DS-NoMJ), with a single-row of multiple jets (DS-SRMJ), and with double rows of multiple jets (DS-DRMJ). The results show that increasing the jet-row number enhances convective heat transfer but significantly modifies the pressure distribution and deformation characteristics. The average Nusselt number increases by about 18% and 27% for DS-SRMJ and DS-DRMJ, respectively, compared with DS-NoMJ. However, temperature uniformity decreases as the number of jet rows increases, with the temperature uniformity index decreasing from 0.757 (DS-NoMJ) to 0.703 (DS-DRMJ). With DS-NoMJ and DS-SRMJ, the plate exhibits a stable wave-like deformation pattern with small amplitudes (7.39 mm and 8.59 mm), associated with symmetric recirculation and gradual pressure redistribution. In contrast, DS-DRMJ produces disordered vortical structures and localized stagnation pressure concentration, leading to high pressure difference between the upper and lower air cushions and severe central bulging of the plate, with maximum deformation reaching 23.74 mm. The single-row multiple-jet configuration provides the best compromise between heat transfer enhancement and aerodynamic stability, achieving intensified heat transfer while maintaining acceptable temperature uniformity and small deformation. These results clarify the role of the number of jet rows in coupling flow organization, pressure loading, and structural response.

1. Introduction

Impinging jet technology has been extensively employed in industrial thermal processes owing to its capability to deliver high convective heat transfer coefficients and controllable thermal environments [1]. Typical applications include metal heat treatment, drying of paper and textiles, cooling of turbine components, and thermal conditioning of continuous products [2]. The superior heat transfer performance of impinging jets is primarily attributed to the formation of stagnation regions, strong shear layers, and intense turbulence in the near-wall region [3].

Among various impinging jet configurations, slot jets have received particular attention for applications involving large-area or continuously moving surfaces [4]. Compared with round jets, slot jets provide

elongated stagnation regions and improved spanwise uniformity of heat transfer, which is advantageous for processing flat plates and sheets [5]. When applied to non-contact heat treatment systems such as air-cushion furnaces, slot jets are also capable of generating stable aerodynamic support while simultaneously enhancing the convective heat transfer [6].

To further improve suspension stability and thermal uniformity, dual-slot nozzle configurations have been proposed and investigated. By employing two inclined slot jets, the pressure distribution on the plate surface can be balanced, and the interaction between wall jets and recirculating flows can be effectively controlled [7]. Previous studies have demonstrated that parameters such as nozzle inclination angle, jet-to-plate spacing, Reynolds number (Re), and plate translational speed strongly influence the flow structure, heat transfer characteristics, and deformation behavior of the processed plate [8]. Nevertheless, under

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Nomenclature			
<i>List of symbols</i>		V	Plate velocity (m/s)
A	Area of the heated surface (m ²)	s	Nozzle-to-nozzle spacing (mm)
b	Nozzle slot spacing (mm)	x	Coordinate in x direction (mm)
c_p	Pressure coefficient	x_i	Coordinate direction
$c_{p,avg}$	Average pressure coefficient	y	Coordinate in y direction (mm)
C_μ	Empirical constant	y^+	Non-dimensional distance between wall and first-layer nodes
c	Specific heat (J/kg·K)	z	Coordinate in z direction (mm)
D_h	Hydraulic diameter (mm)	TUI	Temperature uniformity index
dl	Distance between lower nozzles and plate (mm)	SD	Standard deviation of the surface temperature
du	Distance between upper nozzles and plate (mm)	Δp	Pressure difference between the upper and lower plate surfaces (Pa)
e	Nozzle slot width (mm)	<i>Abbreviations</i>	
h	Jet-to-plate spacing (mm)	DS-NoMJ	Dual-slot nozzle without multiple jets
K_m	Fluid viscosity coefficient of turbulent kinetic energy	DS-SRMJ	Dual-slot nozzle with single-row multiple jets
K_T	Fluid temperature coefficient	DS-DRMJ	Dual-slot nozzle with double-row multiple jets
k	Turbulent kinetic energy (J)	CFD	Computational Fluid Dynamics
k_f	Fluid thermal conductivity (W/m·K)	FSI	Fluid-Structure Interaction
Nu_{avg}	Average Nusselt number	SST	Shear Stress Transport
Nu_x	Local Nusselt number	UDF	User-Defined Function
P	Average pressure (N/m ²)	<i>Greek Symbols</i>	
p_c	Wall pressure (N/m ²)	Ω_{ij}	Angular velocity tensor (1/s)
Pr	Prandtl number	α	Jet flow inclination angle (°)
Re	Reynolds number	δ_{ij}	Kronecker delta
R_{pj}	Plate-to-jet velocity ratio	ε	Turbulence kinetic energy dissipation rate (1/s)
Sc	Schmidt number	μ	Dynamic viscosity of air (kg/m·s)
S_{ij}	Angular deformation rate tensor (1/s)	ρ	Air density (kg/m ³)
T	Average temperature (K)	ν	Kinematic viscosity of air (m ² /s)
T'	Fluctuating temperature components (K)	σ	Normal stress component (Pa)
T_j	Jet temperature (K)	τ	Shear stress component (Pa)
T_w	Wall temperature (K)	ω	Specific dissipation rate (1/s)
t	Time (s)	λ	Thermal conductivity (W/m·K)
U_i	Average velocity (m/s)		
u_i'	Fluctuating velocity components (m/s)		
u_j	Mean jet exit velocity (m/s)		

practical industrial constraints, the achievable heat transfer enhancement of conventional dual-slot nozzle systems remains limited.

In parallel, extensive research has been conducted on multiple-jet impingement systems, where arrays of discrete jets are employed to intensify turbulence and promote jet-to-jet interaction. Multiple jets can significantly enhance local and area-averaged Nusselt numbers (Nu) compared to single-jet configurations, owing to increased entrainment, vortex interaction, and turbulent mixing. Badra et al. [9] concluded that the optimum dimensionless jet-to-plate distance was in the range of 6–7. Caliskan et al. [10] investigated elliptic and rectangular impinging jet arrays and revealed that elliptic jets yield higher local Nu in the impingement region compared to circular jets, which they attributed to enhanced entrainment and coherent structures. Li et al. [11] studied impingement of inline and staggered multiple round jet arrays on a flat plate. They showed that staggered arrangements generally produced higher area-averaged Nu than inline configurations due to stronger jet-to-jet interaction and enhanced turbulent mixing in the wall region. Choi et al. [12] reported that reducing the jet-to-jet spacing intensified flow interaction and increased local heat transfer, while excessively small spacing led to flow interference and deterioration of overall performance. Selimefendigil and Oztop [13] proposed a three-dimensional configuration using enhanced multiple slot jets, providing new insights into the synergistic effect of fluid additives and jet arrays.

Yao et al. [14] investigated heat transfer enhancement mechanisms in multi-jet impingement systems and demonstrated that strong jet-to-jet interaction and vortex-ring dynamics play a dominant role in determining both local and global heat transfer performance. Their

results showed that intensified turbulent mixing and near-wall flow unsteadiness can significantly amplify the wall-normal temperature gradient, leading to substantial increases in average Nu compared to non-enhanced configurations. Singh et al. [15] investigated confined multiple jet impingement using PIV (particle image velocimetry) and heat flux measurements and showed that enhanced jet interaction and turbulence development significantly improve heat transfer performance. Their results indicated that the turbulence intensity near the impingement region increased by approximately 25%, leading to a corresponding Nu enhancement of about 25% compared to that of a flat-plate configuration, particularly in regions dominated by jet-to-jet interaction.

The heat transfer enhancement obtained with multiple-jet configurations does not increase monotonically with jet number or jet-row count [16,17]. Excessive jet interaction may lead to flow interference, weakened impingement strength, and non-uniform heat transfer distributions, particularly under confined conditions. Zhou et al. [18] numerically investigated the cooling performance of single-row jet impingement under the influence of cross flow in a confined target channel. Their results showed that although variations in jet diameter had a limited effect on overall heat transfer levels, optimizing the jet diameter distribution significantly improved heat transfer uniformity, reducing the non-uniformity coefficient of peak spanwise-averaged Nu from about 9–12% to 3–4%. Xi et al. [19] numerically compared the heat transfer and flow characteristics of single-jet impingement with converging, straight, and expanded hole configurations. Their results indicated that converging holes provide the highest heat transfer

performance, with the average Nu increasing by approximately 1.6–2.6 times as the Re increases from 6,000 to 30,000, whereas this enhancement is accompanied by high pressure loss, highlighting the inherent trade-off between heat transfer augmentation and flow performance in single-jet impingement systems.

Existing studies on multiple jets have mainly focused on standalone jet arrays or simplified geometries, emphasizing the effects of jet spacing, arrangement, and row number on heat transfer, while largely neglecting slot-based nozzle systems, in which strong interactions between slot jets and discrete multiple jets may fundamentally alter the flow structure and heat transfer mechanisms. Meanwhile, the effect of integrating additional multiple jets into dual-slot nozzles for performance enhancement has not been adequately evaluated. While standalone jet arrays can significantly intensify heat transfer, their interactions with the primary slot flow in an air-cushion furnace may induce complex flow interference, which potentially compromises the suspension stability of a thin and translating plate. Moreover, previous studies on the jet-row number indicate that heat transfer is not enhanced monotonically with increasing jet-row number, and excessive jet interaction may lead to flow interference and performance degradation, highlighting the necessity of configuration-specific evaluation. In practical applications of the air-cushion furnace, integrating multiple jets into an existing dual-slot nozzle offers a promising strategy to intensify heat transfer without substantially modifying the overall system layout. However, the interaction between slot jets and centrally arranged multiple jets, as well as the role of the jet-row number in determining heat transfer enhancement and flow stability, have not been systematically investigated.

To fill the research gap, the present study numerically and experimentally investigates the heat transfer characteristics of a translating flat plate impinged by a dual-slot nozzle integrated with multiple jets. The novelty lies in revealing the function assumed by the jet-row number in coupling flow organization, pressure loading, and structural response, which has been largely neglected heretofore. Based on the authors' previous investigation on dual-slot nozzle impingement systems for translating plates [20], three nozzle configurations are comparatively studied, i.e., a conventional dual-slot nozzle without multiple jets, a dual-slot nozzle with a single row of multiple jets, and a dual-slot nozzle with double rows of multiple jets. The objectives of the present study are to (i) quantitatively evaluate the heat transfer enhancement achieved by introducing multiple jets into a dual-slot nozzle system, (ii) elucidate the influence of the jet-row number on heat transfer gain and spatial distribution, and (iii) assess the critical trade-off between heat transfer intensification and aerodynamic stability to ensure acceptable temperature uniformity and minimal plate deformation. The obtained conclusions are expected to provide insights for advanced air-cushion furnace systems and other industrial impinging jet applications.

2. Experimental methodology

2.1. Experimental setup

An experimental facility was developed to investigate the streamwise distribution of Nu along a flat plate subjected to jet impingement. A schematic of the experimental apparatus is shown in Fig. 1(a). The system mainly consists of an air supply unit, a heating and flow-conditioning section, a nozzle assembly, a positioning system, and a data acquisition unit, allowing flexible adjustment of operating conditions through independent control of each module.

Air is supplied by a centrifugal fan with a rated volumetric flow rate of 9.5 m³/min and a rated power of 240 W. The fan speed is regulated using a variable-frequency drive to achieve the desired flow rate. The airflow is heated by two electrical heaters, each with a rated power of 3 kW, before passing through a rectangular duct equipped with flow-straightening screens to ensure a uniform and stable inlet flow.

The nozzle assembly consists of a cubic air duct of dimensions 100 × 100 × 100 mm and a dual-slot nozzle mounted downstream. The nozzle cross-section and airflow direction are illustrated in Fig. 1(b), while the key geometric parameters are summarized in Table 1. To promote a uniform velocity distribution at the slot exits, two rows of air distribution holes are arranged inside the nozzle.

Precise positioning of the nozzle relative to the target plate is achieved using a multi-axis positioning system driven by a high-precision servo motor with a positioning accuracy of ±0.05 mm. The system includes a servo-controlled lifting platform, a horizontal translation stage with a maximum travel of 400 mm, and a target aluminum plate with dimensions 200 × 200 × 20 mm. This configuration allows accurate adjustment of the nozzle–plate alignment and impingement distance.

Air temperature within the duct is monitored using a PT100 platinum resistance temperature detector (RTD) with an accuracy of ±0.15 °C. Four additional PT100 RTDs are symmetrically embedded in the aluminum plate to measure the plate temperature. Local heat flux during jet impingement is recorded using a high-precision heat flux sensor (gSKIN XM269C, Switzerland) with dimensions 4.4 × 4.4 × 0.5 mm, which is flush-mounted on the plate surface. A photograph of the experimental setup is shown in Fig. 1(c). The air temperature is regulated by a PID control system. The cross-sectional air velocity in the supply duct is measured using a hot-wire anemometer with an accuracy

Table 1

Main geometric parameters of the nozzle.

Parameters	Values
Separation distance (h), mm	0–60
Slot width (e), mm	2
Impinging angle (α), °	45
Length of plate (L_x), mm	200

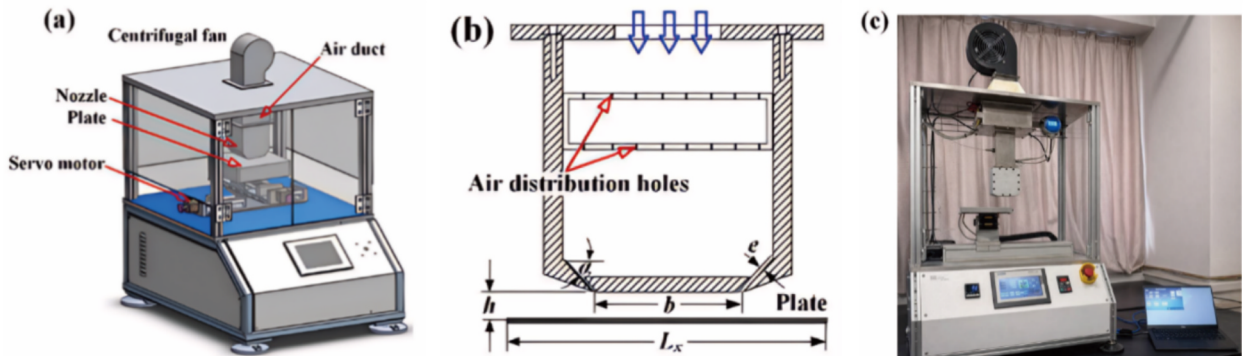


Fig. 1. Experimental rig: (a) schematic view, (b) nozzle cross-section, and (c) photo.

of $\pm 1.0\%$, from which the volumetric flow rate is determined.

To ensure the replicability of the experimental results, main operating parameters adopted in the present study are briefly summarized as follows. The jet Reynolds number is varied within the range of 10,000 to 30,000. The nozzle-to-plate separation is characterized by the normalized spacing (h/e), which is varied between 4 and 8. For the dynamic cases, the translational speed of the plate (V) is adjusted to maintain a plate-to-jet velocity ratio (R_{pj}) between 0.01 and 0.1.

2.2. Measurements of Nu and pressure

The convective heat transfer performance of the impinging jets is evaluated using Nu , which is widely adopted as a dimensionless indicator of heat transfer intensity [21]. As no phase change occurs during the experiments, the aluminum plate is treated as a constant-temperature boundary surface.

The local Nu is calculated based on the heat flux measured by the heat flux sensor according to [7]:

$$Nu = \frac{qD_h}{k(T_w - T_j)} \quad (1)$$

where q is the local heat flux, D_h is the hydraulic diameter, k denotes the thermal conductivity of air, and T_w and T_j represent the wall surface temperature and the reference jet temperature, respectively. The air temperature measured at the nozzle exit is adopted as the reference temperature.

For pressure measurements, the aluminum plate is replaced by a rigid tempered glass plate with dimensions of $200 \times 200 \times 6$ mm. A circular hole of 10 mm diameter is drilled at the measurement location, where a pressure tap is installed. This pressure tap is connected via a flexible tube to the high-pressure port of a HALO differential pressure transmitter. The transmitter has a full-scale measurement range of ± 5 kPa and an accuracy of 0.2%, enabling accurate measurement of the local impingement pressure.

2.3. Uncertainty analysis

An uncertainty analysis was performed following the method proposed by Kline and McClintock [22]. The uncertainty in temperature measurement arises from the combined effects of the PT100 RTD accuracy (± 0.15 °C), data acquisition error (± 0.1 °C), and spatial temperature variation (± 0.2 °C), resulting in a combined standard uncertainty of ± 0.27 °C. With a coverage factor of $k = 2$, the corresponding expanded uncertainty is ± 0.54 °C.

The uncertainty in the calculated Nu is primarily associated with the uncertainty in the temperature difference $\Delta T = T_w - T_j$. For typical experimental conditions where $\Delta T \geq 10$ °C, the relative uncertainty in ΔT is less than 5%. Heat losses are estimated to account for less than 1.5% of the total heat input. Consequently, the overall relative uncertainty in the Nu is evaluated to be $\pm 3.27\%$.

The uncertainty in pressure measurement depends on the accuracy of the differential pressure transmitter (0.2% of full scale), installation errors of the pressure tap, and flow unsteadiness. The combined standard uncertainty of the pressure measurement is estimated to be $\pm 0.38\%$ of full scale, leading to an expanded uncertainty of $\pm 0.76\%$ of full scale.

3. Numerical model

Numerical simulations were performed based on the following assumptions to ensure a balance between computational efficiency and physical fidelity:

- (1) The air was treated as an incompressible Newtonian fluid, and its thermophysical properties were evaluated at the reference jet temperature of 923 K.
- (2) Natural convection effects were neglected, as the high jet Reynolds numbers resulted in a forced-convection-dominated regime.
- (3) Thermal radiation was not explicitly modeled, as the preliminary uncertainty analysis indicated that radiation heat loss contributed less than 1.5% to the total heat transfer under the investigated conditions.
- (4) The aluminum plate was assumed to exhibit linear elastic behaviors during the fluid–structure interaction process.

3.1. Governing equations

The flow and heat transfer characteristics of air jet impingement on a flat plate were simulated by solving the three-dimensional, incompressible Reynolds-averaged Navier-Stokes (RANS) equations coupled with the energy conservation equation. Turbulence effects were modeled using the shear stress transport (SST) $k-\omega$ model, which has been widely demonstrated to provide reliable predictions for impinging jet flows involving strong adverse pressure gradients and near-wall turbulence [23].

The governing equations are formulated based on the conservation of mass, momentum, and thermal energy, and can be expressed as follows [10].

Continuity equation:

$$\frac{\partial U_j}{\partial X_j} = 0 \quad (2)$$

Momentum equation:

$$\rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial X_j} = -\frac{\partial P}{\partial X_i} + \frac{\partial}{\partial X_j} \left[\mu \left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i} \right) - \rho \overline{u_i u_j} \right] \quad (3)$$

Energy equation:

$$\rho \frac{\partial T}{\partial t} + \rho U_j \frac{\partial T}{\partial X_j} = \frac{\partial}{\partial X_j} \left[\frac{\mu}{Pr} \frac{\partial T}{\partial X_j} - \rho \overline{T u_j} \right] \quad (4)$$

where U_i , T , and P denote the time-averaged velocity components, temperature, and pressure, respectively. The fluctuating components of velocity and temperature are represented by u'_i and T' . X_i indicates the Cartesian coordinate directions, ρ is the air density, and μ is the dynamic viscosity. The Prandtl number is defined as:

$$Pr = \mu \frac{c}{k_f} \quad (5)$$

where c is the specific heat capacity at constant pressure and k_f is the thermal conductivity of air.

To close the RANS equations, the Reynolds stresses and turbulent heat flux are modelled using the eddy-viscosity hypothesis:

$$-\rho \overline{u_i u_j} = \mu_t \left(\frac{\partial U_i}{\partial X_j} + \frac{\partial U_j}{\partial X_i} \right) \quad (6)$$

$$-\rho \overline{T u_j} = \alpha_t \frac{\partial T}{\partial X_j} \quad (7)$$

where μ_t and α_t are the turbulent dynamic viscosity and turbulent thermal diffusivity, respectively. The turbulent viscosity is expressed as:

$$\mu_t = \rho C_\mu \frac{k^2}{\epsilon} \quad (8)$$

and the turbulent thermal diffusivity is related to μ_t through the turbulent Prandtl number Pr_t :

$$Pr_t = \frac{\mu_t}{\rho \alpha_t} \approx 0.9 \quad (9)$$

Turbulence effects were modeled using the SST (Shear stress transport) $k-\omega$ turbulence model. The suitability of this model for the present configuration lies in its ability of combining the accurate formulation of the $k-\omega$ turbulence model in near-wall flow region with the free-stream independence of the $k-\epsilon$ turbulence model in the far field. This hybrid model is particularly advantageous for simulating complex interactions between inclined slot jets and discrete multiple jets, which involve strong adverse pressure gradients, local stagnation, and intense shear layers. As demonstrated in the validation study in Section 3.6, the SST $k-\omega$ turbulence model provides higher predictive accuracy for jet impingement heat transfer compared to other commonly used models such as the v^2-f and realizable $k-\epsilon$ turbulence models.

3.2. Heat transfer and pressure characterization

The local convective heat transfer performance is evaluated using the local Nu , defined through the temperature gradient normal to the plate surface [4]:

$$Nu_x = -\left(\frac{D_h}{T_w - T_j}\right)\left(\frac{\partial T}{\partial n}\right)_{z=wall} \quad (10)$$

where D_h is the hydraulic diameter of the nozzle, T_w and T_j denote the wall and jet reference temperatures, respectively, and n represents the unit normal vector pointing outward from the plate surface.

The area-averaged Nu is calculated as:

$$Nu_{avg} = \frac{1}{A} \iint Nu_x dA \quad (11)$$

where A represents the area of the aluminum plate (m^2).

To characterize the pressure distribution on the plate surface, a dimensionless pressure coefficient is introduced [4]:

$$c_p = \frac{2p_c}{\rho u_j^2} \quad (12)$$

where p_c is the local static pressure on the plate surface and u_j is the jet exit velocity. The mean pressure coefficient is obtained by surface averaging:

$$c_{p,avg} = \frac{1}{A} \iint c_p dA \quad (13)$$

The jet Re is defined as:

$$Re = \frac{\rho u_j D_h}{\mu} \quad (14)$$

with D_h calculated according to the geometric dimensions of the nozzle outlet.

The temperature uniformity of the plate surface is quantified through the temperature uniformity index (TUI) [8]:

$$TUI = 1 - \frac{SD}{T_{avg}} \quad (15)$$

where SD is the standard deviation of the surface temperature and T_{avg} is the corresponding mean value. A TUI value approaching unity indicates a highly uniform temperature distribution. TUI represents the degree of thermal deviation across the processing area. A high level of temperature uniformity is essential to achieve consistent metallurgical properties and to prevent the induction of excessive thermal stresses, which could otherwise lead to structural defects or macroscopic deformation of the moving aluminum plate during the heat treatment.

3.3. Structural relations and fluid-structure interaction

To describe the mechanical response of the plate subjected to aerodynamic loading, the equilibrium equations in a three-dimensional Cartesian coordinate system are expressed as [24]:

$$\frac{\partial \sigma_x}{\partial x} + \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{xz}}{\partial z} + X_s = 0 \quad (16)$$

$$\frac{\partial \sigma_y}{\partial y} + \frac{\partial \tau_{xy}}{\partial x} + \frac{\partial \tau_{yz}}{\partial z} + Y_s = 0 \quad (17)$$

$$\frac{\partial \sigma_z}{\partial z} + \frac{\partial \tau_{xz}}{\partial x} + \frac{\partial \tau_{yz}}{\partial y} + Z_s = 0 \quad (18)$$

where σ and τ denote normal and shear stress components, respectively, and X_s , Y_s and Z_s are the body force components per unit volume.

The strain-displacement relations are given by:

$$\epsilon_x = \frac{\partial u}{\partial x}, \gamma_{yz} = \frac{\partial w}{\partial y} + \frac{\partial v}{\partial z} \quad (19)$$

$$\epsilon_y = \frac{\partial v}{\partial y}, \gamma_{zx} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \quad (20)$$

$$\epsilon_z = \frac{\partial w}{\partial z}, \gamma_{xy} = \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \quad (21)$$

The constitutive relations for linear elastic behavior are expressed as:

$$\epsilon_x = \frac{1}{E} [\sigma_x - \nu(\sigma_y + \sigma_z)] \gamma_{yz} = \frac{\tau_{yz}}{\mu_e} \quad (22)$$

$$\epsilon_y = \frac{1}{E} [\sigma_y - \nu(\sigma_z + \sigma_x)] \gamma_{zx} = \frac{\tau_{zx}}{\mu_e} \quad (23)$$

$$\epsilon_z = \frac{1}{E} [\sigma_z - \nu(\sigma_x + \sigma_y)] \gamma_{xy} = \frac{\tau_{xy}}{\mu_e} \quad (24)$$

where E is the Young's modulus and μ_e the Poisson ratio.

To evaluate the influence of aerodynamic loading on the structural response of the plate, a fluid-structure interaction (FSI) framework was employed. The fluid flow field and the solid deformation were coupled through the exchange of pressure and displacement at the fluid-solid interface, ensuring consistency of stress and kinematic conditions.

In the present study, the FSI analysis is primarily used to assess the overall deformation tendency of the plate under jet impingement, rather than to resolve detailed structural dynamics. Therefore, linear elastic material behavior is assumed for the plate, and quasi-steady coupling is adopted. This treatment allows the effect of flow-induced loading to be captured without introducing excessive computational complexity, while maintaining sufficient accuracy for evaluating flow stability and plate deformation characteristics.

3.4. Geometric model

The geometric configuration of the dual-slot nozzle system and the translating aluminum plate is illustrated in Fig. 2. The aluminum plate moves continuously along the streamwise (positive x) direction, while two symmetrically arranged nozzle units provide aerodynamic support and thermal interaction. Heated air is supplied to the nozzle modules through the inlet ducts and is discharged through inclined slot exits, forming oblique impinging jets that interact with the plate surface [25]. After impingement, the air is guided toward the outlet of the nozzle module.

The aluminum plate has a length of 1600 mm, a width of 450 mm, and a thickness of 0.6 mm. Each nozzle unit consists of two parallel slots with a slot gap of 10.0 mm, while the streamwise spacing between

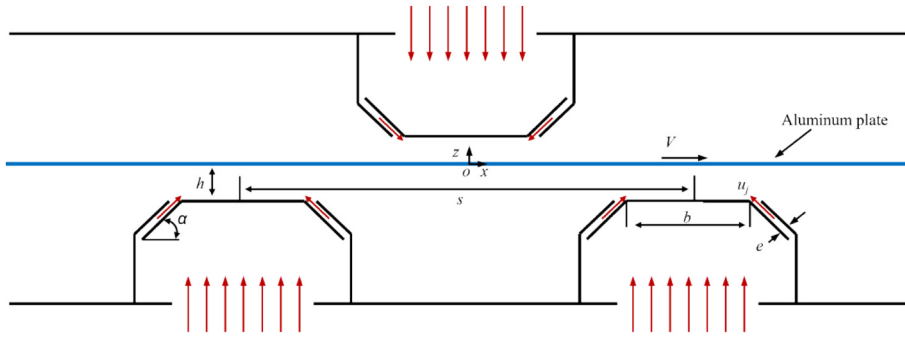


Fig. 2. Schematic view of arrangement of nozzle units and processed plate.

adjacent nozzle openings is 300 mm. The distance between neighboring nozzle units is 800 mm. The nozzle-to-plate spacing is denoted by h , u_j is the jet exit velocity, V is the plate translational speed, and r is the plate-to-jet velocity ratio.

To enhance flow redistribution between the dual-slot nozzles, additional rectangular jet orifices are incorporated into the nozzle module and arranged in either a single-row or a double-row configuration, as shown in Fig. 3. The single-row multiple-jet (SRMJ) configuration consists of a single row of rectangular jet orifices positioned along the spanwise centerline between the two slot nozzles. Each jet orifice has a rectangular cross-section with dimensions 10 mm $\times$ 60 mm, and the spacing between adjacent jet orifices is 31 mm. In contrast, the double-row multiple-jet (DRMJ) configuration features two parallel rows of rectangular jet orifices symmetrically arranged about the spanwise centerline. The streamwise spacing between the two jet rows is 134 mm, while the spanwise distribution and individual jet dimensions are identical to those of the single-row configuration. For both configurations, the geometric dimensions of individual rectangular jet orifices are kept identical, allowing the influence of the jet-row number on the flow and heat transfer characteristics to be examined independently.

Numerical simulations were performed using the commercial CFD (computational fluid dynamics) solver ANSYS Fluent 18.0. Based on typical operating conditions of air-cushion furnace systems, the initial temperature of the aluminum plate was set to 313 K, while the inlet air temperature was 923 K. Thermophysical properties of the air are listed in Table 2 [26]. The velocity inlet boundary condition was imposed at the air inlet, whereas the pressure outlet boundary condition was applied at the domain outlet, with the reference pressure set to atmospheric pressure.

A transient pressure-based segregated solver was employed to solve

Table 2

Thermophysical properties of air of 923 K.

Parameters	Values
Density (ρ), kg/m ³	0.3822
Specific heat capacity (c), J/kg·K	1126
Dynamic viscosity (μ), kg/m·s	4.119×10^{-5}
Thermal conductivity (k_p), W/m·K	0.0626

the incompressible governing equations. The pressure–velocity coupling was handled using the SIMPLE (Semi-implicit method for pressure linked equations) algorithm [27]. For the spatial discretization, a second-order upwind scheme was applied for the momentum, energy, and turbulence transport equations to ensure high numerical accuracy. The temporal discretization was implemented using a second-order implicit scheme. The convergence of the numerical solution was ensured by monitoring the residuals and key physical variables. The motion of the plate was modeled using a dynamic mesh approach, in which the plate was treated as a rigid body moving in the x -direction. The plate motion was prescribed through a user-defined function (UDF), and surrounding grids in the flow domain were updated accordingly with the translating plate.

At the beginning of each simulation, the leading edge of the plate was positioned at the inlet boundary of the computational domain. A uniform grid spacing of 0.6 mm was adopted in the wall-normal direction near the plate to ensure adequate resolution of near-wall gradients. The time step for the unsteady simulations was selected based on plate velocity and grid resolution. Convergence was achieved when the residuals of all monitored variables were lower than 10^{-5} .

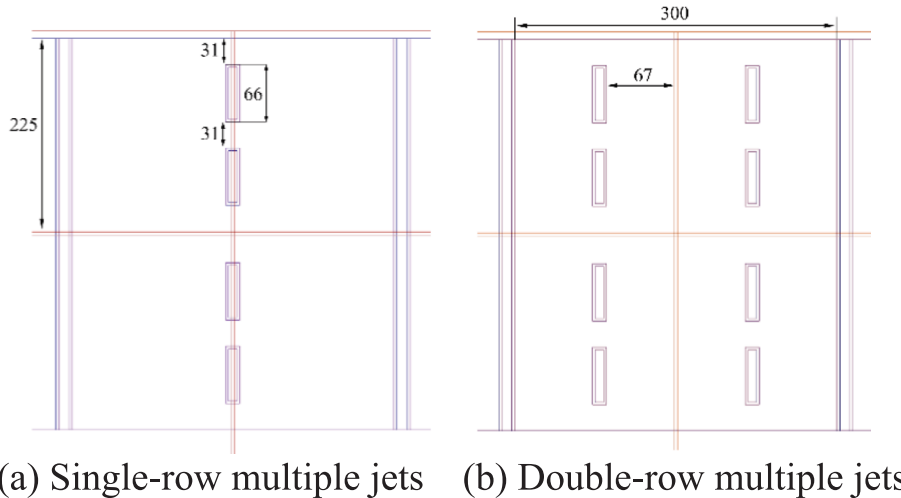


Fig. 3. Geometric arrangement of single-row and double-row multiple jets (Unit: mm).

3.5. Validation of numerical model

The computational domain was discretized using a hexahedral structured grid. Local grid refinement was applied in regions with high velocity and temperature gradients, particularly in the near-wall area and the free jet region, to ensure adequate resolution of the impingement flow and associated heat transfer phenomena. The overall grid topology is shown in Fig. 4.

To assess the sensitivity of the numerical results to grid resolution, a grid independence study was conducted using three grid densities, consisting of 1,762,477, 2,320,579, and 2,815,114 cells, respectively. All three grid schemes were generated for the same geometric configuration with $h/e = 4$, and were simulated under identical numerical settings at a jet Re of 10,000. Fig. 5 compares the streamwise distribution of the local Nu along the plate surface obtained with all three grid resolutions. The results indicate that the predicted Nu profiles converge as the grid is refined, with the relative deviation between Grid Schemes 2 and 3 remaining below 1% over the entire impingement region. In addition, the temperature at the first stagnation point ($x/e = -54$) was monitored for each grid scheme; the corresponding values are summarized in Table 3. The relative difference in stagnation-point temperature between Grid Schemes 2 and 3 was found to be less than 0.3%. Based on these comparisons, Grid Scheme 2 was deemed sufficient to achieve grid-independent results while maintaining reasonable computational cost, and was therefore adopted for all subsequent simulations.

To ensure the applicability of the SST $k-\omega$ turbulence model, the dimensionless wall distance (y^+) was carefully controlled. For accurate near-wall resolution, y^+ values are generally recommended to remain below 5 when using low-Reynolds-number turbulence models [28]. Fig. 6 presents the distribution of y^+ along the plate surface in the streamwise direction, demonstrating that the values consistently remain well below the recommended upper limit, thereby ensuring adequate near-wall mesh resolution.

3.6. Comparison of turbulence models

To further assess the suitability of the turbulence model, numerical predictions obtained using three commonly adopted models (SST $k-\omega$, v^2-f , and realizable $k-\epsilon$ with enhanced wall treatment) were compared against experimental measurements. The comparison was conducted at $Re = 20,000$, $h/e = 8$, and $R_{pj} = 0.1$, and the corresponding streamwise distributions of the local Nu are presented in Fig. 7. Overall, all three turbulence models capture the main heat transfer features, particularly in the stagnation and intermediate regions, which dominate the convective heat transfer process between the impinging jets and the plate.

Among the models examined, the SST $k-\omega$ model exhibits the closest agreement with the experimental data, with a maximum relative deviation in the predicted Nu of less than 2%. In contrast, the peak deviations

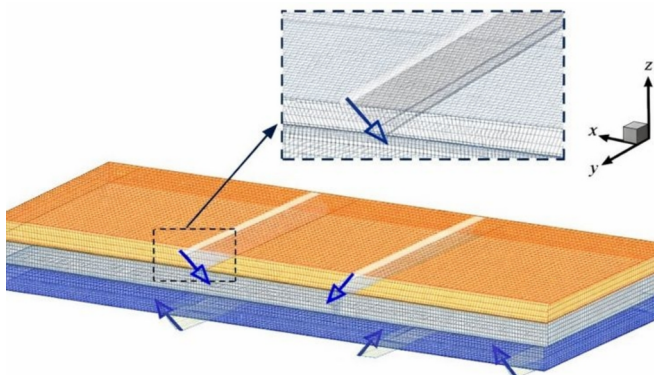


Fig. 4. Computational domain discretized with structured grids.

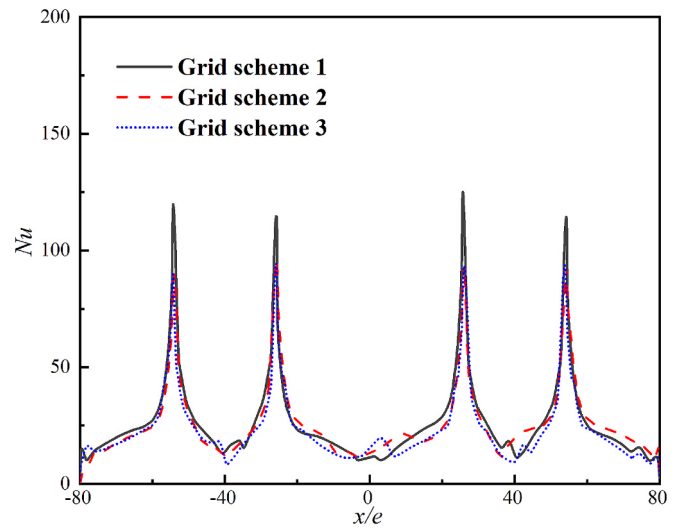


Fig. 5. Nu distribution in x -direction across different grid schemes.

Table 3

Temperature at the first stagnation point obtained with different grid schemes.

Grid scheme	Total grid number	Temperature at the first stagnation point (K)	Relative difference
1	1,762,477	745.3	+4.97%
2	2,320,579	710.2	—
3	2,815,114	708.1	−0.29%

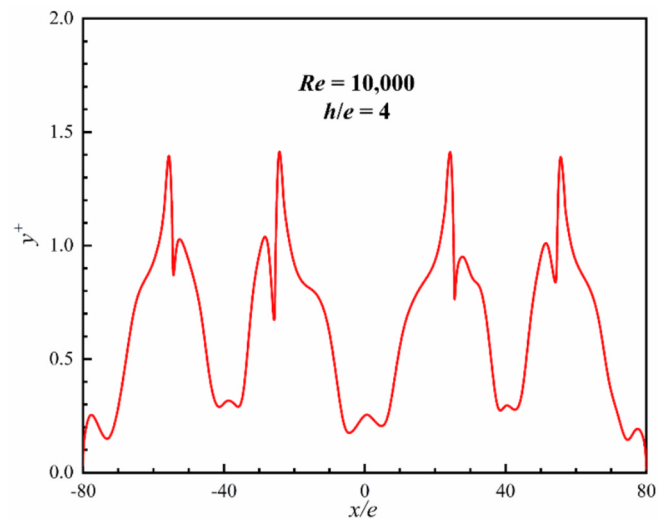


Fig. 6. Variation of non-dimensional distance at the wall (y^+) for Grid scheme 2.

obtained using the v^2-f and realizable $k-\epsilon$ models reach approximately 9.4% and 13.4%, respectively. Considering both predictive accuracy and robustness under the operating conditions relevant to air-cushion furnace systems, the SST $k-\omega$ turbulence model was therefore adopted for the present simulations.

It is noted that the experimentally measured Nu peaks are slightly lower than the corresponding numerical predictions. This discrepancy is primarily attributed to surface roughness effects of the aluminum plate, which cannot be precisely represented in the numerical model and are known to influence convective heat transfer characteristics [29]. Additional deviations observed near the lateral regions of the plate may arise from factors such as neglecting in-plane heat conduction within the

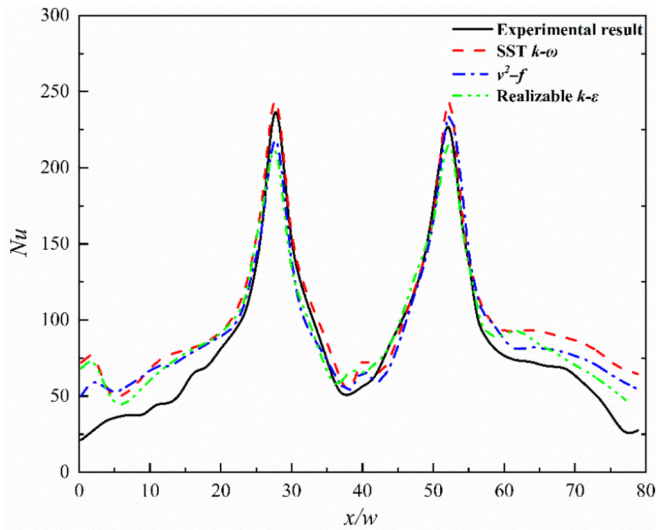


Fig. 7. Comparison of distribution of Nu between numerical results obtained using different turbulence models and experimental result.

plate, uncertainties in flow rate control, temperature-dependent material properties, and external disturbances during the experiments. Nevertheless, the overall agreement between numerical and experimental results is satisfactory, confirming the reliability of the CFD model for the present investigation.

4. Results and discussion

Based on the authors' previous investigations on dual-slot nozzle impingement systems for translating plates [8], the operating conditions in the present study are selected to ensure both representativeness and comparability. All experimental and numerical results reported in this work are obtained under a reference condition of $Re = 16,000$, upper/lower distance ratios $du/dl = 3:5$, plate-to-jet velocity ratio $R_{pj} = 0.01$, and nozzle inclination angle $\alpha = 60^\circ$.

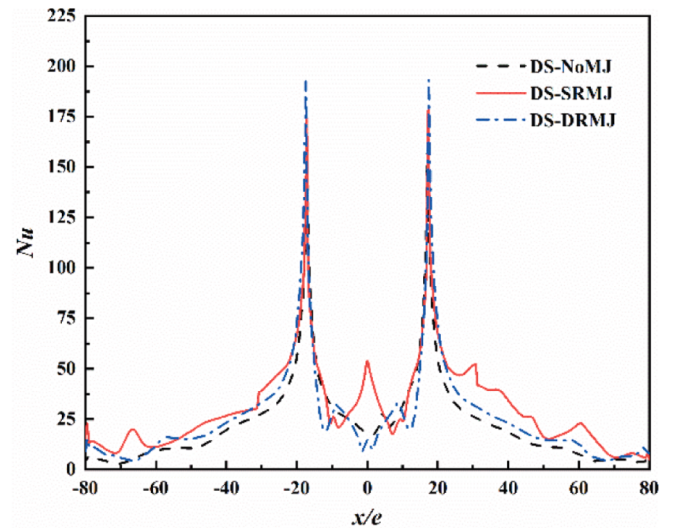
4.1. Heat transfer enhancement induced by multiple jets in dual-slot nozzles

As heat transfer enhancement constitutes the primary objective of the present study, the discussion begins with an analysis of Nu and surface temperature distributions.

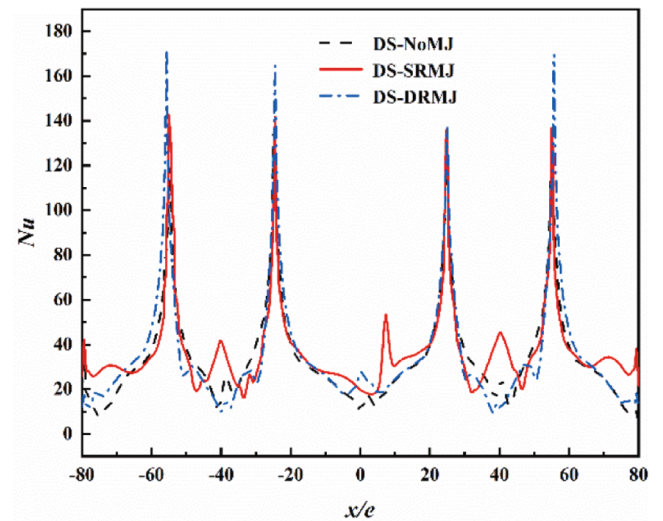
Fig. 8 presents the streamwise distributions of the local Nu on the upper and lower surfaces of the aluminum plate for the three nozzle configurations, namely the dual-slot nozzle without multiple jets (DS-NoMJ), with a single row of multiple jets (DS-SRMJ), and with double rows of multiple jets (DS-DRMJ). For all cases, two dominant Nu peaks are observed in the stagnation regions associated with the impingement of the inclined slot jets, indicating that the primary heat transfer mechanism is governed by strong jet impingement and near-wall acceleration.

With the introduction of multiple jets, a pronounced enhancement of heat transfer intensity is achieved. As the jet-row number increases from zero to one and further to two, the Nu peaks increase monotonically on both plate surfaces, demonstrating that the additional multiple jets effectively strengthen the impingement process and intensify local convective heat transfer. In particular, the DS-DRMJ configuration exhibits the highest stagnation Nu , reflecting the strongest jet-wall interaction and turbulence augmentation induced by the double-row jet arrangement.

However, distinct differences in the spatial distribution of heat transfer are observed between the single-row and double-row configurations. Although the DS-SRMJ case yields lower peak Nu than the DS-



(a) Upper surface of the plate



(b) Lower surface of the plate

Fig. 8. Nu distributions in the x -direction at $y/e = 6$ for various jet-row configurations.

DRMJ configuration, it generates noticeable secondary peaks in the inter-slot region, the central region between adjacent nozzle units, and the laterally confined zones. On the upper surface of the plate, the single-row configuration also leads to a broader Nu peak, indicating an expansion of the effective high heat transfer region. This redistribution of heat transfer intensity contributes to a more uniform surface heat transfer compared with the double-row configuration.

Increasing the number of jet rows enhances the overall heat transfer intensity, while the introduction of a single-row multiple-jet arrangement provides a favorable balance between heat transfer enhancement and spatial uniformity. The DS-SRMJ configuration therefore achieves a relatively high heat transfer level while simultaneously improving surface heat transfer uniformity, which is particularly advantageous for non-contact heat treatment applications involving continuous flat plates.

Fig. 9 presents the temperature contours on the aluminum plate for the three nozzle configurations. For all configurations, the surface temperature exhibits a clear increasing trend along the positive x -

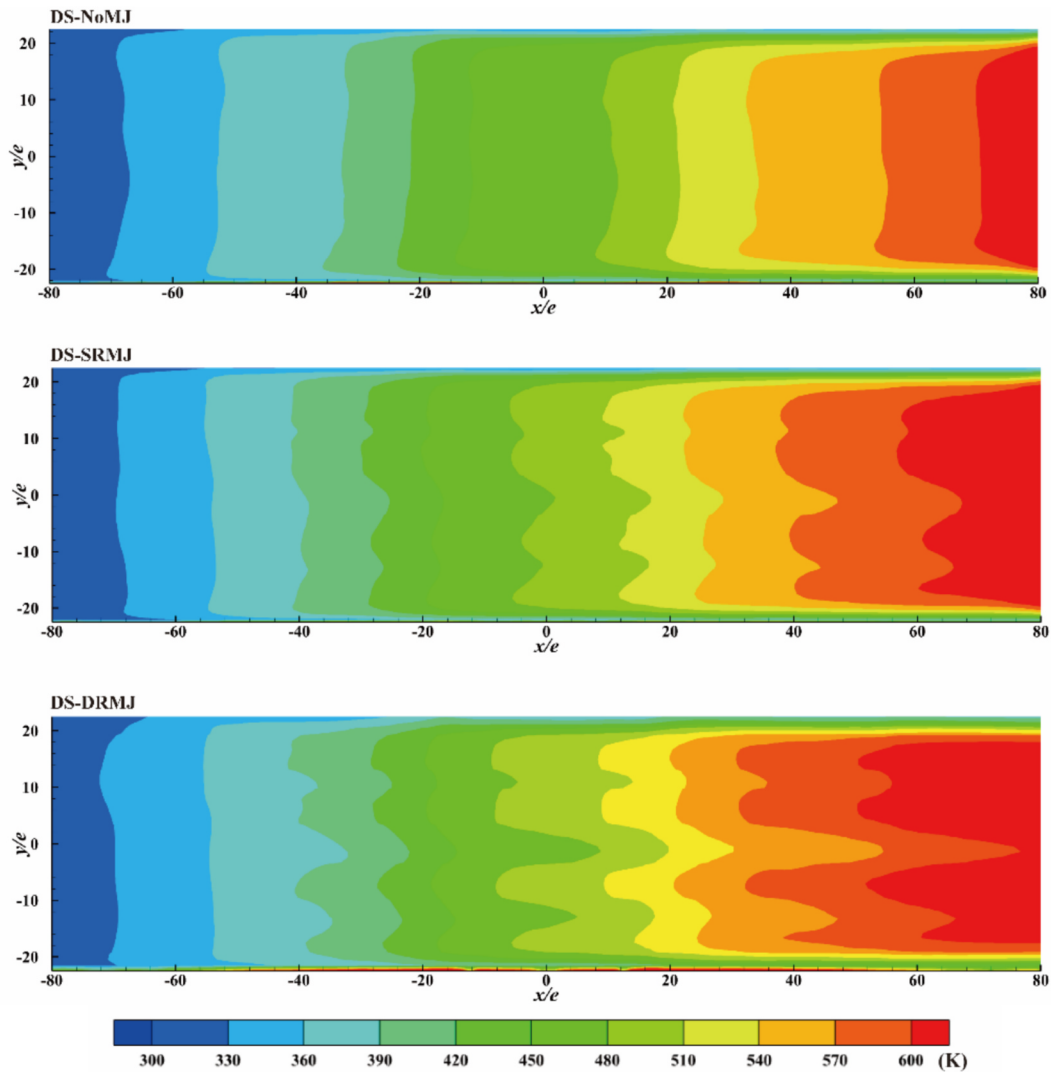


Fig. 9. Distribution of temperature over the plate for various jet-row configurations: dual-slot nozzle without multiple jets (DS-NoMJ), dual-slot nozzle with single-row multiple jets (DS-SRMJ), and dual-slot nozzle with double-row multiple jets (DS-DRMJ).

direction. This behavior is primarily attributed to the translational motion of the plate, which introduces a residence-time difference during the heat treatment process, leading to a longitudinal temperature gradient between the upstream and downstream regions of the plate. Similar temperature evolution characteristics have been reported in moving-plate impingement systems [30].

Among the three configurations, the DS-DRMJ case yields the highest average plate temperature, reaching approximately 495 K, indicating the strongest overall heat transfer intensity. This enhancement can be attributed to the increased number of jets, which intensifies jet-plate interaction and promotes convective heat transfer [31]. However, the temperature contours reveal pronounced spatial non-uniformity in this configuration, characterized by large local temperature gradients in both the streamwise and spanwise directions. As a result, the corresponding temperature uniformity index is relatively low ($TUI \approx 0.703$), suggesting that excessive jet interaction may deteriorate thermal uniformity. Such non-uniform heating is generally unfavorable for industrial heat treatment, as it may induce internal thermal stresses and compromise product quality [32].

In contrast, the DS-SRMJ configuration demonstrates a more balanced thermal performance. Although its average temperature (≈ 476 K) is lower than that of the double-row configuration, it is still significantly higher than that obtained without multiple jets (≈ 463 K

for DS-NoMJ). More importantly, the single-row arrangement effectively smooths the temperature field, resulting in a noticeably reduced temperature gradient across the plate. The calculated TUI reaches approximately 0.742, which is markedly higher than that of the DS-DRMJ case and only slightly lower than that of the DS-NoMJ configuration ($TUI \approx 0.757$). From the engineering perspective, the observed trade-off between the Nusselt number and temperature uniformity implies that nozzle array optimization in the air-cushion furnace must balance thermal efficiency with processing quality. Although the DS-DRMJ configuration offers the highest heating capacity, the resulting decline in TUI is detrimental to the heat treatment of sensitive aluminum plates, where localized temperature deviations may cause non-uniform grain growth or unexpected plate warping. Consequently, the practical design of industrial nozzles should prioritize configurations like DS-SRMJ, which provides a balanced enhancement and intensified convective heat transfer while preserving a uniform thermal environment to ensure structural and metallurgical integrity.

These results show that introducing a single row of multiple jets can substantially enhance heat transfer while maintaining a high level of temperature uniformity. Compared with the double-row configuration, the single-row design avoids excessive jet interference and provides a more uniform thermal environment, which is particularly beneficial for non-contact heat treatment processes requiring both efficient heating

and minimal thermal stress. Similar trade-offs between heat transfer enhancement and thermal uniformity have also been observed in multi-jet impingement systems reported in the literature [33].

4.2. Flow characteristics under different jet-row configurations

Fig. 10 illustrates the instantaneous velocity fields and vortex structures inside the air-cushion furnace for all three nozzle configurations: DS-NoMJ, DS-SRMJ, and DS-DRMJ. Despite the differences in jet arrangement, several common flow features can be identified. For all configurations, after impinging on the moving plate, the inclined slot jets attach to the plate surface and form high-speed wall jets due to the Coandă effect [34]. These wall jets subsequently interact with adjacent jets and are constrained by the confined geometry, giving rise to multiple recirculation zones. Typical recirculation structures are observed in the secondary stagnation region between the dual-slot nozzles, in the inter-nozzle regions beneath the plate, and in the laterally confined zones near the side boundaries. Similar flow features have been reported in confined impinging jet systems and air-cushion furnace configurations [20].

Because the plate translates in the positive x -direction, a pronounced shear effect is imposed on the near-wall flow. As a result, the recirculation zones exhibit an asymmetric deformation: vortices on the upstream side are stretched, whereas those on the downstream side are compressed. This asymmetry is particularly evident in the secondary stagnation region and between adjacent nozzle units, indicating strong coupling between the plate motion and the jet-induced flow structures [35].

The secondary vortices between the inclined slot jet and discrete jet rows play dual roles. While these vortices increase the turbulence kinetic energy and thus enhance local heat transfer, they also induce localized flow separation. As the plate moves, the interaction between these

vortices and the developing boundary layer becomes non-symmetric, shifting the peak Nusselt number slightly downstream. In the DS-NoMJ configuration, the velocity field remains relatively symmetric and orderly. The recirculation zones are well defined, with clear vortex cores and limited interaction between neighboring structures. However, the overall velocity magnitude is relatively low, which restricts turbulence intensity and convective heat transfer enhancement, consistent with the lower Nu observed and discussed in Section 4.1.

When a single row of multiple jets (DS-SRMJ) is introduced, the flow field undergoes a noticeable reorganization. The additional jets significantly increase the local velocity magnitude and promote stronger jet-jet and jet-wall interactions. As a result, the number of coherent recirculation structures increases markedly, from about 12 vortices in the DS-NoMJ case to about 20 vortices in the DS-SRMJ configuration. These vortices are more uniformly distributed along both the streamwise and spanwise directions, leading to enhanced turbulent mixing and more effective redistribution of momentum and thermal energy. Notably, in the DS-SRMJ case, pairs of counter-rotating vortices are clearly formed in the confined regions on both sides of the upper plate surface. Such symmetric vortex pairs are absent or poorly developed in the DS-NoMJ and DS-DRMJ configurations. The presence of these organized vortical structures contributes not only to heat transfer enhancement but also to improved spatial uniformity [36], which explains the favorable balance between high Nu and relatively high temperature uniformity observed for the single-row configuration.

In contrast, although the DS-DRMJ configuration produces the highest local velocities due to the increased number of jets, the flow organization deteriorates. The excessive jet interaction leads to vortex fragmentation in the secondary stagnation region, while the large recirculation zones beneath the plate tend to merge and lose coherence. The resulting flow field appears highly disordered, with irregular vortex shapes and unstable interactions. This phenomenon is consistent with

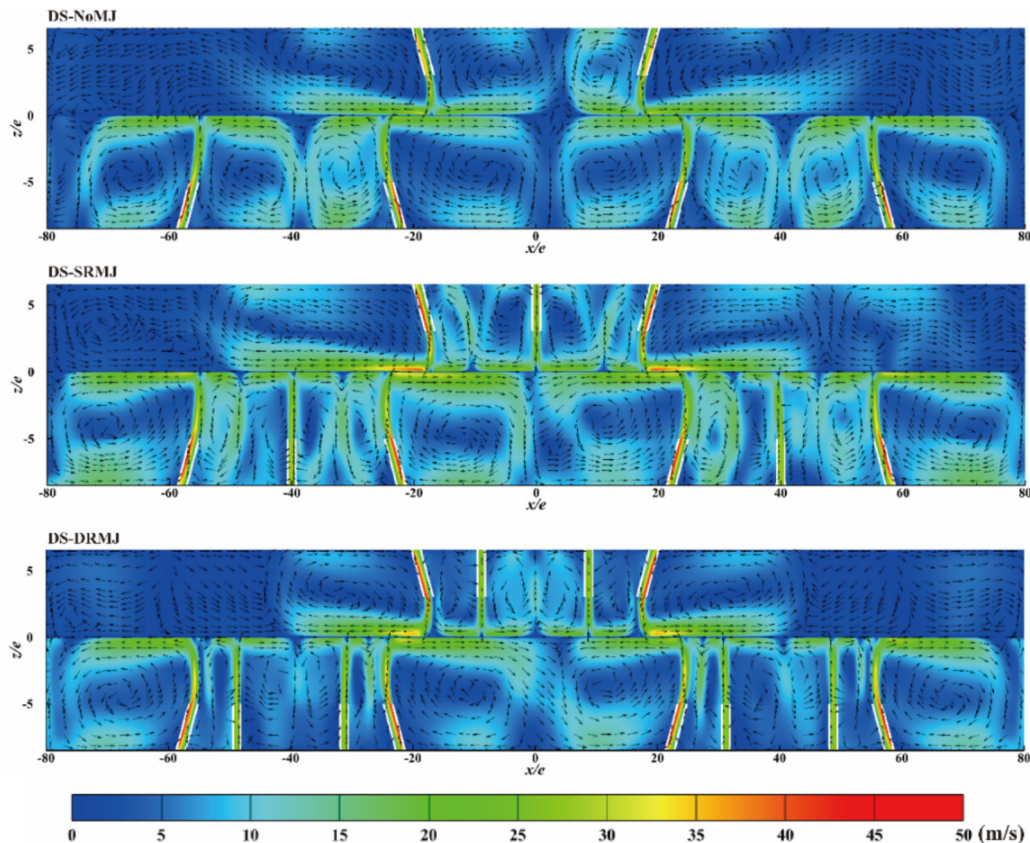


Fig. 10. Velocity vectors distribution at $y/e = 6$ at various jet-row configurations.

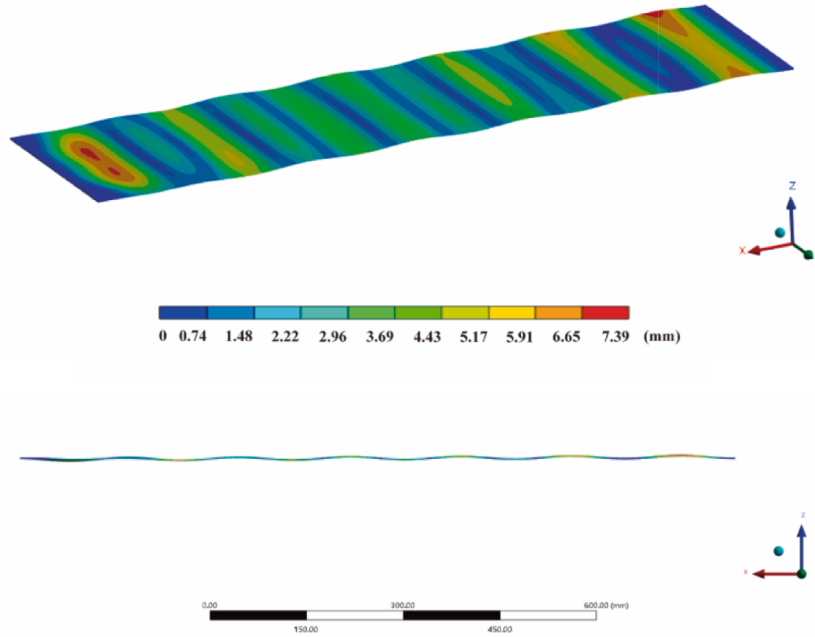
the findings in multi-jet impingement systems [37]. The resulting flow field appears highly disordered, with irregular vortex shapes and unstable interactions. This explains why the heat transfer distribution becomes more fragmented as the number of integrated jet rows increases. Such chaotic flow behavior weakens the ability of the system to uniformly redistribute heat, despite the locally intensified impingement,

thereby explaining the degraded temperature uniformity observed in Section 4.1 [38].

Overall, increasing the jet-row number does not necessarily improve flow organization or heat transfer performance. The single-row multiple-jet configuration provides an optimal compromise, in which enhanced jet-induced turbulence is achieved without triggering

DS-NoMJ

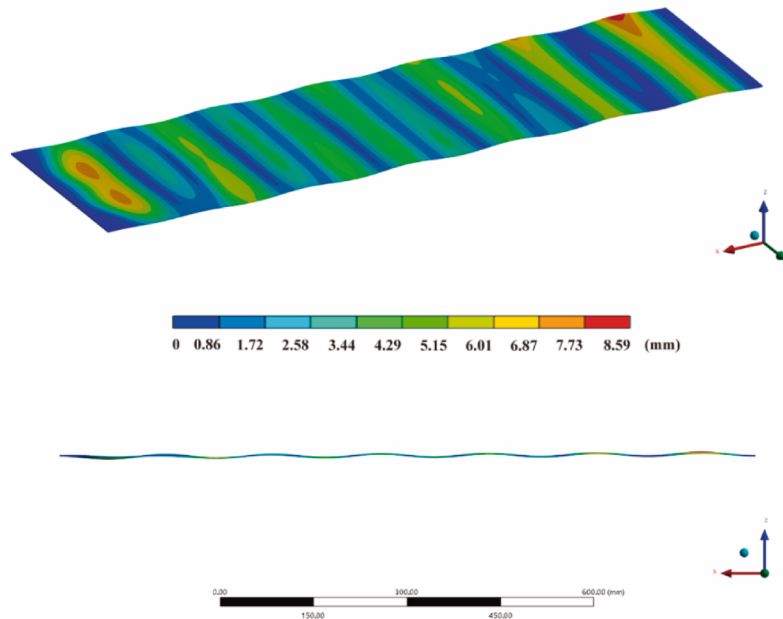
Max : 7.39 mm



(a) Dual-slot nozzle without multiple jets (DS-NoMJ)

DS-SRMJ

Max : 8.59 mm



(b) Dual-slot nozzle with single-row multiple jets (DS-SRMJ)

Fig. 11. Comparison of the deformation distributions over the aluminum plate for DS-NoMJ and DS-SRMJ.

excessive flow interference. The characteristics of a strengthened and well-organized flow field make the DS-SRMJ configuration attractive for heat treatment applications. Since the flow structures directly influence the pressure distribution and aerodynamic loading on the plate, their impact on plate deformation is further examined.

4.3. Effects of jet-row number on plate deformation and flow stability

Deformation of the aluminum plate is caused by the combined effect of aerodynamic loading, thermal gradients, and plate motion. Under three jet-row configurations (DS-NoMJ, DS-SRMJ, and DS-DRMJ), the deformation characteristics of the plate are comparatively investigated. For both the DS-NoMJ and DS-SRMJ configurations, the deformation profiles exhibit similar overall features. The plate surface shows a quasi-periodic wavy pattern along the streamwise direction, resembling a sinusoidal or cosine-type fluctuation. The maximum deformation remains relatively small, reaching 7.39 mm for DS-NoMJ and 8.59 mm for DS-SRMJ, as indicated in Fig. 11.

This deformation behavior is closely associated with the flow structures (see Fig. 10). After impingement, the slot jets attach to the plate surface due to the Coandă effect, forming high-momentum wall jets that distribute pressure along the surface [39]. The interaction among multiple slot jets produces cross-flow effects, which redistribute momentum laterally and weaken strong local pressure gradients [40].

In the wall-jet region, the interaction of spent flows from adjacent jets promotes relatively symmetric recirculation zones, helping balance the aerodynamic loading on both sides of the plate [41]. Meanwhile, because the plate moves in the x-direction, shear interaction between the moving wall and the gas shifts the vortical structures downstream, stretching vortices on the upstream side and compressing those downstream; however, this does not significantly break symmetry of the pressure field.

In the DS-SRMJ case, the introduction of a single row of multiple jets increases jet momentum and enhances mixing, leading to more vortical structures and higher velocity levels. Nevertheless, the additional jets also improve pressure redistribution and prevent strong localized pressure peaks. As a result, the plate deformation remains small and uniform, which is beneficial for stable plate transport.

A fundamentally different deformation mode appears under the DS-DRMJ configuration. The deformation becomes highly non-uniform and is concentrated in the central region of the plate, showing a pronounced upward bulging (in positive z-direction), as seen in Fig. 12. The maximum deformation sharply increases to 23.74 mm, which is nearly three times larger than in the other two cases. This behavior is closely linked to the altered flow topology in DS-DRMJ (see Fig. 10). Although the addition of a second jet row increases jet momentum, the vortical structures become disordered and unstable. Large-scale vortices beneath the plate begin to merge, while strong impingement from the upper jets creates localized high-pressure stagnation regions [42]. Consequently, the vertical pressure distribution becomes highly non-uniform. The resultant pressure difference between the upper and lower air cushions generates a net upward aerodynamic force that exceeds the plate weight and structural resistance, leading to a global bending-type deformation with the center lifted upward.

Plate deformation in an air-cushion furnace is governed by a strong coupling between flow-field organization and aerodynamic pressure distribution. Configurations that promote symmetric recirculation and gradual pressure redistribution maintain small, periodic deformation, whereas configurations that induce localized stagnation pressure concentration and breakdown of coherent vortex structures lead to large, centralized deformation. To further elucidate this mechanism, pressure distribution and pressure coefficient along the plate surface are analyzed in the following section.

4.4. Effects of jet-row number on pressure distribution and aerodynamic loading

To further elucidate the relationship between plate deformation and aerodynamic loading, the pressure fields on the upper and lower surfaces of the plate as well as the pressure coefficient c_p along the streamwise direction were examined under all three jet-row configurations (DS-NoMJ, DS-SRMJ, and DS-DRMJ). As shown in Fig. 13, the high-pressure regions are mainly located at the stagnation zones where the slot jets directly impinge on the plate, the secondary stagnation regions between adjacent slot jets, and the intermediate zones between nozzle modules. These locations correspond well with the jet

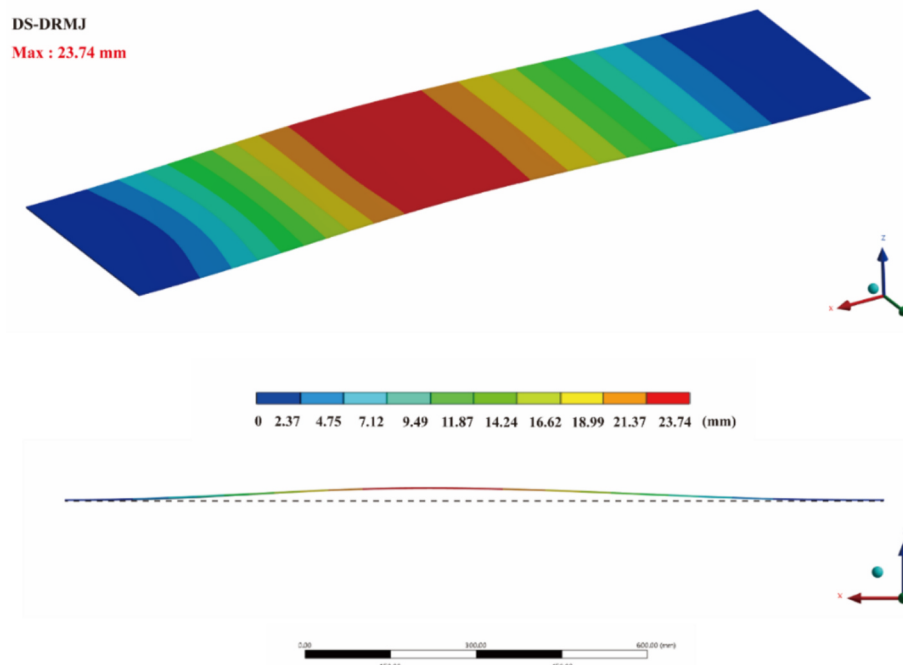
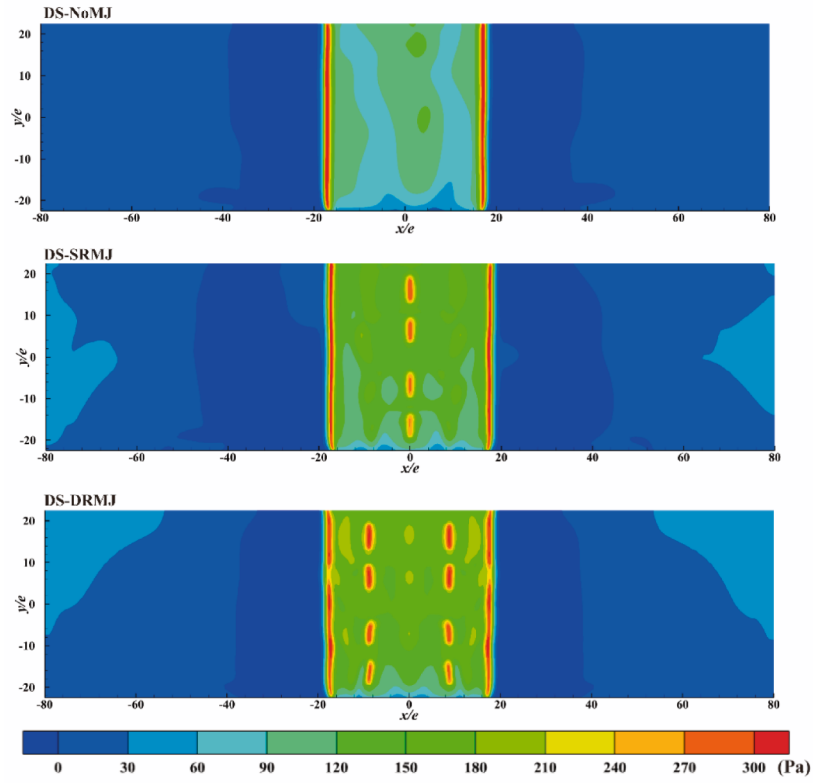
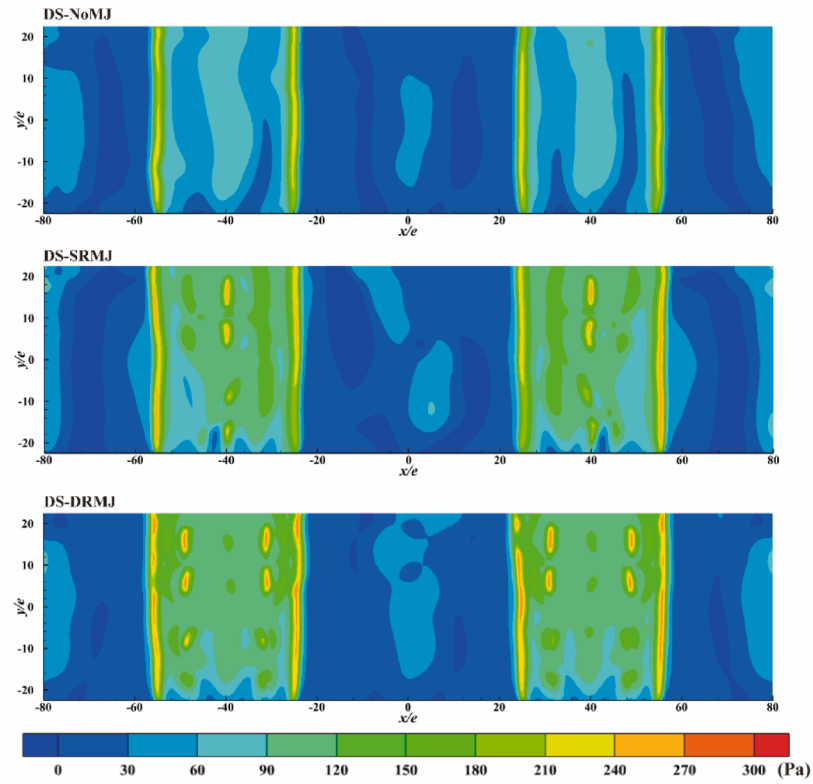


Fig. 12. Deformation distribution for dual-slot nozzle with double-row multiple jets (DS-DRMJ).



(a) Upper surface of the plate



(b) Lower surface of the plate

Fig. 13. Pressure distribution on the plate at various jet-row configurations.

impingement and wall-jet interaction regions identified in the flow-field analysis, confirming that the aerodynamic pressure pattern is governed by jet momentum, cross-flow interaction, and spent-flow recirculation [43].

Compared with DS-NoMJ, both DS-SRMJ and DS-DRMJ show intensified high-pressure spots in the central processing zone. However, the spatial distribution characteristics differ markedly. In DS-SRMJ, the additional single-row jets enhance jet momentum and mixing, but the pressure rise remains relatively continuous and spatially distributed. The high-pressure areas are elongated along the streamwise direction and connected through the wall-jet region, indicating effective pressure redistribution due to enhanced cross-flow interaction and symmetric recirculation structures [44]. In DS-DRMJ, the pressure field becomes highly localized. Discrete high-pressure cores appear beneath the multiple jet impingement points, especially near the plate center, while the surrounding regions remain at comparatively lower pressure. This indicates that the additional jet row strengthens local stagnation effects but weakens global pressure equalization; this is consistent with the breakdown and merging of vortical structures observed previously in the flow field.

These features are quantitatively reflected in the pressure coefficient distributions shown in Fig. 14. Sharp c_p peaks occur at the primary jet stagnation zones. After introducing multiple jets, the peak c_p values in both the primary and secondary stagnation regions increase markedly. The DS-DRMJ case exhibits the highest peaks, followed by DS-SRMJ, while DS-NoMJ shows the lowest values. Away from the stagnation zones, however, the differences among all three configurations are relatively small, indicating that the jet-row number mainly influences localized impingement loading rather than the background pressure level.

Combining these results with the deformation analysis reveals a clear coupling mechanism. For DS-NoMJ and DS-SRMJ cases, the pressure distribution is more laterally continuous and relatively symmetric between the upper and lower air cushions. The aerodynamic loads are therefore distributed over a wider area, producing small, quasi-periodic bending of the plate. In contrast, the highly concentrated high-pressure cores in the DS-DRMJ case generate strong local pressure gradients and a significant imbalance between the upper and lower surfaces. The resultant net upward force in the central region exceeds the plate's gravitational and structural resistance, leading to the pronounced central bulging observed in the deformation field.

Therefore, although DS-DRMJ provides the strongest local impingement and heat-transfer intensity, its highly non-uniform

pressure distribution deteriorates mechanical stability. DS-SRMJ achieves a more favorable balance: it enhances convective heat transfer compared with DS-NoMJ while maintaining relatively smooth pressure redistribution, resulting in moderate aerodynamic loading and improved plate stability.

To further clarify the mechanism linking aerodynamic loading to plate deformation, the pressure difference between the upper and lower plate surfaces (Δp) is introduced. The average Δp rises from 38 Pa for DS-NoMJ to 52 Pa for DS-SRMJ, and further to 131 Pa with DS-DRMJ. This trend closely follows the deformation magnitude, where the maximum plate deflection increases from 7.39 mm to 8.59 mm and further to 23.74 mm, respectively.

The nearly 2.5-fold increase in Δp from DS-SRMJ to DS-DRMJ leads to almost a threefold rise in deformation, indicating a strong aerodynamic-structural coupling effect. For DS-SRMJ, the moderate increase in Δp enhances heat transfer without inducing excessive bending, whereas DS-DRMJ generates an over-intensified vertical load that exceeds the structural tolerance of the thin plate [45].

As summarized in Table 4, DS-SRMJ provides the best compromise between heat transfer enhancement and mechanical stability. Although DS-DRMJ delivers the highest Nu , it causes excessive pressure concentration and plate deformation. DS-SRMJ, by contrast, achieves significant thermal improvement while maintaining acceptable pressure gradients and structural integrity, making it the most suitable configuration for practical air-cushion furnace applications.

5. Conclusions

This study numerically investigated the effects of the jet-row number in a dual-slot air-cushion furnace on the flow field, heat transfer performance, pressure distribution, and thermo-mechanical deformation of a moving aluminum plate. Three configurations were compared: DS-NoMJ, DS-SRMJ, and DS-DRMJ. The main conclusions drawn from the study are as follows:

- (1) The introduction of multiple jets significantly modifies the internal flow topology. DS-SRMJ increases jet momentum while maintaining relatively symmetric and coherent recirculation structures. The number of vortical structures increases and mixing between wall jets and spent flows is enhanced, promoting momentum redistribution. In contrast, DS-DRMJ produces disordered and partially broken vortical structures, especially in the secondary stagnation region, leading to strong localized

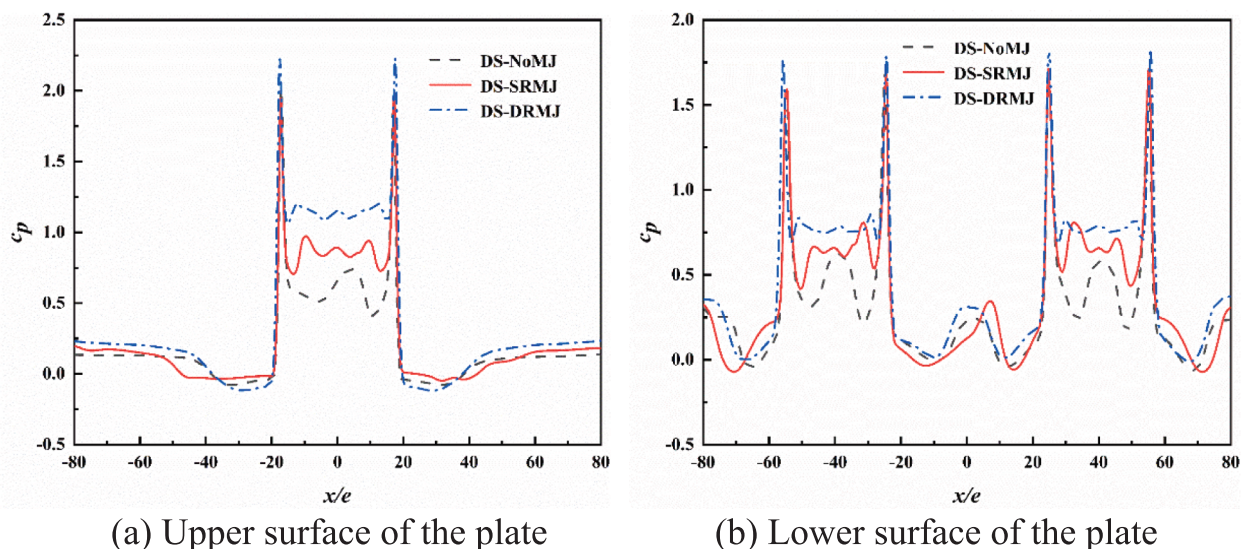


Fig. 14. Distribution of c_p in x -direction for various jet-row configurations.

Table 4

Comprehensive performance comparison of three nozzle configurations.

Metric	DS-NoMJ	DS-SRMJ	DS-DRMJ	Physical implication
Normalized Average Nu	1.00 (baseline)	1.18	1.26	Heat transfer intensity
TUI	0.757	0.742	0.703	Thermal uniformity
Δp (upper-lower)	38 Pa	52 Pa	131 Pa	Vertical aerodynamic load
Max deformation	7.39 mm	8.59 mm	23.74 mm	Structural stability
Overall assessment	Stable but weaker heat transfer	Best balance	Strong heat transfer but unstable	—

impingement and uneven pressure fields. DS-SRMJ maintains well-organized, counter-rotating vortex pairs. The DS-DRMJ configuration triggers excessive vortex fragmentation and unstable inter-jet interference, which impairs flow stability.

- (2) Both multi-jet configurations enhance heat transfer compared with DS-NoMJ. The average Nu increases by about 18% for DS-SRMJ and 26% for DS-DRMJ. However, thermal uniformity shows a different trend. Although DS-DRMJ yields the highest heat transfer intensity, it causes larger temperature gradients. DS-SRMJ maintains a high Temperature Uniformity Index ($TUI \approx 0.742$), only slightly lower than DS-NoMJ (0.757) and much better than DS-DRMJ (0.703), indicating that single-row jets achieve effective enhancement without severely weakening temperature uniformity. The superior performance of DS-SRMJ is attributed to its ability of maintaining stable thermal boundary layers during continuous plate translation, effectively balancing the heat transfer intensity and spatial uniformity.
- (3) The peak pressure coefficient and high-pressure area increase with the number of jet rows. The average pressure difference between the upper and lower air cushions increases from 38 Pa to 52 Pa and 131 Pa, indicating that DS-DRMJ generates excessive vertical aerodynamic loading.
- (4) Plate deformation is governed by the coupling between pressure distribution and structural response. DS-NoMJ and DS-SRMJ both exhibit quasi-periodic, distributed deformation patterns with small amplitudes (7.39 mm and 8.59 mm). For DS-DRMJ, deformation becomes highly centralized, with a maximum deflection of 23.74 mm, caused by concentrated stagnation pressure and large upper-lower pressure imbalance. Furthermore, the plate motion imposes a shear effect that induces non-symmetric pressure distributions, which is particularly distinct in the DS-DRMJ case, leading to aerodynamic instability. This configuration therefore compromises the dynamic stability of the moving plate.
- (5) Considering heat transfer, temperature uniformity, pressure distribution, and structural stability together, DS-SRMJ provides the best comprehensive performance. While DS-DRMJ offers the highest heat transfer rate, it induces excessive pressure concentration and deformation. DS-SRMJ achieves a favorable balance between thermal enhancement and mechanical stability, making it the most suitable jet arrangement for practical applications.

CRediT authorship contribution statement

Licheng Wu: Writing – original draft, Methodology, Investigation, Conceptualization. **Can Kang:** Writing – review & editing, Validation, Funding acquisition. **Patrick G. Verdin:** Writing – review & editing, Supervision, Methodology. **Jin Yin:** Validation, Software, Formal analysis.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

Data will be made available on request.

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