

Optimization of heat transfer and deformation control in aluminum plate heat treatment: Effects of nozzle inclination angle

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Abstract:

The deformation behavior of aluminum plates during heat treatment in an air-cushion furnace is numerically investigated, focusing on the effect of nozzle inclination angle. Validation of the numerical results was implemented with experimental data obtained from a specifically built experimental rig. The results reveal that at nozzle inclination angles of 30°, 45°, and 75°, the plates exhibit significant downward deflection in the central region, with maximum deformation (60.3 mm) occurring at 45°. In contrast, a distinct wave-like deformation pattern with minimal deformation (7.39 mm) arises at 60°, attributed to cross-flow interactions and shear effects induced by the plate motion. Consistency of the wave-like pattern is demonstrated at adjacent angles (57° and 62°). At an inclination angle of 60°, symmetric recirculating flows are reinforced, and the most uniform temperature distribution (average surface temperature of 463 K and temperature uniformity index of 0.757) is achieved. These findings highlight the critical role of nozzle inclination angle in reducing deformation and improving heat treatment quality, offering practical insights for industrial applications.

Keywords: air-cushion furnace; aluminum plate; deformation; heat transfer; Reynolds number; nozzle inclination angle

1. Introduction

Air-cushion furnace (ACF) serves as an advanced thermal processing equipment for the heat treatment of aluminum alloy plates [1]. The furnace utilizes arrays of floatation nozzles to eject high-velocity, high-temperature air streams, creating a stable air cushion that supports and heats the plate without physical contact [2]. This non-contact approach not only minimizes the risk of surface damage but also enables simultaneous double-sided heat treatment, significantly enhancing processing efficiency [3]. The floatation nozzle assumes dual functions, i.e., supporting the plate and facilitating efficient heat transfer between air and the plate. Its operation directly influence stability of the air cushion and uniformity of heat distribution over the processed plate [4]. The design of the floatation nozzle is based on the dynamics of impinging jets, which are widely used in industrial applications such as paper drying, turbine blade cooling, glass tempering, and metal annealing [5]. For ACF, the impinging jets issued from the nozzles disrupt the thermal boundary layer on the plate surface, thereby enhancing convective heat transfer and ensuring uniform temperature distribution [6].

Previous studies on ACF and floatation nozzles have primarily focused on structural aspects such as nozzle arrangement and nozzle shape optimization. Tepe et al. [7] investigated a 1×6 in-line array jet impingement configuration and found that the highest heat transfer performance was achieved at a dimensionless nozzle-to-target plate gap of 2.0. Badra et al. [8] simulated impinging jets using the SST $k-\omega$ turbulence model, and based on the stagnation Nusselt number (Nu), they concluded the optimum dimensionless jet-to-plate distance was in the range of 6-7. Afroz and Sharif [9] systematically analyzed effects of Reynolds number ($Re = 23,000$ and $50,000$), dimensionless jet-to-jet distance (0, 2, and 4), and jet-to-plate distance (2.6, 4, and 6) on jet impingement performance. Their results showed that the average Nu was strongly dependent on Re and impingement angle, but weakly and inversely related to the jet-to-plate distance. Baydar and Ozmen [10] conducted an experimental and numerical study to investigate the impingement of a confined jet on the upper surface. the dimensionless nozzle-to-plate spacing between 0.2 and 6. It was revealed that the heat transfer was

promoted with decreasing nozzle-to-plate spacing.

Flow and heat transfer are inter-related and of great significance for the application of jet impingement. Bentarzi et al. [11] analyzed the heat transfer between twin turbulent slot-jets and a heated wall. A comparison of heat transfer characteristics between perpendicular and oblique jets indicated that locations of stagnation points and the maximum Nu varied with jet direction. Caliskan et al. [12] examined flow and heat transfer characteristics of elliptic and rectangular impinging jet arrays. They proved that elliptic jets achieved relatively higher Nu in the impingement regions. Weng et al. [13] argued that doublet jets could enhance heat transfer by up to 233% compared to the single jet. This improvement is ascribed to turbulent behaviors such as enhanced entrainment and accelerated vortex pairing. Liu et al. [14] further demonstrated that jet array achieved 6.64% higher Nu than the single jet at equal flow rates, and they associated this enhancement to optimized turbulent mixing characteristics (notably the suppression of large-scale coherent structures). Heretofore, most of the studies have been dedicated to single-nozzle configurations and fixed nozzle inclination angles, and dynamic behaviors of the processed plate and its deformation under jet impingement and thermal stress have often been neglected.

Effects of main operation parameters such as Reynolds number of the jet, jet temperature, and the translational speed of the plate have been widely investigated. Senter and Sollicec [15] experimentally investigated the confined slot jet impinging on a moving flat surface at different Reynolds numbers. They showed that flow patterns were independent of Reynolds number within the tested range. In contrast, a slight variation of the surface-to-jet velocity ratio could significantly alter the flow field. Huang et al. [16] simulated a planar jet impinging a plate in motion and concluded that when the plate moved at a high speed, Nu decreased in the region opposite to the jet, while it increased in the region inclining towards the direction of plate motion. Shah [17] adopted different surface-to-jet velocity ratios (0.25, 0.50, 0.75, 1.0, and 2.0) in simulating jet impingement on a moving flat surface. They showed that local Nu varied with the velocity ratio, and in the stagnation zone, Nu decreases with increasing velocity

ratio. Chen et al. [18] investigated the effect of Re on the forced convection heat transfer coefficient. The results indicated that the optimum speed of the processed steel plate varied with its thickness when performance and safety requirements were fulfilled. Demir and Turgut [19] studied the combined circular-helical jets and concluded that heat transfer in the stagnation zone was enhanced because of the generation of strong secondary flows and intensified turbulent kinetic energy dissipation. Complementary to these findings, Li et al. [20] simulated the impact of rotational turbulence, revealing that variations in turbulent boundary layer structure, especially weakened streamwise vortices and near-wall turbulent fluctuations led to a decreased in Nu by 12–25%.

Although jet impingement and relevant heat transfer have been widely studied, critical knowledge gaps still persist in two main aspects. Firstly, structural behaviors such as deformation of the suspended plate have been neglected. With a thin plate, effects of dynamic behaviors on processing quality and operation stability are prominent. Secondly, the nozzle inclination angle has seldom been discussed for multi-nozzle configurations, and the effect of such a parameter is conceivably complex in the presence of jet stream interaction. Furthermore, when the plate is in motion, the relation between flow, heat transfer, and structural parameters has not been interpreted. This knowledge deficit is particularly pronounced when the design and orientation of nozzle arrays lack the guidance of generalizable conclusions.

To address these limitations, the present study was carried out based on a previous work [21]. The dimensions and operation parameters of an industrial-scale air-cushion furnace were adopted. The setup included two rows of nozzle units, with the plate positioned at a 3:5 ratio between the upper and lower nozzles. Using the computational fluid dynamics (CFD) technique, the study investigated effects of the nozzle inclination angle (α) on flow and heat transfer characteristics, and plate deformation was considered. An experimental rig was built to measure distributions of Nu and pressure on the impinged plate, thereby validating the numerical result. The relations between main flow parameters, Nu , pressure coefficient (cp), and plate deformation were comprehensively analyzed. The obtained conclusions are expected to provide valuable

insights for the optimization of nozzle unit of ACF and contribute to the understanding of the correlation between flow, heat transfer, and plate deformation.

2. Experimental methodology

2.1 Experimental setup

An experimental rig was designed and built to measure the distribution of Nu along the X -direction. A schematic view of this rig is shown in Fig. 1(a). It mainly consists of a centrifugal fan, a nozzle module, a positioning module, and a data acquisition module. Different operating conditions can be reached through regulating the operating parameters of each module. Different operating conditions can be obtained through regulating these modules. The rated volumetric flow rate and power of the centrifugal fan are $9.5 \text{ m}^3/\text{min}$ and 240 W , respectively. The rotational speed of the fan is adjusted through a variable frequency adjustable-speed equipment. Two heaters of 3 kW rated power are used to heat the flowing air. Driven by the centrifugal fan, air flows through a rectangular duct equipped with flow-stabilization screens.

The nozzle module comprises a $100 \times 100 \times 100 \text{ mm}$ cubic air duct and a dual-slot nozzle. A cross-section of the nozzle is shown in Fig. 1(b), where the blue arrows indicate the airflow direction, and the main dimensions of the nozzle are listed in Table 1. Two rows of air distribution holes are deployed inside the nozzle to ensure uniform velocity distribution at slot exit. The positioning module features a high-precision servo motor with $\pm 0.05 \text{ mm}$ positioning accuracy, ensuring accurate control of nozzle alignment. It includes a $200 \times 200 \times 20 \text{ mm}$ aluminum plate, a servo-driven lift platform, and a horizontal displacement module with a 400 mm displacement range. A multi-axis linear module leverages the servo motor's exceptional precision to dynamically adjust the nozzle's relative orientation to the plate.

A PT100 platinum resistance temperature detector (RTD) with an accuracy of $\pm 0.15 \text{ }^\circ\text{C}$ was deployed in the air duct to measure air temperature. Another four PT100 RTDs were symmetrically positioned on the aluminum plate to monitor plate temperature. A high-precision heat flux sensor (gSKIN XM269C, Switzerland) with dimensions of $4.4 \times 4.4 \times 0.5 \text{ mm}$ was mounted on the aluminum plate to record instantaneous heat flux

during air impingement. An image of the experimental rig is shown in Fig. 1(c). With a PID control system, air temperature can be precisely regulated. Cross-sectional air velocity in the air duct was measured using a hot-wire anemometer with an accuracy of $\pm 1.0\%$, and then the air flow rate was calculated.

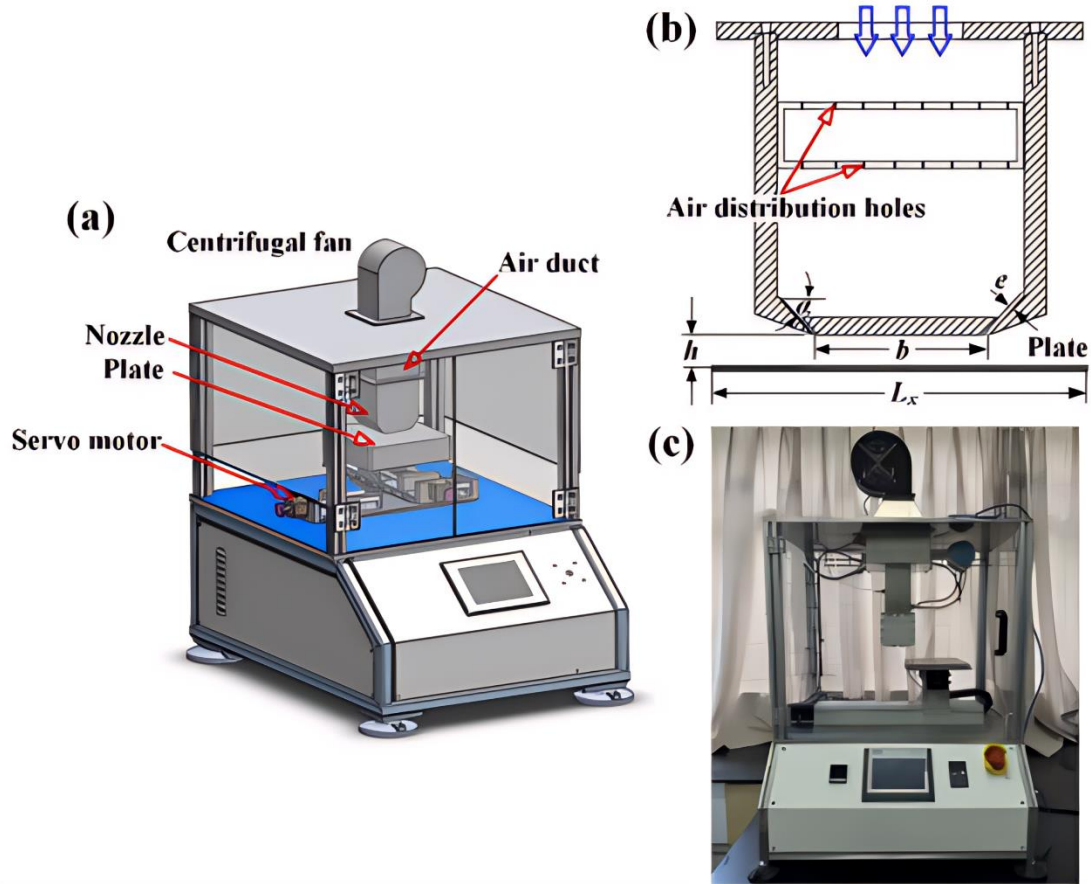


Fig. 1. Experimental rig: (a) schematic view, (b) nozzle cross-section, and (c) image.

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

2.2 Measurements of Nu and pressure

In the experiment, Nu was used to evaluate the convective heat transfer efficiency during the jet impingement process [22]. Since no phase change is involved, a simplified form of the thermodynamic boundary condition, specifically, a constant temperature boundary condition, is imposed on the plate. To describe heat transfer characteristics of the plate, the data acquired using the heat flux sensor were used to determine the distribution of Nu , which is calculated from:

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

where k represents the thermal conductivity, q the heat flux, and T_w and T_j the wall surface and reference temperatures, respectively. The air temperature at the nozzle outlet is selected as the reference temperature.

For the pressure measurement, a tempered glass measuring 200 mm × 200 mm × 6 mm was used as a rigid plate. A 10 mm diameter hole was drilled into the glass, where a pressure sensor was installed. The pressure sensor was connected to the high-pressure side of a differential pressure transmitter (HALO) via a flexible hose. The differential pressure transmitter has a full scale of ±5 kPa and an accuracy of 0.2%.

2.3 Uncertainty analysis

The experimental uncertainty was evaluated through the Kline and McClintock method[23]. For temperature measurement, the combination of the accuracy of PT100 RTDs (±0.15 °C), data acquisition error (±0.1 °C), and spatial variation (±0.2 °C) yielded a standard uncertainty of ±0.27 °C. At a coverage factor $k = 2$, the expanded uncertainty of ±0.54°C was obtained. For the calculation of Nu , the primary uncertainty originated from the temperature difference ΔT , resulting in a relative uncertainty of <5% for typical conditions ($\Delta T \geq 10$ °C). The heat loss accounted for <1.5% of the total heat input. Therefore, the overall relative uncertainty was ±3.27% for the calculation of Nu . The rig error depends on the accuracy of the HALO differential pressure transmitter (0.2%FS), installation error of pressure taps, and flow unsteadiness. The overall standard uncertainty was determined to be ±0.38%FS, resulting in an expanded

uncertainty of $\pm 0.76\%$ FS.

3. Numerical model

3.1 Governing equations

Regarding the cases characterized by air jet impingement on a plate, the fluid dynamics are described by the Reynolds-Averaged Navier-Stokes (RANS) equations, coupled with the energy conservation equation [24]. To achieve closure for these governing equations, the SST $k-\omega$ turbulence model was considered. This approach ensures accurate representation of turbulent flow behavior and energy transfer within the system. The governing equations describing three-dimensional, incompressible air flow are formulated through the conservation principles of mass, momentum, and thermal energy, expressed mathematically as:

Continuity:

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

Momentum:

$$\rho \frac{\partial U_i}{\partial t} + \rho U_j \frac{\partial U_i}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \rho \overline{u'_i u'_j} \right] \quad (3)$$

Energy:

$$\rho \frac{\partial T}{\partial t} + \rho U_j \frac{\partial T}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\frac{\mu}{Pr} \frac{\partial T}{\partial x_j} - \rho \overline{T' u'_j} \right] \quad (4)$$

where U_i , T and P represent the mean velocity, temperature, and pressure, respectively. T' and u'_i denote the fluctuating components of temperature and velocity. X_i indicates the coordinate directions. ρ is the air density, μ is the dynamic viscosity of air, and Pr is the Prandtl number, which can be obtained from:

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

Where c represents the specific heat capacity measured at constant pressure, k_f denotes the thermal conductivity of air.

The Reynolds stress and turbulent heat flux, which appear in Eqs. (1) and (3), are expressed as functions of variables resolvable on the computational grid.

$$-\rho \overline{u'_i u'_j} = \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \quad (6)$$

$$-\rho \overline{T' u'_j} = \alpha_t \frac{\partial T}{\partial x_j} \quad (7)$$

where μ_t and α_t correspond to the turbulent dynamic viscosity and turbulent thermal diffusivity, respectively.

The turbulent dynamic viscosity can be expressed as:

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

with C_μ representing a constant, while α_t is connected to the eddy viscosity through the turbulent Prandtl number Pr_t :

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

The intensity of convective heat transfer during air jet impingement on the plate is evaluated using Nu , determined by the thermal resistance characteristics of the aluminum plate, and expressed as:

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

The characteristic length of the nozzle is represented by D_h (m), while T_w and T_j correspond to the plate and air temperatures measured in K. The unit vector perpendicular to the plate's surface is indicated by n .

For the average Nusselt number calculation, the following expression is used:

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

where A represents the area of the aluminum plate (m²).

A dimensionless parameter, the pressure coefficient (c_p), is introduced to characterize the pressure distribution, calculated as the ratio of the static pressure on plate surface to nozzle outlet dynamic pressure:

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

where p_c denotes the static pressure on the plate surface, and u_j represents air velocity at nozzle outlet. The mean pressure coefficient ($c_{p,avg}$) is determined by:

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

Re of the air jet is defined by:

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

where the hydraulic diameter (D_h) is calculated based on the dimension of the nozzle gap.

The uniformity of temperature distribution was quantified using the temperature uniformity index (TUI), which is defined by:

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

Where σ is the standard deviation of the temperature of plate surface, and T_{avg} is the average temperature. When TUI approaches 1, high uniformity of temperature distribution is obtained.

3.2 Geometric model

The arrangement of the aluminum plate and nozzle arrays is shown in Fig. 2. The aluminum plate undergoes translational motion in the x -direction. The two nozzle arrays enable a stable suspension of the plate. Heated air is input into the nozzle modules through the designated air entry. This process results in the creation of obliquely oriented air streams via the specially designed slot nozzles, which then collide with the target plate [25]. Eventually, air is discharged from the module through the exhaust outlet. The aluminum plate has a length of 1,600 mm, a width of 450 mm, and a thickness of 0.6 mm. The configuration features a 10.0 mm gap for each slot, with successive nozzle openings separated by 300 mm. Each nozzle assembly is positioned 800 mm from its neighboring unit. The parameter h denotes vertical distance between nozzle outlet and plate surface, while u_j characterizes the air velocity at nozzle outlet. The parameter V denotes plate speed, and r denotes the plate-to-jet velocity ratio.

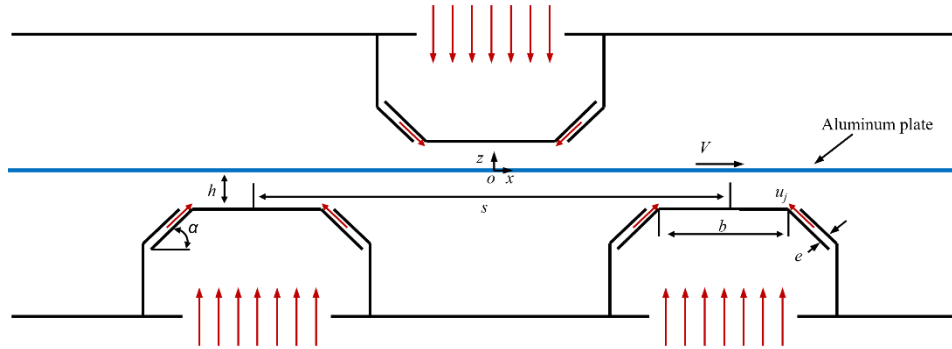


Fig. 2. Schematic view of arrangement of nozzle units and processed plate.

The numerical work was carried out using the commercial code ANSYS Fluent 18.0. Based on practical operating conditions of ACF, the initial temperature of the aluminum plate was set to 313 K, and air temperature 923 K. Physical properties of high temperature air are listed in Table 2 [26], The velocity inlet boundary condition was specified at the inlet of the computational domain, and the pressure outlet boundary condition was set at the outlet. The total pressure was set equal to the atmospheric pressure. A transient segregated solver was adopted. The continuity, momentum, energy, and turbulent transport equations were discretized using the second-order upwind scheme, and the pressure-velocity coupling was solved through the SIMPLE (Semi-Implicit method for pressure-linked equations) method [27]. A dynamic grid technique was used to handle the motion of the plate. In the computational model, the plate was approximated as a perfectly rigid structure, with its x -axis translation governed by a user-defined function (UDF). Grids in the fluid domain were modified in synchronization with plate motion. Computational initialization positioned the substrate's extremity at the domain boundary, with a fixed vertical cell dimension of 0.6 mm. The time step for the unsteady analysis was optimized based on grid translation velocity and spatial resolution. The convergence criterion of 10^{-5} was set for all monitored variables.

Table 2. Main thermophysical parameters of air at 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_f), W/m·K	0.0626

3.3 Validation of numerical model

Hexahedral structured grids were used to discretize the computational region. Grid refinement was implemented in wall-adjacent zones and free jet zones to capture those flow phenomena characterized by high spatial and temporal gradients. The computational domain discretized by grids is shown in Fig. 3.

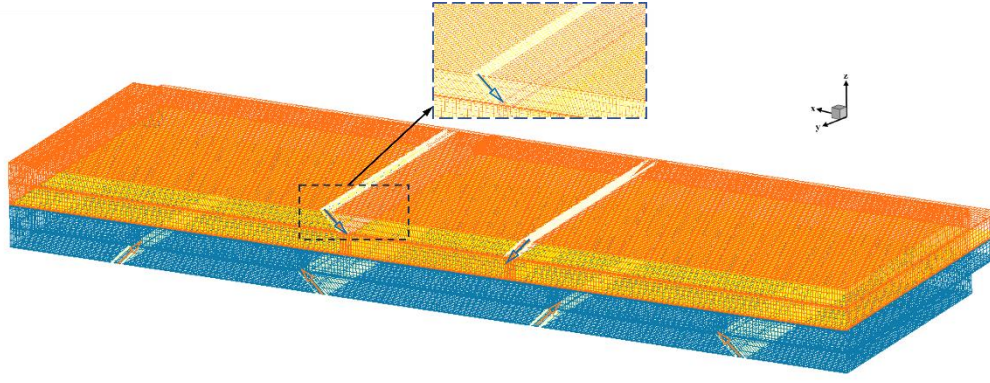


Fig. 3. Computational domain discretized with a structured grid.

To minimize the influence of total grid number on the numerical result, a grid independence examination was performed with three grid schemes, which involve 1,762,477, 2,320,579 and 2,815,114 grids, respectively. The grid schemes correspond to the configuration of $h/e = 4$. Numerical simulations were carried out with the same numerical settings at $Re = 10,000$. A comparison of the distribution of Nu along the x -direction on the plate obtained with the three grid schemes is shown in Fig. 4. It is seen that the relative discrepancy between Grid schemes 2 and 3 is lower than 1%. Furthermore, the temperatures at the first stagnation point ($x/e = -54$) obtained with the three grid schemes were also compared, and the result is listed in Table 3. Regarding the monitored temperatures, the relative discrepancy between Grid schemes 2 and 3 is lower than 0.3%. Therefore, Grid Scheme 2 was selected for subsequent simulations to optimize computational efficiency.

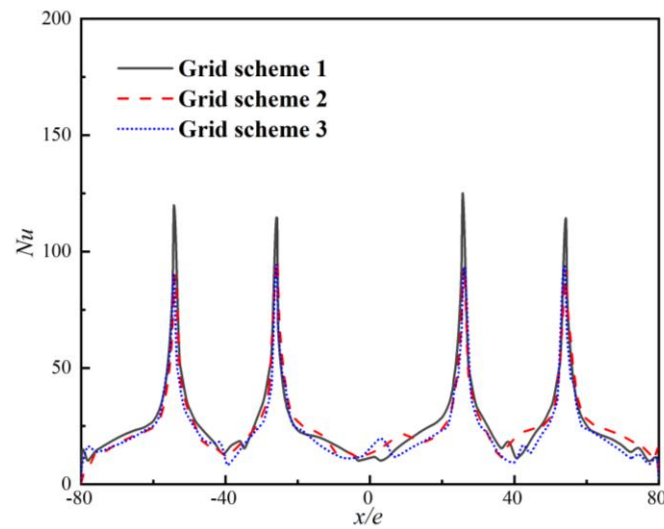


Fig. 4. Nu distribution in x -direction across different grid schemes.

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%

To ensure an appropriate use of the SST $k-\omega$ turbulence model, the y^+ value representing the normalized distance from the wall surface to the initial grid cell, should be less than 5 [28]. The corresponding y^+ on the surface of the plate in the X -direction is shown in Fig. 7, showing values well below the maximum recommended threshold.

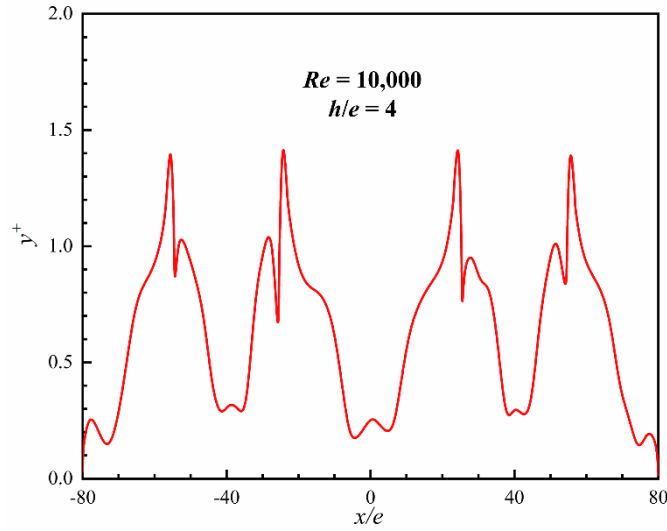


Fig. 5. Variation of non-dimensional distance at the wall (y^+) for Grid scheme 2.

To validate the numerical scheme, experimental data obtained from the experimental rig were used as a reference. A comparison of the distribution of Nu along the x -direction between numerical and experimental results is shown in Fig. 6 at $Re = 16,000$, $h/w = 8$, and $R_{pj} = 0.1$. It is clearly evident that the two distributions of Nu are similar. Overall, the relative deviation between the simulation data and the experimental results is quite low, particularly in the stagnation and intermediate regions of the two jets, with a maximum error of only 6.7%. The Nu peak values obtained through experiments are however slightly smaller than those obtained numerically. This discrepancy might be due to the surface roughness of the plate, which cannot be accurately predefined in the numerical setup. This will also affect the prediction of the convective heat transfer between air and the plate [29].

There is a significant discrepancy between the numerical data and experimental results on both sides of the plate. This discrepancy may be attributed to several factors within the numerical simulations, including surface roughness, the lack of accounting for conduction in the plate, variable air flow rates, material property variations, and external disturbances [30]. Altogether, the relative deviation of the numerical data from the experimental results is however reasonably low.

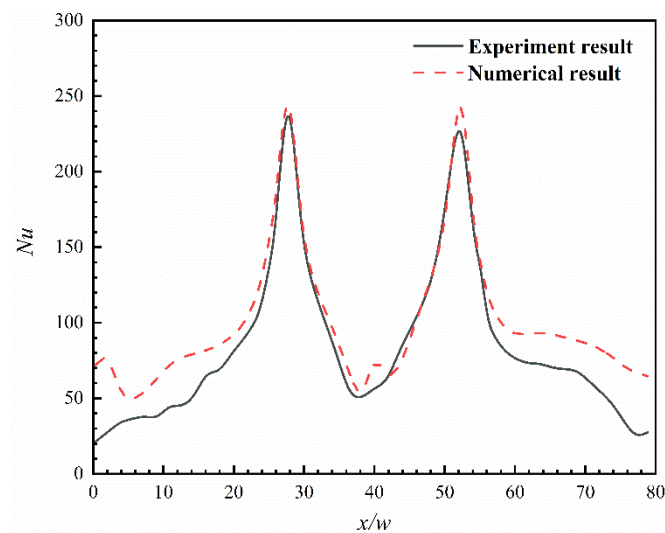


Fig. 6. Comparison of Nu distribution between numerical and experimental results.

Variation of pressure on the plate surface with Re in Fig. 7. It is seen that the experimental and numerical results exhibit consistent trends. As α decreases, the pressure on the plate increases. At $Re = 16,000$ or lower, the discrepancy between the two results is relatively small. However, as Re increases, the difference becomes more pronounced, and the numerical result increases faster than the experimental data. The primary reasons for the discrepancies are rig errors, such as air leakage and surface roughness, which affect the pressure drop [31]. Overall, the findings demonstrate a reasonable agreement between the experimental and numerical results within the specified Re range, providing a reliable basis for subsequent studies.

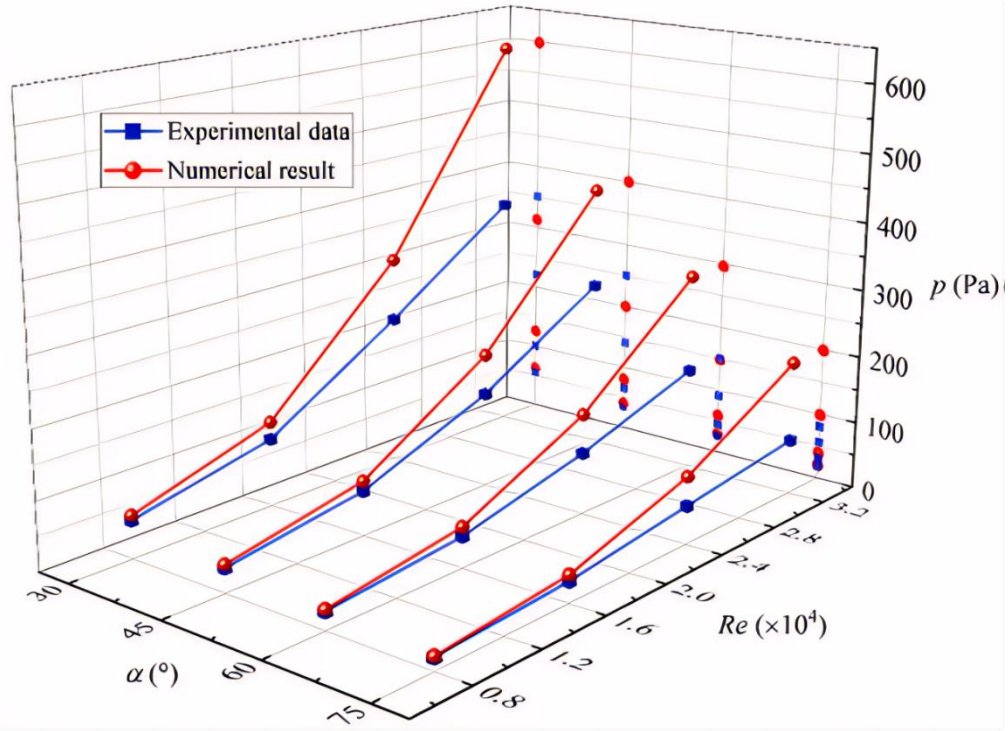


Fig. 7. Variation of pressure on plate surface with Re for various nozzle inclination angles.

4. Results and discussion

4.1 Effect of nozzle inclination angle on plate deformation

During the heat treatment of the aluminum plate in an air-cushion furnace, deformation is inevitable and significantly influences the final processing quality of the plate [32]. The nozzle inclination angle plays a critical role in determining the stability of plate suspension. To systematically investigate this relationship, the deformation behavior of the aluminum plate under various nozzle inclination angles is investigated. Four inclination angles ($\alpha = 30^{\circ}$, 45° , 60° , and 75°) were selected. When the inclination angle approaches 90° , i.e., vertical impingement, the formation of a uniform air cushion over the plate is difficult. When the inclination angle is smaller than 30° , the dual-slot jets could interact before impacting the plate. This will destabilize the air cushion and cause non-uniform heat transfer over the plate, and even plate vibration can be intensified. As shown in Fig. 8, the four deformation profiles are qualitatively similar, with the greatest deformation localized in the central region of the plate, manifesting as a downward deflection along the z -axis. This deformation pattern aligns with the stagnation region

of the upper jet, indicating that the pressure differential between the upper and lower flows is insufficient to fully counteract the plate weight [33].

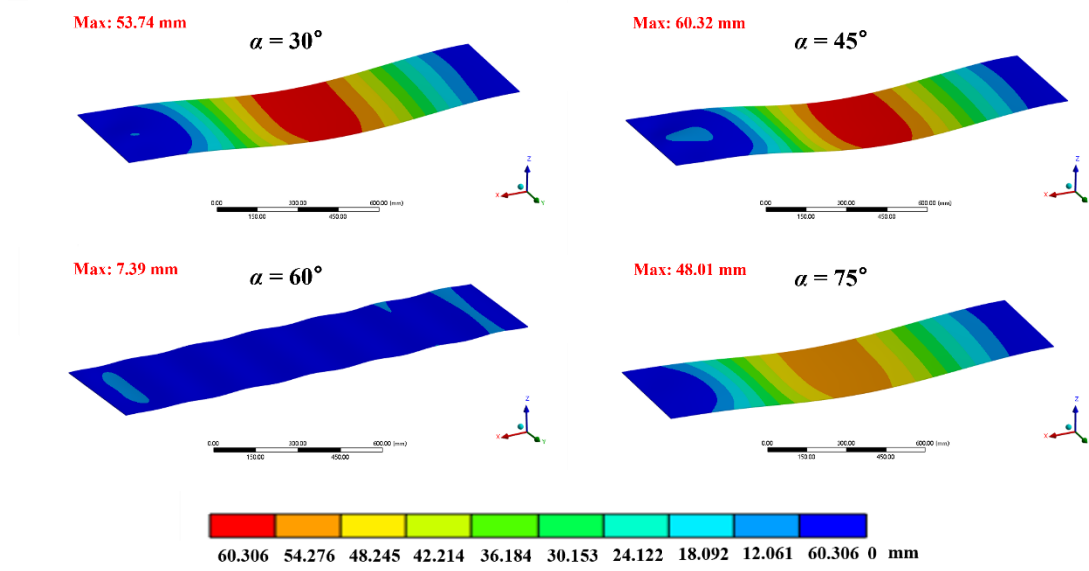


Fig. 8. Comparison of the deformation distributions over the aluminum plate at various inclination angle. ($Re=16,000$; $du/dl=3:5$; $R_{pj}=0.01$)

Quantitatively, the maximum deformation occurs at $\alpha = 45^\circ$, reaching 60.3 mm, while deformations at $\alpha = 30^\circ$ and 75° are 53.7 mm and 48.0 mm, respectively. All values are below the allowable deformation threshold of 62.5 mm to prevent mechanical interference. In contrast, at $\alpha = 60^\circ$, the deformation profile undergoes a marked transformation, adopting a wave-like morphology akin to a sine function, with a maximum deformation of 7.39 mm. This distinct behavior is likely attributable to the complex interaction of airflow dynamics within the furnace at this specific angle [34]. To rule out the possibility that the observed deformation at $\alpha = 60^\circ$ represents an anomalous result, additional simulations were conducted at adjacent inclination angles of 57° and 62° , and the result is shown in Fig. 9.

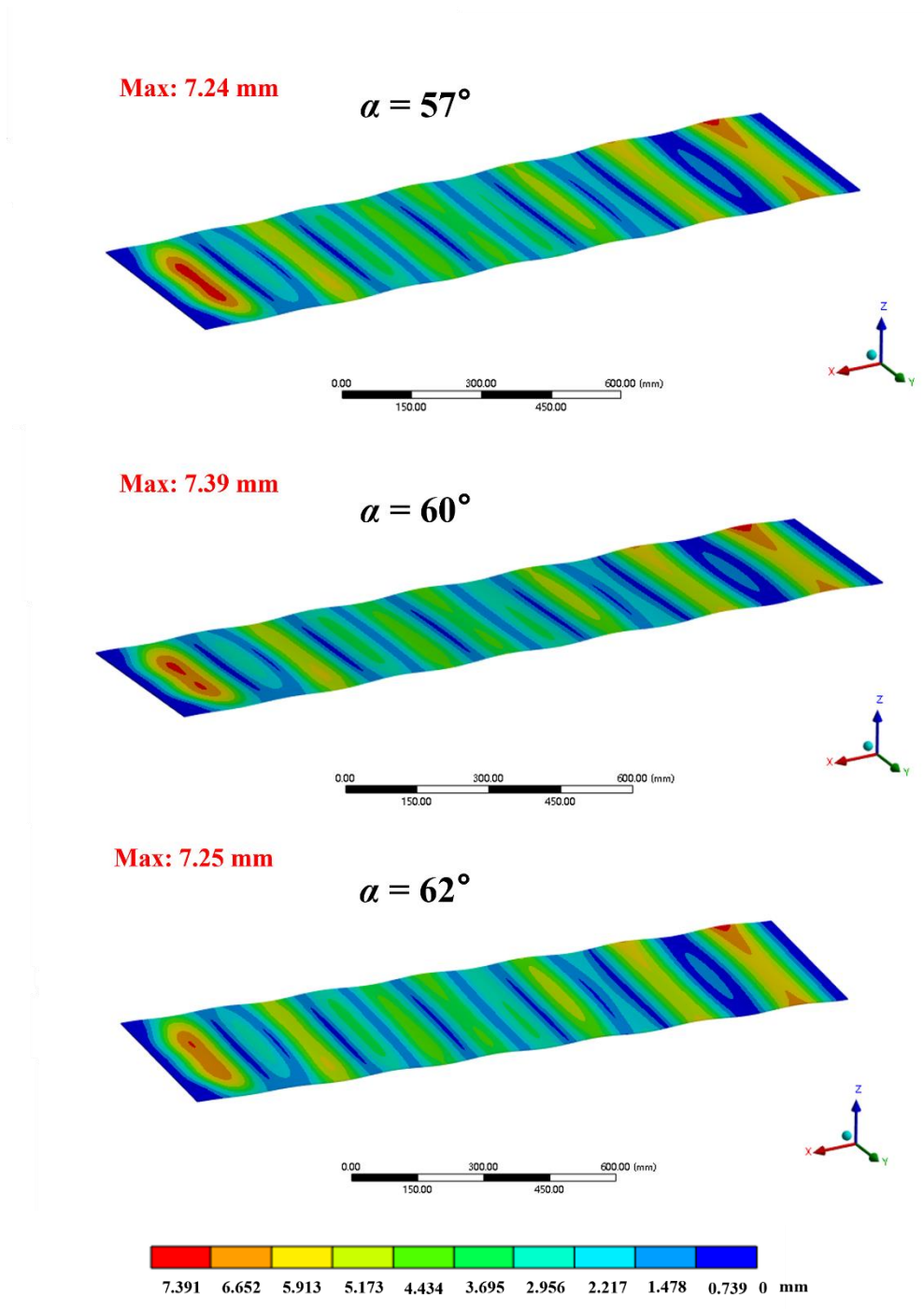


Fig. 9. The deformation distribution around the 60° inclination angle. ($Re=16,000$; $du/dl=3:5$; $R_{pj}=0.01$)

It is observed that the deformation trends at $\alpha = 57^\circ$, 60° , and 62° are remarkably consistent, with all plates displaying a wave-like pattern. The corresponding deformation magnitudes are 7.24, 7.39, and 7.25 mm, respectively. These findings confirm that an inclination angle around 60° induces a significant reduction in plate

deformation during heat treatment. This phenomenon can be explained by a combination of factors, including the interaction between the slot nozzle jet and the plate, the cross-flow effects arising from multiple slot nozzle jets, the interplay of spent flows within the wall jet region, and the shear effect generated by the plate motion [35]. Given these insights, further investigation into the associated flow and heat transfer characteristics is warranted to elucidate the underlying mechanisms driving this behavior.

4.2 Effects of nozzle inclination angle on flow and heat transfer

4.2.1 Flow structure

The flow patterns inside ACF at different nozzle inclination angles are shown in Fig. 10. Through a comparative analysis, it is observed that at $\alpha = 30^\circ$, 45° , 60° , and 75° , the nozzle jets, upon contacting the plate surface, form high-speed wall jets due to the Coanda effect [36]. These wall jets exhibit distinct turbulent characteristics, with evident flow disorder and directional change in the shear layers. Subsequently, due to the interaction between multiple air streams, a secondary stagnation zone between the dual-slot nozzles and the nozzle units forms symmetric recirculating flows in opposite directions. Additionally, a single recirculating flow is formed within the enclosed area bounded by the jet's confining walls, the plate, and the side walls. The turbulent mixing in this confined region is visibly stronger at $\alpha = 75^\circ$, where the flow displays distorted vortex cores and fragmented structures [14]. As the plate moves in the positive x -direction, the airflow undergoes shear effects, causing the recirculating flows to shift in the direction of plate movement at all nozzle inclination angles [37]. This shear-induced turbulence preferentially elongates the upstream recirculation zone while compressing the downstream zone.

Among these, at $\alpha = 30^\circ$, 45° , and 60° , the velocity distribution is relatively uniform, with clear and intact vortex structures, suggesting that these configurations offer more stable and predictable flow conditions. The coherence of these vortices implies suppressed turbulent energy dissipation compared to the case of $\alpha = 75^\circ$. This stability is crucial for maintaining the quality and consistency of the plate surface during

processing. In contrast, at $\alpha = 75^\circ$, the flow field appears disorganized, signifying fully developed turbulence. These potential instabilities could lead to uneven heating or cooling, thereby affecting the plate's mechanical properties [38]. At $\alpha = 30^\circ$ and 45° , the velocity difference in the flow around the plate is significant, with the maximum velocity differences reaching 90.2% and 46.4%, respectively. In comparison, at $\alpha = 60^\circ$, the recirculating flows are more symmetric, with the maximum velocity difference around the plate of 28.1%. Notably, a distinct fountain flow that forms within the secondary stagnation zone not only provides substantial support to the plate but also significantly enhances heat transfer [39]. This dual effect plays a critical role in precise thermal management and, more importantly, contributes to the effective reduction of plate deformation by maintaining uniform temperature distribution and minimizing thermal stresses [40].

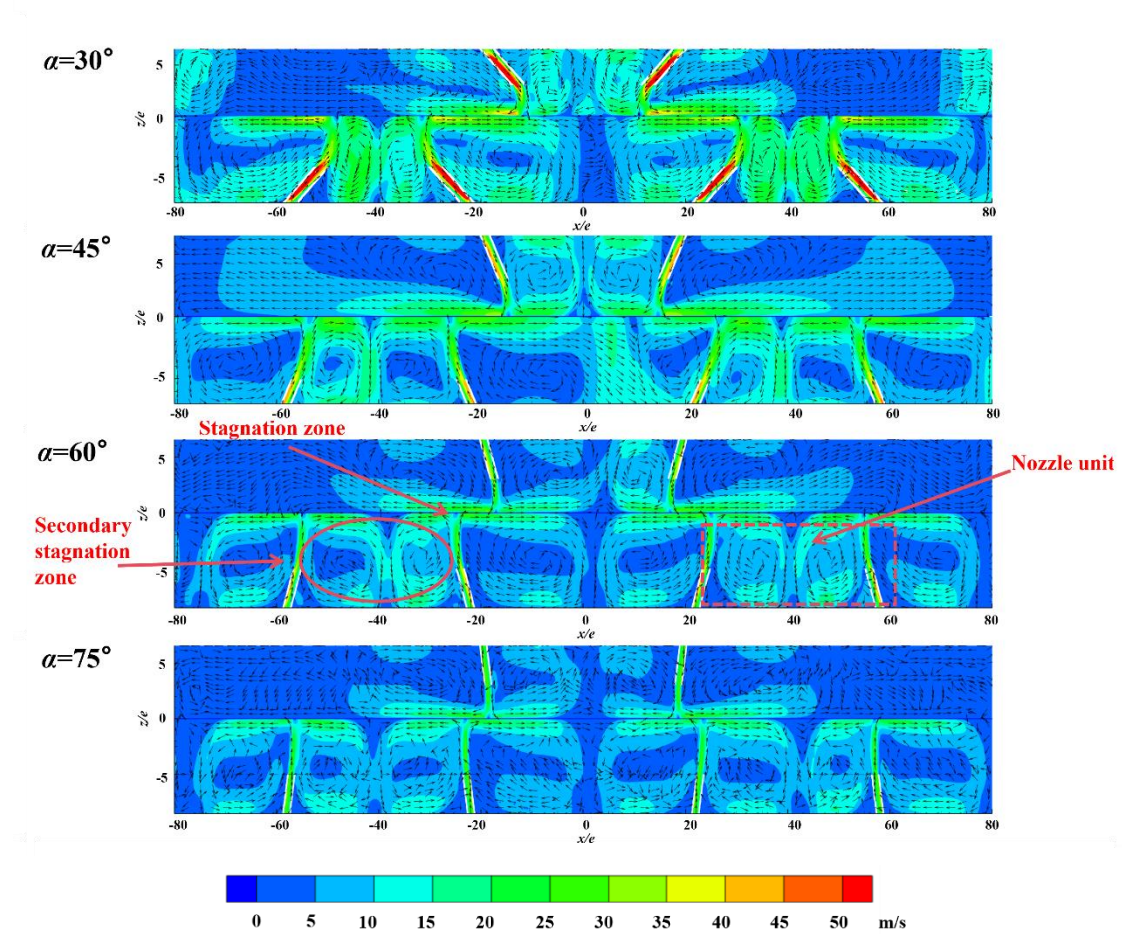


Fig. 10. Velocity vectors distribution at $y/e=0$ at various inclination angles.

4.2.2. Temperature and Nu distributions over the plate

The temperature distribution over the aluminum plate during the heat treatment process under various nozzle inclination angles are shown in Fig. 11. It is observed that the temperature distribution on the plate exhibits an increasing trend in the x -direction at any nozzle inclination angle. This phenomenon is attributed to the plate movement at a certain speed towards the positive direction of the x -axis, which introduces a time lag in the heat treatment process as the plate enters the furnace [41]. Consequently, a temperature gradient is established between the front and rear ends of the plate. At $\alpha = 30^\circ$, the average temperature of the plate reaches its maximum of 489 K with the lowest temperature uniformity index ($TUI = 0.734$). This is due to higher jet velocity at this nozzle inclination angle, which results in a stronger impact on the plate and the most intense convective heat transfer process [42]. However, the temperature distributions reveal that temperature gradients on the plate surface are significant at this nozzle inclination angle, leading to a highly non-uniform temperature distribution. This nonuniformity is detrimental to heat treatment effectiveness for the plate, as it fails to eliminate internal thermal stresses [43]. In contrast, the temperature distribution uniformity is considerably higher for the other three nozzle inclination angles. Specifically, at $\alpha = 60^\circ$, the average temperature on the plate surface reaches its maximum of 463 K and corresponding TUI is 0.757.

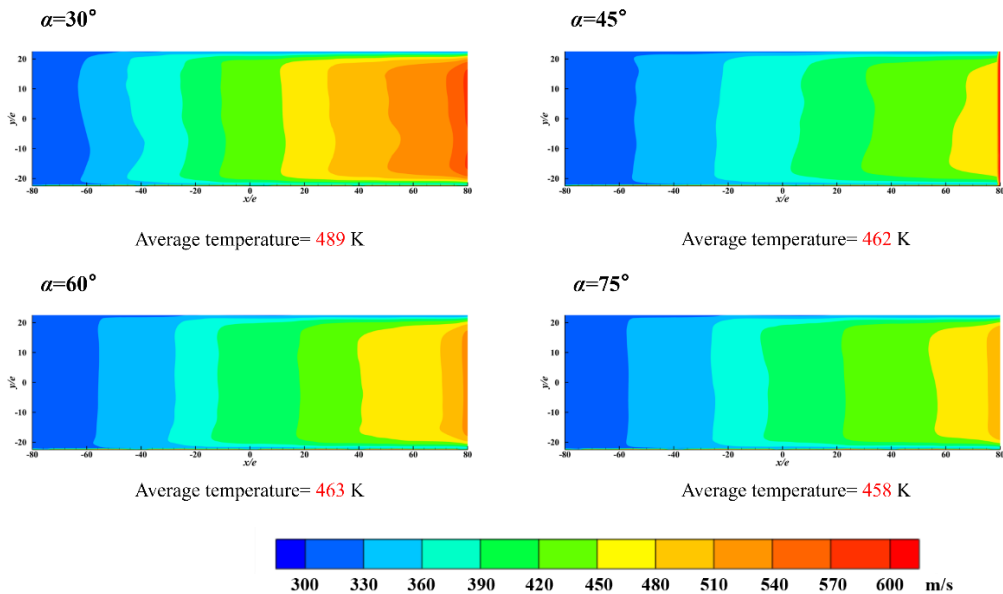


Fig. 11. Distribution of temperature over the plate at various nozzle inclination angles. Distributions of Nu on the upper and lower surfaces of the plate at various nozzle inclination angles are shown in Fig. 12. It is observed that dual primary peaks of Nu consistently emerge within the stagnation regions where the slot nozzle jets impinge on the plate. As α increases, the magnitudes of the Nu peaks gradually decrease. Meanwhile, the positions of these dual primary peaks shift along the positive and negative directions of the x -axis in response to the movement of the stagnation point, resulting in an increased separation between the dual peaks. This phenomenon significantly contributes to enhancing the uniformity of heat transfer across the plate surface. At $\alpha = 60^\circ$, distinct secondary Nu peaks arise due to intensified turbulence and cross-flow interactions in the secondary stagnation zone [44]. Inclined jets cause vortical structures downstream of the primary impingement regions, which reinforces the mixing between the spent flow and fresh jets. This turbulence-driven momentum transfer stimulates development of the thermal boundary layer, creating the secondary enhancement of heat transfer [45]. Furthermore, this configuration is responsible for relatively high primary Nu peaks. The combined effect of primary and secondary peaks enables a more continuous high- Nu region, and overall uniformity of heat transfer is improved. Therefore, the nozzle inclination angle of 60° is remarkably suitable for applications requiring balanced thermal distribution.

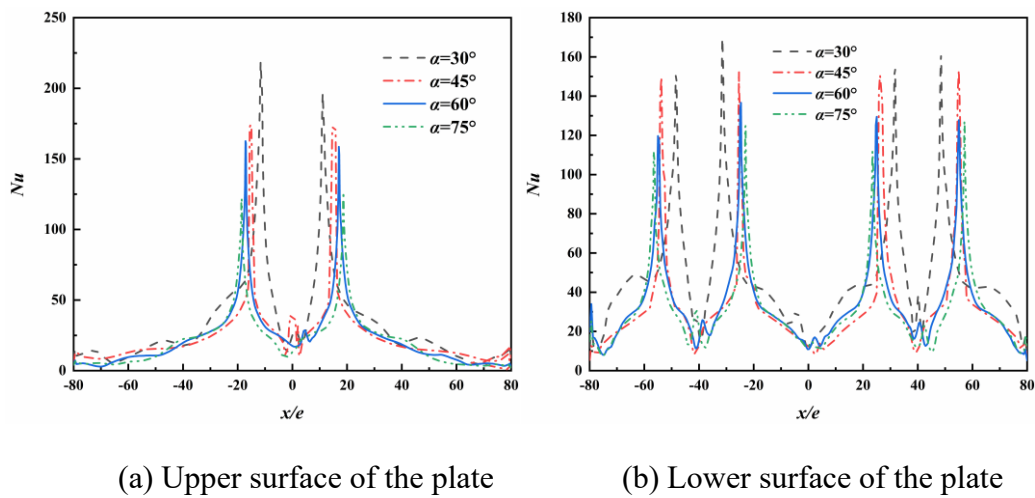
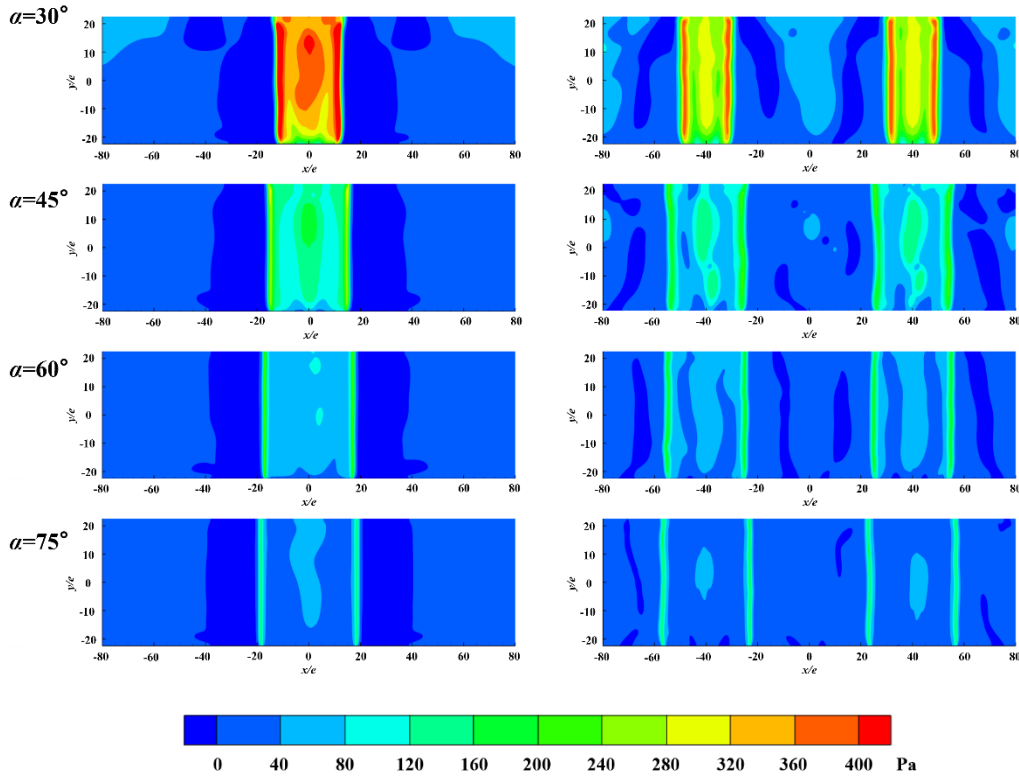


Fig. 12. Nu distribution along the x -direction at $y/e=0$ at various nozzle inclination angles.

4.3. Effect of nozzle inclination angle on pressure distribution

To further investigate the relationship between deformation and pressure during the heat treatment process, coefficient and distribution of pressure on the upper and lower surfaces of the plate are analyzed. As shown in Fig. 13, high-pressure regions are identified in the stagnation zones where the slot nozzle jets impinge on the plate, as well as in the secondary stagnation zones and the intermediate regions between the nozzle units. This is related to the interaction between the jets and recirculating flows formed due to the impact of the jet streams on the plate [46].



(a) Upper surface of the plate

(b) Lower surface of the plate

Fig. 15. Pressure distribution on the plate at various nozzle inclination angles.

Pressures on the plate at $\alpha = 30^\circ$ are significantly higher than those observed in the other three cases, leading to strengthened loading on the plate and concave deformation. This tendency aligns with the pressure coefficient distributions shown in Fig. 14. Consequently, a highly non-uniform pressure distribution across the plate surface is

formed. However, as α increases to 45° , both pressure and peak pressure coefficient decrease substantially, and the distance between the peaks within the high-pressure regions is enlarged, resulting in increased overall concave deformation. When α is further increased to 60° , relatively high pressure coefficients are maintained, which enhances convective heat transfer. Moreover, localized strain concentration is suppressed, leading to wave-like plate deformation with a minimal maximum, deflection of 7.49 mm. This indicates that the deformation mode depends on both pressure magnitude and the distribution of pressure peaks. The wave-like deformation suggests elastic buckling due to uniform loading. At $\alpha = 75^\circ$, concave deformation reappears, demonstrating the critical role of pressure distribution uniformity in controlling plastic strain distribution. The reduction in peak pressure coefficients becomes less pronounced, and the positional changes of the peaks are not as significant, as seen in Figs. 13 and 14. At $\alpha = 60^\circ$, the plate maintains relatively high peaks in the pressure coefficient, which enhances the convective heat transfer, and it also exhibits smoothed peaks, thereby ensuring a more uniform pressure distribution across the plate surface.

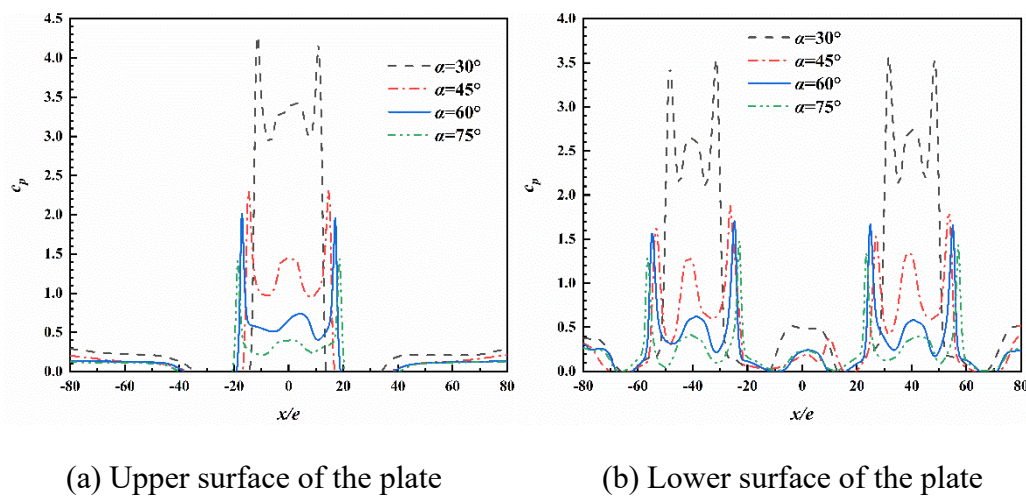


Fig. 14. Distribution of c_p in x -direction at various nozzle inclination angles.

5. Conclusions

(1) The nozzle inclination angle significantly influences the deformation behavior of the aluminum plate heat-treated in the air-cushion furnace. At nozzle inclination angles of 30° , 45° , and 75° , the plate exhibits a downward deflection along the z -axis, with the

maximum deformations of 53.7 mm, 60.3 mm, and 48.0 mm, respectively. In contrast, at the nozzle inclination angle of 60° , the deformation pattern transitions to a wave-like form with a considerably reduced maximum deformation of 7.39 mm. This phenomenon is attributed to the complex interaction of airflow dynamics within the furnace at this specific angle, which creates a more symmetric and stable flow field.

The 60° inclination angle optimizes flow and heat transfer characteristics, resulting in a distinct fountain flow within the secondary stagnation zone. This configuration enhances heat transfer efficiency and provides substantial support to the plate, effectively reducing deformation and improving thermal management. Additionally, the flow field at 60° exhibits symmetric recirculating flows with a maximum velocity difference of 28.1%, ensuring stable and predictable flow conditions. This stability is crucial for maintaining consistent heat treatment outcomes, whereas higher angles (e.g., 75°) result in disorganized flow fields and potential instabilities.

(2) At the nozzle inclination angle of 60° , optimum flow and heat transfer characteristics are obtained, resulting in a distinct fountain flow within the secondary stagnation zone. This configuration enhances heat transfer efficiency and provides substantial support to the plate, effectively reducing deformation and improving thermal management. Moreover, the flow field involves symmetric recirculating flows with a maximum velocity difference of 28.1%, ensuring stable and predictable flow conditions. This stability is crucial for maintaining consistent heat treatment outcomes, whereas greater nozzle inclination angles (e.g., 75°) result in disorganized flow fields and potential instabilities.

(3) Uniformity of temperature and pressure distributions is significantly high at the nozzle inclination angle of 60° . The average surface temperature reaches 463 K. Smooth peaks of pressure coefficient and minimized thermal stress ensure high processing quality. This contrasts with the non-uniform temperature distribution and high pressure observed at the nozzle inclination angles of 30° and 45° , which lead to significant temperature and pressure differences across the plate surface. High uniformity of temperature distribution at the nozzle inclination angle of 60° contributes

to a uniform effect of heat treatment on the plate.

(4) The appearance of secondary peaks in the distribution of Nu at the nozzle inclination angle of 60° further enhances heat transfer uniformity. The secondary peaks, located between dual secondary stagnation zones and nozzle units, contribute to a more balanced thermal profile across the plate surface. Combined with the smooth pressure distribution and stable flow field, the nozzle inclination angle of 60° represents an optimum condition, under which minimal deformation, uniform heat transfer, and high processing quality are obtained.

Based on the present study, three aspects are expected to be further investigated: (1) a comparison of heat transfer between plates of different materials, (2) effects of turbulent fluctuations on heat transfer within the confined space, and (3) development of a lab scale experimental model of the air-cushion furnace. Advancement in these interconnected aspects will deepen the understanding of the thermal-fluid dynamics related to the air-cushion furnace and improve corresponding industrial implementation.

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Nomenclature

Greek Symbols

μ	Dynamic viscosity of air (kg/m·s)
ν	Kinematic viscosity of air (m ² /s)
ρ	Air density (kg/m ³)
Ω_{ij}	Angular velocity tensor (1/s)
δ_{ij}	Kronecker delta
α	Jet flow inclination angle (°)
ε	Turbulence kinetic energy dissipation rate (1/s)

List of Symbols

A	Area of the heated surface (m ²)
b	Nozzle slot spacing (mm)
c_p	Pressure coefficient
$c_{p,avg}$	Average pressure coefficient
C_μ	Empirical constant
c	Specific heat (kJ/kg·K)
D_h	Hydraulic diameter (mm)
dl	Distance between lower nozzles and plate (mm)
du	Distance between upper nozzles and plate (mm)
e	Nozzle slot width (mm)
h	Jet-to-plate spacing (mm)
K_m	Fluid viscosity coefficient of turbulent kinetic energy
K_T	Fluid temperature coefficient
k	Turbulent kinetic energy(J)
k_f	Fluid thermal conductivity (W/m·K)
Nu_{avg}	Average Nusselt number
Nu_x	Local Nusselt number
P	Average pressure (N/m ²)
p_c	Wall pressure (N/m ²)

Pr	Prandtl number
Re	Reynolds number
R_{pj}	Plate-to-jet velocity ratio
Sc	Schmidt number
S_{ij}	Angular deformation rate tensor (1/s)
T	Average temperature(K)
T'	Fluctuating temperature components(K)
T_j	Jet temperature (K)
T_w	Wall temperature (K)
t	Time (s)
U_i	Average velocity (m/s)
u_i'	Fluctuating velocity components (m/s)
u_j	Mean jet exit velocity (m/s)
V	Plate velocity (m/s)
x	Nozzle-to-nozzle spacing (mm)
X	Coordinate in the X -direction (mm)
X_i	Coordinate direction
Y	Coordinate in Y direction (mm)
y^+	Non-dimensional distance between wall and first-layer nodes
Z	Coordinate in Z direction (mm)
TUI	Temperature Uniformity Index

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Optimization of heat transfer and deformation control in aluminum plate heat treatment: effect of nozzle inclination angle

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