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ENGINEERING TRANSACTIONS

ROZPRAWY INŻYNIERSKIE – TRAITE d'INGENIERIE

**QUARTERLY
VOLUME 74 ISSUE 3**



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WARSAW – METZ – POZNAN 2026

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Research Paper

Modeling Defect Detection in 3D-Printed Structures Using Ultrasonic Guided Waves

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Carbon fiber-reinforced thermoplastic (CFRTP) composites are widely utilized in 3D printing via the fused filament fabrication (FFF) process. However, defects within printed components can compromise their performance. This study investigates the application of ultrasonic non-destructive testing (NDT) techniques to structures manufactured through FFF. Common printing defects, such as weak interlayer bonding, surface imperfections, and internal cracks, are simulated using a finite element (FE) model. The model analyzes how guided ultrasonic waves interact with these defects. The proposed methodology incorporates digital transducer control and evaluates the energy distribution among incident, reflected, and transmitted wave modes. Three representative defect types are modeled, and the approach is applied to two structural geometries: plates and pipes. The study investigates the correlation between defect size and the behavior of energy reflection and transmission. A signal processing approach is applied, using transducers positioned symmetrically along the wave propagation axis to detect mode conversion and analyze wave reflections and transmissions. The phenomenon of mode conversion is analyzed in detail, and the results are validated by ensuring energy-balance consistency.

Keywords: ultrasonic testing, finite element simulation, guided waves, defect detection, non-destructive testing.



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1. INTRODUCTION

In recent years, 3D printing has undergone significant advancements across nearly all industrial branches, particularly in the automotive, aerospace, and healthcare industries. These industries have experienced rapid growth and diversification in printing technologies. The various techniques employed differ based

on the material used, the complexity of the part being printed, and considerations such as production and maintenance costs. For instance, methods such as selective laser melting (SLM) [1] and selective laser sintering (SLS) [2] are well-suited for metallic materials, while fused deposition modeling (FDM) [3] is commonly used for polymer filament printing. Despite these advances, numerous defects can still arise in printed components, stemming from a variety of causes. For example, defects may originate from materials themselves, such as low interfacial bonding between fibers and the matrix in composite structures [4, 5]. The fabrication process may also cause complications such as volumetric defects, porosities, cracks, and surface pitting [6]. Additionally, thermal effects during production can lead to problems, such as the formation of a thermally affected zone (TAZ) [7] notably in metal additive manufacturing processes involving welding. Given these potential defects, it is crucial to monitor and assess the integrity of such structures. Ultrasonic guided waves (UGWs) [8, 9] present a promising method for non-destructive testing (NDT) of these types of structures. They are highly effective for propagating over long distances with minimal attenuation, making them ideal for assessing structural integrity of components. To perform accurate NDT using UGWs, it is essential to understand the wave dispersion characteristics. The transducer must generate frequencies corresponding to specific modes indicated by the dispersion curves. Ultrasonic sensors are then used to measure the normal displacements on the surface of the structure being tested. Post-processing techniques, such as signal analysis using the Fourier transform, wavelets, or other methods, are essential for detecting and characterizing potential defects within a structure. Several researchers developed numerical models to simulate the propagation of UGWs in various structures. For example, DE LUCA *et al.* [10] utilized a finite element (FE) model in ABAQUS to study UGW propagation in a carbon fiber reinforced polymer (CFRP) laminated plate, analyzing how defect size and orientation impact structural integrity. CHUI *et al.* [11] focused on delamination detection in a 3D laminate composed of four composite layers. RHIMINI *et al.* [9] explored the interaction of UGWs with an embedded volumetric defect in a three-layer aluminum/epoxy/aluminum assembly, also using ABAQUS. In that study, post-processing involved applying a double Fourier transform to the data. In recent years, several researchers have started investigating defect detection in 3D-printed parts using ultrasound. Among them, EVERTON *et al.* [12] focused on identifying sub-surface defects through laser-generated ultrasound. CUMMINGS *et al.* [13] developed an in situ method for detecting defects as they arise in plastic parts produced by the FDM process, although post-processing of the data still presents significant challenges. CAMINERO *et al.* [14] employed ultrasound combined with the phased array technique to assess damage in laminated 3D-printed composites. Despite these efforts, the use of UGWs for defect

detection remains underexplored, and post-processing techniques for interpreting the results continue to be a major hurdle.

In this study, we aim to carry out a numerical non-destructive inspection of CFRTP composite parts using guided waves [15]. CFRTP composites are widely used in filament-based manufacturing processes due to their superior mechanical properties compared with those of pure thermoplastics [16, 17]. Additionally, the high prototyping flexibility and various printing parameters specific to the FDM process, such as raster angle and filament size, enable the fabrication of complex structures with CFRTP, making it a popular choice in industries such as automotive manufacturing and aerospace. Nevertheless, the stiffness of printed CFRTP composites can deteriorate due to imperfections such as ruptures, layer detachment, and notches, often caused by inappropriate printing parameters [18, 19].

In our analysis, the structures considered exhibit rectangular notches and cracks. To inspect these structures, we first generated dispersion curves for the material, representing the frequency-wavenumber pairs that could excite UGWs in our waveguide [20, 21]. Additionally, we plotted normalized displacement profiles with energy flow across the section by applying a spatial distribution of displacements over 10 cycles, using a Hanning window centered at the excitation frequency for weighting these displacements.

Based on these elements, we developed a numerical FE simulation of the UGW implementation process in composite plates and cylindrical pipes. We also simulated the acquisition of signals both before and after encountering a defect using FE software. By calculating reflection and transmission coefficients, we analyzed how defect size and position affect the selection of the appropriate ultrasonic mode for inspection. As the defect size increases, the reflected wave packet also increases. Moreover, when the defect is asymmetrically positioned with respect to the median plane of the plates, mode conversion occurs, a phenomenon we addressed using a technique based on summing and subtracting the signals captured by the ultrasonic transducers.

2. THEORY OF UGWs

2.1. DISPERSION NETWORK

We analyze the propagation of UGWs through a composite waveguide. The material under consideration is a CFRTP commonly used in 3D printing, particularly in the FDM process. UGWs enable comprehensive scanning of the entire waveguide structure. However, to achieve this, it is essential to understand the dispersion patterns of these waves. Since UGWs are highly frequency-dependent, even a small frequency change can significantly alter their behavior. These dis-

person patterns, known as dispersion curves, identify the frequency (f) and wave number (k) pairs that can generate UGWs within the composite waveguide. Various methods have been applied to characterize this dispersive behavior. These approaches include analytical techniques to find the zeros of a function [22], as well as numerical techniques to plot these curves [23]. In our study, we implemented a MATLAB program that uses a numerical spectral method [22] to plot dispersion curves for composite materials. The method applies a spectral scheme to approximate the differential equations governing motion and the associated boundary conditions. By determining the eigenvalues (frequencies) for a specified range of wavenumbers and using a vibrational state-based mode separation algorithm, we obtain dispersion curves in terms of frequency and wavenumber. Additionally, we can represent these curves in other forms, such as phase velocity V_p versus frequency or group velocity V_g versus frequency, which provide insights into the energy transport characteristics of the wave packet.

We consider two waveguides. The first is a 4 mm thick CFRTP composite plate (Fig. 1a) and the second is a CFRTP composite pipeline with an inner radius of 50 mm and an outer radius of 56 mm (Fig. 1b). The values of the components of the elastic stiffness tensor and the density of the composite are shown in Table 1. We note that the stiffness constants are expressed in the Cartesian system.

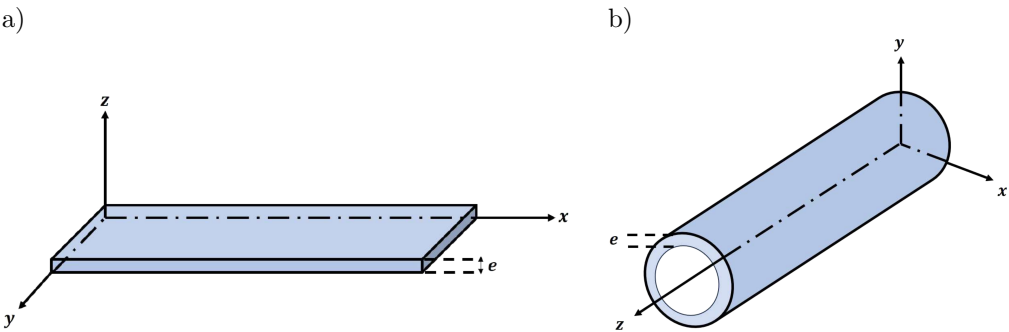


FIG. 1. a) Planar waveguide, b) cylindrical waveguide; (x, y, z) -Cartesian coordinate system, e -waveguide thickness.

TABLE 1. Material properties of the CFRTP composite [4].

Density [g/cm ³]	$C'_{11} = C'_{22} = C'_{33}$	$C'_{12} = C'_{13}$	$C'_{55} = C'_{66}$	C'_{44}	C'_{23}
1.6	43.3294	6.4745	9.51	18.4291	6.4712

Figure 2a presents the dispersion curves for the CFRTP composite plate in the frequency-thickness product versus phase velocity plane. The plate exhibits Lamb-wave modes, including symmetric modes (S_n) and antisymmetric modes (A_n) with respect to the propagation planes. The fundamental modes, denoted as S_0 and A_0 , are commonly used in UGW non-destructive testing [24, 25].

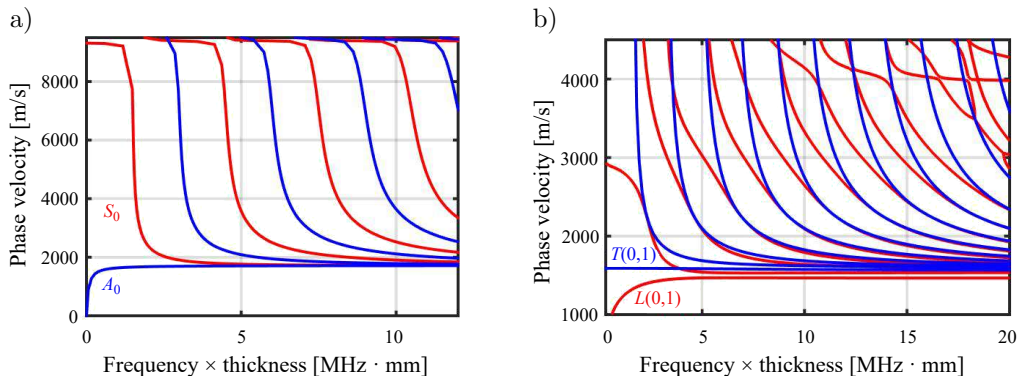


FIG. 2. Dispersion curves generated using the spectral method for a CFRTP composite plate with a 4 mm thickness (a) and a cylindrical pipe with an inner radius of 50 mm and an outer radius of 56 mm (b).

Figure 3 illustrates the vibrational states (von Mises stress) of the plate under these modes. We can see that the symmetrical S_n mode is symmetric with respect to the u_x displacement (displacement in the direction of propagation x) and antisymmetric with respect to the u_y displacement (displacement normal to the direction of propagation). The opposite behavior is observed for the antisymmetrical mode A_n .

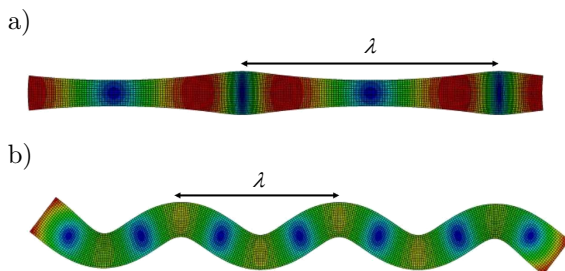


FIG. 3. Vibrational states of the plate: a) symmetrical Lamb mode S_n , b) antisymmetrical Lamb mode A_n . λ represent the wavelength.

Figure 2b shows the dispersion curves of the CFRTP composite cylindrical pipe. Due to the pipe curvature, the symmetry in vibration is lost [26], resulting in longitudinal modes $L(0,n)$ and torsional modes $T(0,n)$. Additional modes, such as flexural modes $F(m,n)$, are also present; however, these are excluded from our study. The vibration states of the pipe under the various modes are displayed in Fig. 4.

Here, n represents the mode index, determined by the order in which the mode appears at low frequencies, and m represents the order of the flexural mode.

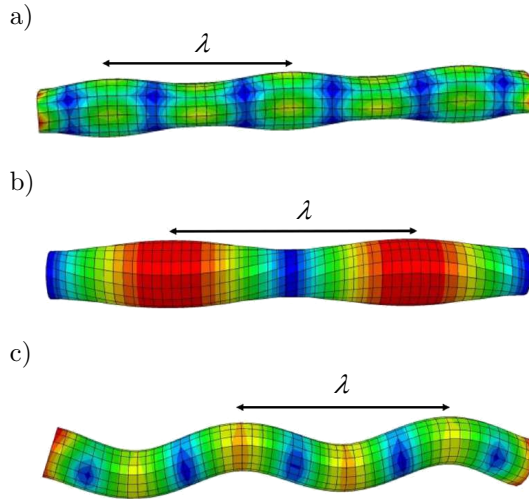


FIG. 4. Vibrational states of the cylindrical pipe: a) longitudinal mode $L(0, n)$, b) torsional mode $T(0, n)$, c) flexural mode $F(m, n)$.

2.2. FIELD OF DISPLACEMENT

NAYFEH *et al.* [27] proposed the expression for displacement fields that describe the propagation of a harmonic wave in plane and cylindrical structures that are homogenous and anisotropic in shape:

$$(2.1) \quad \begin{cases} (u_x, u_y, u_z) = \sum_{q=1}^6 (1, V_q, W_q) U_{1q} e^{ik(x + \alpha_q z - ct)}, \\ (u_r, u_\theta, u_z) = U_\beta^{mn}(r) \Theta_\beta^{mn}(m\theta) e^{i(\omega t - k^{mn} z)}, \end{cases}$$

where q is the summation index, x is the UGW propagation direction, c is the wave phase velocity, k is the wavenumber, U_{1q} , V_q , W_q are the displacement amplitudes [27], and α_q is the ratio between the wavenumbers in the x - and z -directions. Furthermore, U_β^{mn} and the sinusoidal function Θ_β^{mn} are the radial and circumferential components of velocity, respectively.

3. NUMERICAL SIMULATION

3.1. FORMULATION

We employ the finite element method (FEM), where the structure is discretized into FEs. Within each element, the displacement field is approximated using interpolation functions at each node, resulting in the so-called nodal displacements. The global equation system is then constructed by assembling these

individual element displacements. For the undamped case, the system is expressed as follows:

$$(3.1) \quad [M]\{\ddot{U}\} + [K]\{U\} = \{F\},$$

where $[M]$ represent the global mass matrix, $[K]$ is the global stiffness matrix, $\{U\}$ is the global displacement vector, $\{\ddot{U}\}$ is the global acceleration vector, and $\{F\}$ is the global force vector.

In order to solve Eq. (3.1), we apply Newmark's method [28], which approximates the spatial partial derivatives. By utilizing extended mean value theorem, Newmark's method provides the following approximation to solve the first time derivative:

$$(3.2) \quad \{\dot{U}_{t+\Delta t}\} = \{\dot{U}_t\} + \Delta t\{\ddot{U}_\gamma\}$$

with

$$(3.3) \quad \{\ddot{U}_\gamma\} = (1 - \gamma)\{\ddot{U}_t\} + \gamma\{\ddot{U}_{t+\Delta t}\}.$$

Since we have partial derivatives with respect to time in the acceleration vector, the theorem can also be written as

$$(3.4) \quad \{U_{t+\Delta t}\} = \{U_t\} + \Delta t\{\dot{U}_t\} + \Delta t^2((1 - \beta)\{\ddot{U}_t\} + \beta\{\ddot{U}_{t+\Delta t}\}),$$

where γ and β are the Newmark integration parameters, and Δt is the time step.

3.2. FE ANALYSIS MODEL

We examine the propagation of UGWs in two types of CFRTP structures. The first structure is a homogeneous plate with a length of $L = 500$ mm and a thickness of $e = 4$ mm. The second structure is a hollow cylindrical pipe with an inner radius of 50 mm, an outer radius of 56 mm, and a length of $L = 600$ mm. We consider a linear homogeneous dynamic calculation with no damping. The waveguide is initially not subjected to any stress, and the contact between the sensors and the structure is considered to be perfect. The small-perturbation assumption is also applied.

This study addresses a spatial-temporal problem and is simulated in ABAQUS. Details regarding simulation the parameters, including the spatial and temporal discretization steps Δx and Δy , will be provided later. To mesh our models, we select a control frequency of $f = 500$ kHz (based on the dispersion curves) and use an element size of 0.5 mm, which meets the following condition [29]:

$$(3.5) \quad \max(\Delta x, \Delta y) \leq \frac{\lambda_{\min}}{10},$$

where $\lambda_{\min}$ is the minimum wavelength [29].

This criterion specifies that, to ensure computational convergence, the element size used to discretize the waveguide should be no larger than one-tenth of the smallest wavelength of the propagating modes in the structure.

For time discretization, ensuring convergence in ultrasonic simulations demands that the shortest wavelength within the UGW pulse needs to be divided into at least 20 nodes. For a frequency of $f = 500$ kHz, we get a temporal step of $0.1 \mu\text{s}$, satisfying the following condition [30]:

$$(3.6) \quad \Delta t = \frac{1}{20f},$$

where f is the mode frequency selected to produce ultrasonic guided-wave mode.

3.3. UGW GENERATION

To numerically generate UGWs in waveguides, several researchers have modeled the excitation signal as a specific point force, typically represented by a sinusoidal waveform modified by a Hanning window [31, 32]. Other studies have employed piezoelectric transducers to generate UGWs by applying nodal loads around the circumference of a circle [32]. Additionally, another approach incorporated the Snell–Descartes laws to determine the wave incidence angle [32]. Despite the advantages of these methods, using a cyclic force can lead to edge reflection issues. This problem has been addressed in two ways: either by treating the boundary conditions as absorbing (resulting in no reflection) or by applying post-processing techniques to the acquired signals. To mitigate these issues and ensure the excitation of the desired mode, RHIMINI *et al.* [9] utilized analytical displacements through the thickness of an isotropic multilayer laminate to effectively excite the Lamb-wave modes.

In this study, we focus on the latter approach to model the propagation of UGWs in both an anisotropic plate and a cylindrical pipe. Excitation based on analytical displacements scaled by acoustic power is applied. For the plate, we will generate the antisymmetrical A_0 mode according to previous research that has shown that the S_0 and A_0 modes are the most commonly utilized in ultrasonic inspection, as they are situated at low and high frequencies, respectively. Additionally, the mode conversion phenomena associated with these modes are relatively straightforward [24, 25] and both can be generated at low frequency values. For the cylindrical pipe, the first longitudinal mode $L(0, 1)$ is selected [33, 34].

Equation (2.1) provides the analytical displacement field that contains two components: the first component describes the spatial coordinate and the second one the temporal coordinate. The spatial component describes the mode shape generated across the thickness of the waveguide. For example, the chosen A_0

mode shows an antisymmetric longitudinal displacement (blue curve in Fig. 5a) and a symmetric transverse displacement throughout the plate thickness (red curve in Fig. 5a). Figure 5b displays the displacement profiles for the chosen longitudinal mode $L(0, 1)$.

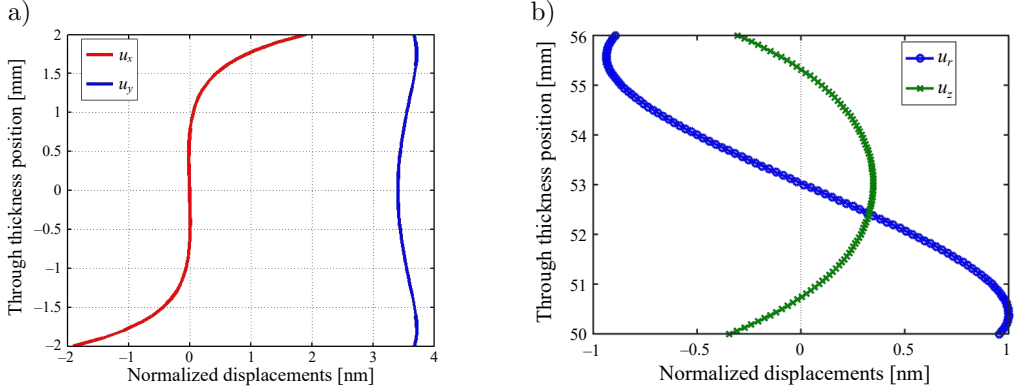


FIG. 5. Normalized analytical displacements scaled by acoustic power of the selected mode A_0 of the composite plate ($f = 500$ kHz, $k = 1431$ m $^{-1}$) (a) and the $L(0, 1)$ mode of the composite pipe corresponding to the pair ($f = 500$ kHz, $k = 67$ m $^{-1}$) (b).

The temporal component is represented by the exponential term, resulting in the analytical spatial displacements oscillating according to either a cosine or sine function. The displacement fields can be expressed directly as functions of cosine or sine in isotropic materials [27]. However, in the case of highly anisotropic materials, such as the CFRTP in our study, it is not possible to definitively determine if the displacements are described by a cosine or sine dependence. To address this, during mapping spatial displacements at a particular position x and constant time t , the exponential term is solely dependent on the z -coordinate ($e^{i\varphi} = \cos(\varphi) + i\sin(\varphi)$). Thus, an imaginary displacement corresponds to a sine function, whereas a real displacement corresponds to a cosine function. For the plate, the displacement u_x follows a sine function, while the displacement u_z follows a cosine function (mode A_0). The same approach is applied to the $L(0, 1)$ mode.

The spatial displacement distribution is used for 10 cycles, with weighting based on a Hanning window centered around the excitation frequency [35]. The use of a Hanning window excludes disturbances from the incoming signal (Fig. 6).

3.4. INSPECTION OF THE PLATE

On the left side of the plate, UGWs are induced to travel through its thickness. Transducers are placed on the plate surfaces $y = \pm h$ to track their move-

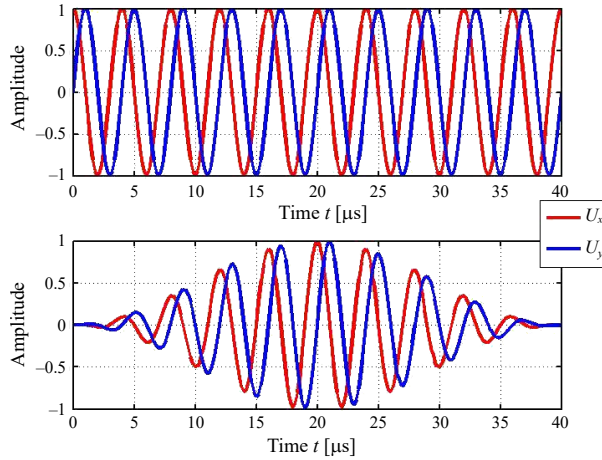


FIG. 6. Hanning window applied to the incident signal of the CFRTP composite plate.

ment along the plate, before and after the defect. Proper positioning of these transducers with respect to the plate edges and the defect is critical. UGWs consist of propagating modes, which are utilized in inspection, and evanescent modes [36, 37]. Due to their pronounced attenuation in the material, evanescent modes are not viable for ultrasonic non-destructive testing of large-scale structures. Evanescent modes become evident when reflections occur due to the plate edges or defects. Consequently, ultrasonic sensors placed close to the defect or the edge of the structure may pick up signals not only from the propagating modes but also from the evanescent modes. This can complicate the interpretation of the results, particularly if there is insufficient information regarding the characteristics of the evanescent waves. In this study, the transducers are positioned to maintain a minimum distance of at least 10 mm from both the defect and the plate edge. The selected transducers measure the displacements u_x and u_y at 180 nodes along the plate interface, with the nodes spaced evenly at 0.5 mm (mesh step). In scenarios where the defect location is unknown (as in real cases), it is essential to conduct a thorough investigation of the evanescent modes to understand and effectively address near-field wave-defect interaction phenomena.

Temporal displacements at the plate upper and lower surfaces, before the defect, are acquired using transducers 1 and 2. After the defect, transducers 3 and 4 record the temporal displacements at the upper and lower surfaces of the plate (Fig. 7). Transducers 1 and 2 record signals that encompass two wave packets. The first one originates from incident wave and the second one is due to wave reflection. The signals collected by transducers 3 and 4 corresponds to the transmission of the wave across the defect. The arrangement places transducer 1 opposite transducer 3, and transducer 2 opposite transducer 4. This configura-

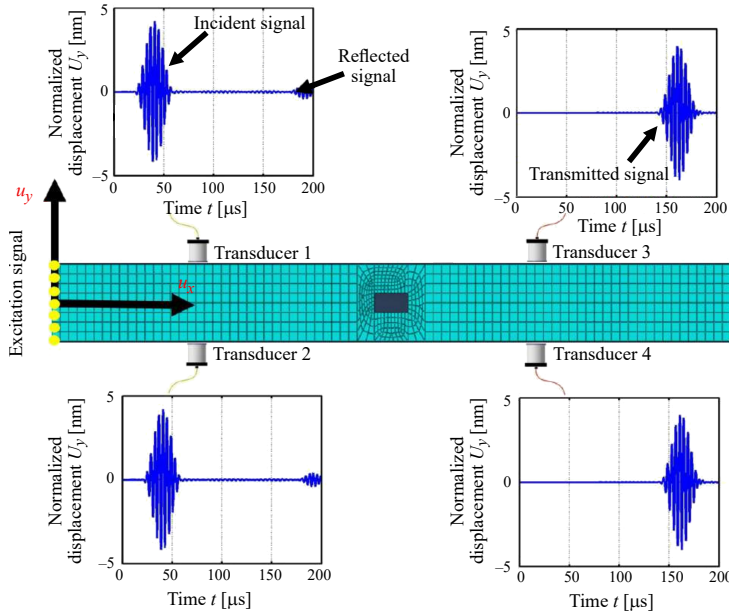


FIG. 7. Inspection of a symmetric defect in the composite CFRTP plate using the selected A_0 mode ($k = 1431 \text{ m}^{-1}$, $f = 500 \text{ kHz}$); $p = 2 \text{ mm}$ and $w = 1 \text{ mm}$.

tion allows us to determine the defect position, by assessing the time of flight between the incident and reflected waves, since we know in advance the phase velocity of the emitted wave. Additionally, the nature of the modes in the reflected and transmitted wave packets can be identified by examining the symmetry and antisymmetry characteristics of the S_0 and A_0 modes. To distinguish the modes present in the reflected and transmitted waves, some researchers utilized tools such as the 2D FFT of the recorded signals and the wavelet transform [9, 28].

In the initial step, the notch is assumed to be symmetrically located with respect to the plate median plane. The presented configuration prevents mode conversion. We first excite the A_0 mode, and the wave reaches the defect. The energy of this mode is partly reflected and partly transmitted, while the A_0 mode nature is preserved throughout the interaction. Subsequently, an asymmetric positioning of the notch relative to the plate median plane is considered. In this scenario, as the mode interacts with the defect, the reflected and transmitted wave packets include energy contributions from both the incident mode and the converted mode due to mode conversion phenomena. We therefore focus on assessing the influence of the notch dimensions and position on mode conversion and energy distribution.

Initially, we vary the defect parameters (length p and height w) and compute the energy coefficients of the reflection and transmission. The accuracy of the

calculations presented is monitored through the energy balance. Equation (3.7) and Eq. (3.8) report the formulas of the energy coefficients:

(3.7)
$$R = \frac{\phi_r}{\phi_i},$$

(3.8)
$$T = \frac{\phi_t}{\phi_i},$$

where ϕ_r , ϕ_t , and ϕ_i are the reflected, transmitted, and incident energy flows, respectively.

The reflection and transmission coefficients and the associated error are presented in Table 2. Only the A_0 and S_0 modes coexist in the plate because a frequency below the A_1 mode cut-off frequency is chosen. The tested specimens exhibit differences in defect length and height. As the defect length increases ($p \uparrow$), indicating a tendency toward a longitudinal crack, the reflected energy rises while the transmitted energy decreases, maintaining an overall energy balance close to unity. Similarly, increasing the defect height ($w \uparrow$, transverse crack) results in a higher reflection coefficient and a lower transmission coefficient.

TABLE 2. Energy coefficients describing the interaction of the first two fundamental modes A_0 and S_0 of the composite plate with defects of various dimensions.

Selected modes	Defect parameters (p = length, w = height) [mm]	Reflection coefficient (R)	Transmission coefficient (T)	Energy conservation ($R + T$)	Error [%]
Fundamental antisymmetric mode A_0 of the plate ($k = 1431 \text{ m}^{-1}$, $f = 500 \text{ kHz}$)	$p = 2, w = 1$	0.095	0.9093	1.0043	0.43
	$p = 5, w = 1$	0.7845	0.2364	1.0209	2.09
	$p = 1, w = 1$	0.0076	0.9927	1.0003	0.03
	$p = 1, w = 3$	0.8956	0.1134	1.0090	0.90
Fundamental symmetric mode S_0 of the plate ($k = 621 \text{ m}^{-1}$, $f = 500 \text{ kHz}$)	$p = 2, w = 1$	0.086	0.907	0.993	0.7
	$p = 5, w = 1$	0.8502	0.1672	1.0174	1.74
	$p = 1, w = 1$	0.0023	0.9966	0.9989	0.11
	$p = 1, w = 3$	0.8534	0.1542	1.0076	0.76

Figure 8 shows the scan of a plate that contains a defect asymmetrically positioned with respect to the median axis x . Two reflected signals are observed by transducers 1 and 2, while transducers 3 and 4 display two transmitted signals. This is due to the phenomenon of mode conversion. The A_0 mode generated initially interacts with the defect, reflecting a portion of its energy that depends on the defect dimensions. In this case, the reflected energy is split into two wave packets, a pattern also observed in the transmitted signals. To identify the type

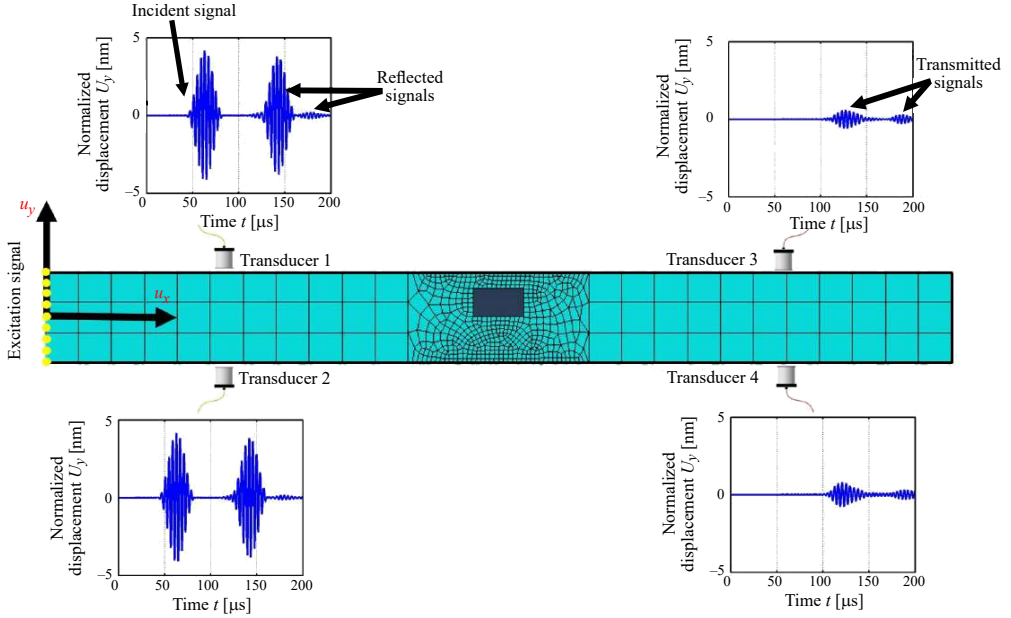


FIG. 8. Inspection of an asymmetrical defect in the composite CFRTP plate using the selected A_0 mode ($k = 1431 \text{ m}^{-1}$, $f = 500 \text{ kHz}$); $p = 2 \text{ mm}$, $w = 1.5 \text{ mm}$.

of reflected mode, several authors have applied the Fourier transform or wavelet transform [9, 27] to the recorded signal, enabling a transformation from the (X, t) domain to the (k, f) domain. This (k, f) domain represents the numerically obtained dispersion curves. By comparing these curves with analytically derived curves, the reflected and transmitted modes can be identified. In this study, we propose a new method based on the symmetry of propagating modes to characterize the mode conversion phenomenon. Specifically, a symmetric mode is defined by symmetric longitudinal displacement ($u_x(+d) \times u_x(-d) > 0$) and antisymmetric transverse displacement ($u_y(+d) \times u_y(-d) < 0$), while an antisymmetric mode exhibits the opposite behavior ($u_x(+d) \times u_x(-d) < 0$, $u_y(+d) \times u_y(-d) > 0$) (see Fig. 3). By summing the signals from transducers 1 and 2 and separately from 3 and 4, the antisymmetric modes are suppressed, leaving only the symmetric modes. The symmetric modes can be filtered out by subtracting the signals from transducers 1 and 2 and 3 and 4, leaving only the antisymmetric modes. Through this approach, the reflected captured signal contains a component that corresponds to the A_0 mode. The other component is due to the conversion between A_0 and S_0 modes. A similar observation applies to the transmitted wave packet.

We then calculate the energy coefficients for reflection and transmission and report the results in Table 3. Two signals are observed in the reflected wave packet due to the defect asymmetric location relative to the propagation axis.

TABLE 3. Energy coefficients describing the interaction of the A_0 and S_0 modes with an asymmetrically positioned defect.

Mode	Reflection coefficient (R)	Transmission coefficient (T)	Energy conservation ($R + T$)	Error [%]
Fundamental antisymmetric mode A_0 of the plate	$0.310 A_0$ $0.227 S_0$	$0.309 A_0$ $0.183 S_0$	1.029	2.9

By applying the summation and subtraction technique, we determine the proportion of the reflected energy that retained the original mode (A_0 , $R = 0.8334$) and the proportion that underwent mode conversion (A_0 – S_0 , $R = 0.023$). The defect dimensions play a role in determining these values. Similar conversion results were obtained for the transmitted wave packet.

Next, we analyzed the case of an inclined defect with an inclination angle of 45° relative to the propagation axis. Figure 9 shows the scan of the composite plate with an inclined defect. In the reflection region (transducers 1 and 2), two reflected wave packets were observed, and a similar pattern also appears in the transmission region (transducers 3 and 4). This indicates that an inclined defect induces mode conversion. To quantify the reflected and transmitted energy, we applied the summation and subtraction method (as described previously) and calculated the energy coefficients, which is presented in Table 4.

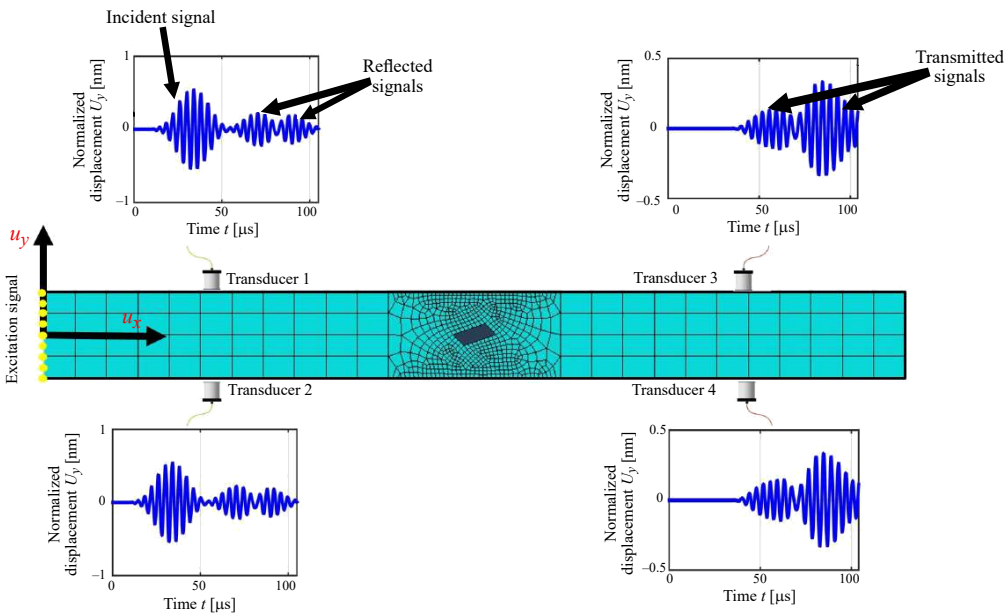


FIG. 9. Inspection of an inclined defect in the composite CFRTP plate using the selected A_0 mode ($k = 1431 \text{ m}^{-1}$, $f = 500 \text{ kHz}$); $p = 2 \text{ mm}$, $w = 1 \text{ mm}$.

TABLE 4. Energy coefficients describing the interaction of the A_0 and S_0 modes with the inclined defect.

Mode	Reflection coefficient (R)	Transmission coefficient (T)	Energy conservation ($R + T$)	Error [%]
Fundamental antisymmetric mode A_0 of the plate	0.8334 A_0 0.023 S_0	0.08 A_0 0.048 S_0	0.9764	2.32

The results indicate a clear presence of A_0 – S_0 mode conversion in both the reflected and transmitted wave packets. Additionally, the inclined defect resulted in an almost equal distribution of reflected and transmitted energy.

3.5. INSPECTION OF THE PIPE

We now consider a cylindrical pipe with a rectangular notch defect. We generate a longitudinal $L(0,1)$ mode along the pipe radius. Two transducers are used: one to capture the reflected energy and the other to capture the transmitted energy. We observe two reflected and two transmitted wave packets (Fig. 10), suggesting the occurrence of mode conversion [32].

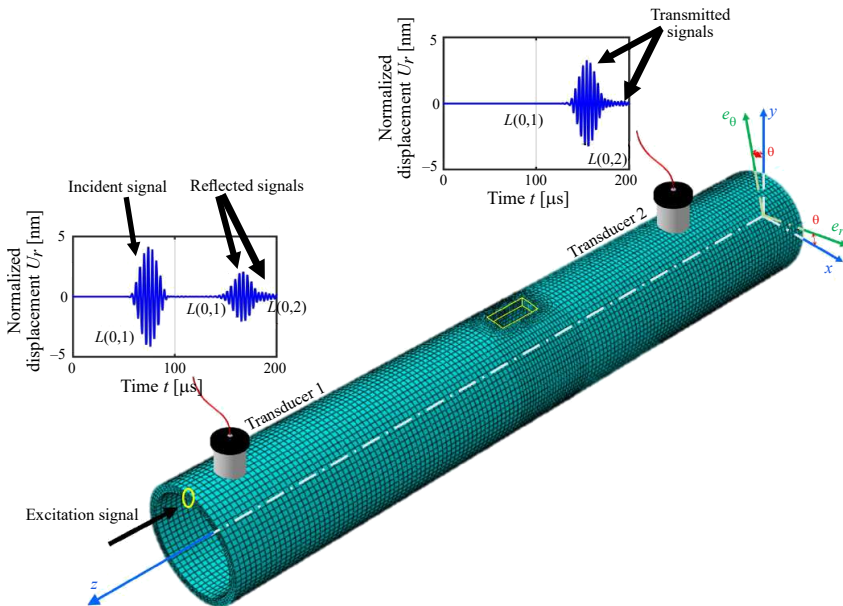


FIG. 10. Scan of the CFRTP composite cylindrical pipe using the longitudinal $L(0,1)$ mode, ($f = 500 \text{ kHz}$, $k = 67 \text{ m}^{-1}$) with a defect dimension ratio of $p/w = 3$.

The cylindrical geometry of the pipe disrupts mode symmetry (as shown by the dispersion curves, Fig. 2b), which prevents the use of the previously

proposed summation and subtraction method. To address this, we analyze the phase velocity to identify the modes present in the reflected and transmitted wave packets. Given the analytical phase velocity for the generated mode with $f = 500 \text{ kHz}$ and $k = 65 \text{ m}^{-1}$ is $V_p = 4688 \text{ m/s}$, we calculate the travel time between the incident and reflected waves [38]. Using the known distance between the transducer and the defect, we find a numerical phase velocity of $V_p = 4679 \text{ m/s}$, corresponding to the $L(0,1)$ mode (in close agreement with the analytical prediction). A second phase velocity is $V_p = 9809 \text{ m/s}$, aligning with the $L(0,2)$ mode (analytical value $V_p = 9817 \text{ m/s}$). We then compute the energy coefficients for different length-to-height (p/w) and height-to-length (w/p) ratios of the defect, and the results are presented in Fig. 11. Figure 11a shows that as the p/w ratio increases (indicating a more circumferential crack), the reflected energy rises while the transmitted energy decreases, maintaining an energy balance close to unity. Conversely, in Fig. 11b, an increase in the w/p ratio (indicating a more longitudinal crack) results in a decrease in reflected energy and an increase in transmitted energy. Thus, when the defect type is unknown, the defect dimensional ratio and therefore its nature can be inferred from the energy coefficient trends shown in Fig. 11. Experimentally, in order to inspect plates and pipelines using UGWs, several authors have used PZT transducers [39, 40] (known for their low weight and ease of bonding to the waveguide surface without signal loss) to detect signals before and after the defect.

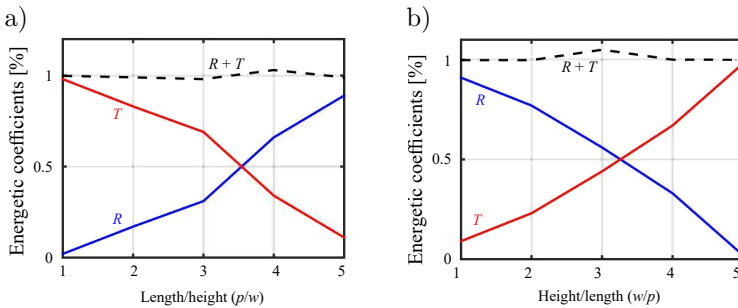


FIG. 11. Variation of the reflection and transmission energy coefficients as a function of the p/w ratio (a) and the w/p ratio (b) for the cylindrical pipe.

Several recent studies confirm the importance of guided-wave-based techniques for crack detection in plates and pipelines. The work of LIU *et al.* [41] stands as a major methodological reference for imaging circumferential cracks in pipes, notably through the use of full-wavefield data. More recently, SILITONGA *et al.* [42] provided a comprehensive review of Lamb-wave-based localization methods, detailing the conditions for selecting modes and frequencies to achieve optimal defect detection. SHAH *et al.* [43] demonstrated, through both experimental and numerical approaches, the capability of guided waves to detect

the growth of internal cracks in polyethylene pipes. ZHANG *et al.* [44] subsequently showed that the use of electromagnetic acoustic transducers (EMATs) and circumferential shear horizontal (SH) modes enables the identification of defects hidden beneath coatings or supports, paving the way for truly non-contact inspections. Finally, MARUYAMA *et al.* [45] deepened the understanding of reflection and transmission phenomena at junctions in fluid-filled pipelines, highlighting the need for precise modeling to correctly interpret wave-defect interactions in complex geometries.

4. CONCLUSION

In this study, we simulated the propagation of UGWs in composite CFRTP components produced using the FDM 3D printing technique, employing an FEM. These structures were designed to include internal volumetric defects, such as rectangular notches and cracks, which are characteristic of FDM 3D printing. We detailed the selection of various spatial and temporal discretizing parameters, and the numerical models were created by generating analytical displacements with a Hanning window weighting across the waveguide thickness. Strategically positioned transducers captured the reflected and transmitted signals, ensuring that only propagating waves were recorded. We implemented a novel signal-processing technique based on the summation and subtraction of the captured signals to differentiate between the various modes detected. This approach proved effective for analyzing mode conversion phenomena. The accuracy of the results was evaluated through the energy balance, specifically by calculating the reflection and transmission energy coefficients. Our findings indicated that these energy coefficients are significantly influenced by the dimensions of the defects. We modeled three types of defects (positioned symmetrically or asymmetrically in relation to the propagation direction, as well as inclined defects) within composite plates and cylindrical pipes. The results demonstrate the potential of UGWs to inspect parts with internal defects arising from the FDM 3D printing process. We also conducted a study on defect dimensions and reported a novel method for determining the modal characteristics during mode conversion according to the symmetry criterion of the recorded displacements at the surfaces. Looking ahead, we will explore correlations between defect parameters and the mechanical properties of the structures, encompassing all types of modes, including both propagating and evanescent modes.

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CONFLICT OF INTEREST

The authors declare that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHORS' CONTRIBUTIONS

Ismaine Zitouni conceptualized the study and wrote the original draft. Hassan Rhimini performed the analysis and contributed to data interpretation. Abdelkerim Chouaf contributed to data interpretation. All authors reviewed and approved the final manuscript.

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Research Paper

Investigation of the Effect of Fin Geometry on the Thermal Performance of Microelectronic Devices

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This study systematically evaluates the optimal heat sink geometry for enhanced thermal performance in electronic cooling applications. ANSYS-Icepak and COMSOL Multiphysics software assessed four distinct fin geometries (square, rectangular, circular, and conical), all uniformly sized for accurate comparison. Vital performance parameters were analyzed, including maximum temperature reduction, pressure drop, and airflow velocity. These indicators provide a holistic assessment of cooling efficacy and aerodynamic characteristics. The findings highlight the effectiveness of rectangular fins, which significantly reduce maximum temperatures due to their efficient balance of conductive and convective heat transfer. At the same time, conical fins display lower pressure drops and circular fins achieve higher airflow velocities, these attributes do not consistently enhance cooling capabilities. The importance of this study resides in its provision of definitive guidance for selecting fin geometry in designing efficient heat sinks, crucial for electronic devices. This knowledge is particularly vital for developing advanced cooling systems in compact, high-power electronics, underscoring the significant impact of fin geometry on overall thermal management efficiency. Quantitative data supporting these findings are available in the full study.

Keywords: fin geometry, electronic cooling, heat sink, thermal management, ANSYS-Icepak.



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1. INTRODUCTION

The progress in electronic chip technology has reached a stage where highly potent chips can now be accommodated within more compact packages [1]. Consequently, the resulting increase in power densities in these devices makes it imperative to implement more efficient thermal management solutions to prevent

these devices from potentially surpassing their maximum allowable operating temperatures. To mitigate heat dissipation from the components, it is conceivable to incorporate fins made of materials exhibiting high thermal conductivity [2, 3]. However, to achieve even greater heat dissipation rates, forced air or liquid cooling techniques involving these fins are employed, although these methods entail the utilization of additional components such as pumps or fans. Such approaches can lead to increased system costs and physical dimensions, making them potentially unsuitable for all applications [4]. Passive approaches, such as employing phase-change materials (PCMs), provide a method to delay the accumulation of sensible heat during a chip's operational cycle. These PCMs are composed of materials characterized by a substantial latent heat of fusion and are selected in such a way that their melting point is lower than the maximum operational temperature of the electronic device components. Instead of experiencing a gradual increase in sensible temperature, the PCM undergoes a phase change, transitioning from a solid to a liquid state by absorbing the heat generated by the chips [5]. As long as the PCM has not completely transformed into a liquid, the assembly will maintain a temperature close to the material's melting point. It is only when the latent heat storage within the PCM is exhausted will the temperature begin to rise once more. This approach is well-suited for devices with established duty cycles wherein the PCM undergoes a phase change, melts during operation, and resolidifies during shutdown.

Due to the inherent limitations in thermal conductivity of PCM materials, they may not efficiently dissipate heat quickly enough to prevent chip overheating [6]. To enhance the effective thermal conductivity of the system, fins made of highly conductive materials, such as aluminum, are introduced. These fins aid in the efficient transfer of heat from the chip to the PCM. Consequently, the addition of fins results in an increase of thermal conductivity, albeit at the expense of some latent heat storage capacity [7–10]. In the cooling enhancement of electronic devices, heat sinks are used to achieve optimum cooling performance in the electronics industry [8–17]. The challenge for designers lies in effectively dissipating the heat generated by the components while simultaneously maintaining the component's temperature at the necessary level to ensure reliable system operation [18]. Over the past two decades, researchers have devised a solution to improve heat dissipation by using different shapes of micro pin-fin heat sinks [19–22]. The objective is to achieve the most efficient cooling performance for the designed system, taking into account both cost and size considerations. To pursue this goal, various geometries must be investigated to evaluate the influence of heat sink designs on cooling performance. JALAL *et al.* [23] numerically investigated the impact of thermal performance of micro pin fins. Several fins with different configuration such as in-line and staggered layouts were analyzed. The results indicated an enhancement in cooling performance ranging from 7.9 %

to 9.3 % with an increase in the number of fins from 902 to 1312, and an improvement of 10.2 % to 11.1 % when the number of fins increased from 902 to 1640.

OGUZHAN *et al.* [24] conducted experimental investigations to assess the cooling performance of plate-fin and pin-fin heat sinks utilizing both water (as the base fluid) and Al_2O_3 nanofluid as cooling mediums. For the plate fin-heat sink, the results showed that the maximum enhancement in heat transfer, compared to an unmodified surface, was approximately 64.25 % for the base fluid and 82.8 % for the nanofluid. EKPU [25] studied the influence of fin arrangement on the thermal performance of microelectronic devices by using ANSYS finite volume design software. The findings clearly demonstrated that the configuration of heat sink fins has a substantial influence on both the thermal resistance and the overall efficiency of the microelectronic device. Furthermore, the rectangular fin configuration showed advantages over the other fin configurations studied. TANDA [26] conducted experimental research to explore heat transfer and pressure drop in a rectangular channel equipped with arrays of diamond-shaped fins. The study investigated both in-line and staggered fin arrangements. The results demonstrated that incorporating diamond-shaped fins resulted in enhanced heat transfer compared to a rectangular channel lacking fins. This improvement reached a maximum of 4.4 times when maintaining an equal mass flow rate and 1.65 times when keeping the pumping power constant.

DEBICH *et al.* [27] conducted a parametric study to assess the influence of various factors on the overall thermal efficiency of a PCM heat sink assembly. The study utilized plate-style fins and varied parameters such as the PCM type, the number of cavities in the assembly, and the materials of both the matrix and the fins. According to their findings, the geometry of the fins had the most significant effect on the chip's ability to transfer heat into the PCM. CHOUDARI *et al.* [28] performed a numerical analysis to examine different cross-sectional shapes for fins encircling a high-temperature battery. The study considered five distinct shapes, each with different PCM thicknesses and fin counts. Their findings revealed that by implementing the optimal design, the temperature of the battery was reduced by 9.28 °C. SHATIKIAN *et al.* [29] conducted a numerical investigation focusing on a design featuring internal fins with exposed tips, allowing contact with the surrounding air. In their study, they varied both the thickness of the fins and the proportion of the fins exposed to the surrounding air. The computational findings indicated that the transient phase-change process, as quantified by the volume melt fraction of the PCM, is influenced by the thermal and geometrical characteristics of the system, including the thickness and exposed portion of the fins.

JAWORSKI [30] observed that to achieve a more uniform heat distribution, employing a heat sink featuring thin-walled pipes filled with paraffin wax was advantageous. This design offered a significant surface area directly in contact

with the PCM. When compared to a larger assembly filled with wax, the results showed that this approach lowered the heat resistance between the fins and the wax. The enlarged contact area expedited the rapid distribution of higher heat density into the wax, resulting in a noticeable improvement, particularly in situations with higher heat flux. KAMKARI and SHOKOUHMAND [31] conducted an experimental study that focused on the operation of a PCM heat sink assembly with fins located on the side wall. Their study involved varying the number of fins, and it was found that the configuration with three fins exhibited the highest overall effectiveness. PAKROUH *et al.* [32] optimized a square housing with pin fins by varying the number of fins from 0 to 100 to find the ideal fin-to-wax ratio. They considered parameters such as the number of fins, fin height, fin thickness, and base thickness for optimization. The simulations factored in natural convection and PCM volume changes during melting at different critical temperatures (50 °C, 60 °C, 70 °C, and 80 °C). For a critical temperature of 50 °C, the optimal configuration featured 100 pin fins with a thickness of 4 mm, resulting in a 60.61 % PCM volume fraction.

In a study conducted by BONDAREVA and SHEREMET [33], the heat source was placed in close proximity to the PCM. This approach could be beneficial for electronic packaging designs that need to minimize the risk of PCM degradation. ALI *et al.* [34] conducted a study to investigate the influence of rectangular, round, and triangular pin fins on the heat storage ratio of a heat sink, while considering a range of energy levels from 5 W to 8 W. Their findings indicated that triangular pin fins were particularly effective because they allowed for more fins to be packed within the same area while simultaneously reducing the surface area ratio. Consequently, this configuration resulted in a higher heat storage ratio. SENTHILKUMAR *et al.* [35] examined three extruded fin shapes combined with PCMs possessing a 28 °C to 44 °C melting range. They improved melting duration and thermal conductivity through PCM volume fraction adjustments. In a related study, they analyzed a rectangular brass fin for free convection cooling in diverse applications. Thermocouples measured temperature and heat transfer rates. They enhanced heat transfer by coating the brass with carbon nanotubes. Using the Taguchi method, they investigated temperature and heat transfer characteristics, finding a 12 % increase in the heat transfer rate for the carbon-nanocoated rectangular brass fins. JI *et al.* [36] proposed a heat sink with varying fin lengths to ascertain the optimal length ratio between adjacent fins for PCM melting. Additionally, non-standard fin shapes, such as branch-shaped fins, were explored by XIE *et al.* [37]. However, it is worth noting that these non-standard fin shapes may entail higher manufacturing costs compared to more conventional solutions.

The work in this article will analyze the effect of fin geometry on the dissipated heat generated from heat sinks using ANSYS-Icepak, a common compu-

tational software for the thermal analysis and design of electronic devices and systems.

SANA *et al.* [38] investigated heat sinks with in-line and staggered fin arrangements using computational fluid dynamics (CFD) software to explore the optimal heat sink design for microelectronic devices. Their study provides an analysis of the thermal performance and pressure drop of two different types of fin cross sections. Two fin configurations, in-line (CIA), and staggered (CSA), are analyzed with 36 fins each. When comparing the performance of circular pin and cone fins at the same power inputs and mass flow rates, the results indicated the maximum temperature for circular pin fins was 0.46 % of the maximum temperature for cone fins, and they had a higher pressure drop, especially in staggered arrangements. Moreover, the maximum temperature for staggered arrangements was lower than that for in-line configurations by a factor of up to 1.17 % and 2.035 % for circular fins and conical pin fins, respectively.

Based on the described findings, several previous studies dealt with the same topics, in particular shape, the number of fins in the heat sink, and their subsequent effects on heat transfer. In the current research, five profiles of fins are studied and compared. Specifically, fin geometries, including longitudinal, square, rectangular, circular, and conical are assessed. These five configurations are considered among the most common fin profiles and are used in electronic devices. The developed design methodology is versatile enough to be applied to different circuit boards, and provides manufacturers with highly applicable, comparative performance data of the different fin assemblies. The results obtained include the cooling performance of each heat sink, maximum temperature reduction, pressure drop, and airflow velocity selected as performance indicators. In the current study, numerical simulation considering cross-sectional variations at a constant fin thickness and base height is conducted to determine, optimal dimensions of the heat sink to achieve the minimum peak operating temperature of the heat sink. Furthermore, the significance of this research lies in its ability to offer clear recommendations for choosing appropriate fin shapes when designing effective heat sinks, which is essential for electronic devices. This understanding becomes particularly crucial for the advancement of cooling systems in compact, high-power electronics, highlighting the substantial influence that fin shapes have on the overall effectiveness of thermal management.

2. MATERIALS AND METHODS

2.1. PHYSICAL MODEL AND PROBLEM DESCRIPTION

In this research, a heat sink with fins of different cross sections is studied. A schematic diagram of the heat sink with different fin cross-sectional

configurations is presented in Fig. 1. All the walls of the heat sink are insulated except for two regions: the inlet, where air enters as the cooling fluid at a specified velocity and temperature and the outlet, which is exposed to atmospheric pressure. The current study is conducted to investigate the influence of fin cross-sectional geometry on overall heat sink performance.

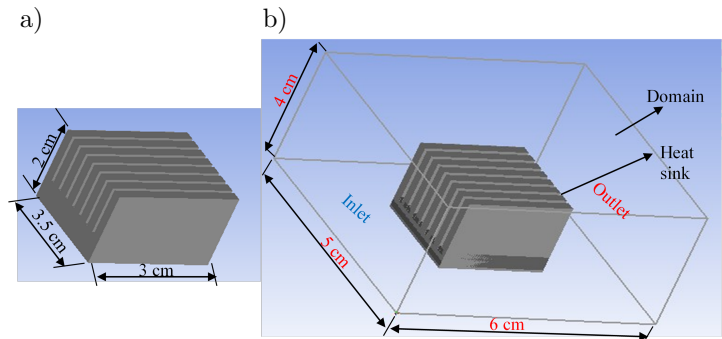


FIG. 1. Schematic diagram of the physical model:
a) heat sink, b) computational domain enclosing the heat sink.

The physical model is an enclosure containing a heat sink mounted on an electronic heat source (chip). The heat sink is made of aluminum and air serves as the cooling fluid. The total dissipated power is set at 8 W, and the heat sink dimensions are given in Table 1. The dimensions of the computational domain are based on the investigation presented in [39].

TABLE 1. Geometric dimensions of the computational domain and the heat sink assembly.

Parameter	Value			
Enclosure	6 cm × 5 cm × 4 cm			
Source	0.3 cm × 0.8 cm × 0.1 cm			
Heat sink	3 cm × 3.5 cm × 2 cm			
Fins	Number of fins = 8			
Base height	0.2 cm			
Cross section	Circular	Square	Rectangular	Conical
	$D = 0.0845$ cm	$L = 0.3$ cm	0.25 cm × 0.36 cm	$R_{\text{large}} = 0.2$ cm $R_{\text{small}} = 0.05$ cm

Five shapes of fins are studied: longitudinal, rectangular, square, circular, and finally conical as shown in Fig. 2. In these five cases, the inlet conditions are constant and the amount of heat flux was applied by the heat source located at the base of the heat sink. The five considered cases vary by their fin cross sections and all the other boundary conditions (BCs) are fixed and do not change.

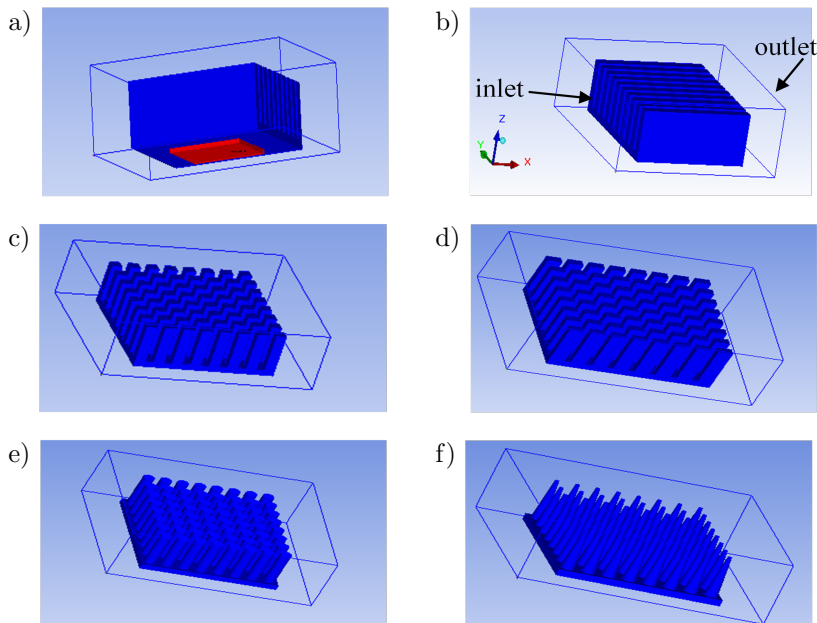


FIG. 2. Schematic of the studied fin geometries: a) whole domain, b) longitudinal fins, c) square fins, d) rectangular fins, e) circular fins, f) conical fins.

ANSYS-Icepak uses the FLUENT solver. In this study, a hexa-dominant mesh was generated in the preprocessing module. The aim of the obtained results is to identify the optimal fin cross section that can be chosen for the fins in order to reduce the temperature of the heat sink and prevent exposing it to high temperatures that exceed what the metal of the heat sink can withstand. In addition, the pressure gradient along the section was evaluated, as it varies depending on the profile change of each fin profile.

2.2. NUMERICAL ANALYSIS ASSUMPTIONS AND GOVERNING EQUATIONS

A numerical analysis is conducted using ANSYS-Icepak software to assess the cooling of the heat sink with air as the cooling fluid. The Icepak software relies on the ANSYS FLUENT solver to compute the thermal and fluid flow fields and to create a diagram, add BCs, and extract graphics and values for all the studied cases. The problem under consideration concerns the flow through the cabinet and heat sink. Heat transfer in the heat sink unit is a conjugate process, which combines conduction heat transfer in the solid parts and convection heat transfer through the fluid. The simulation utilizes the 3D Navier–Stokes and energy equations, in addition to the conservation of mass, to model temperature and flow fields. This involves the application of simplifying assumptions to the momentum and energy equations:

1. 3D, incompressible, laminar, and steady-state fluid flow condition is assumed.
2. A no-slip BC is assumed at all solid walls.
3. Viscous dissipation in the energy equation is negligible.
4. Thermal adiation is negligible.
5. The effect of gravity is negligible.
6. Properties of fluid and solid phases are constant.

After applying this set of assumptions, the 3D equation system, which governs the single-phase model, includes the continuity, momentum, and energy equations, described as [40]:

$$(2.1) \quad \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0.$$

The momentum conservation equations in x -, y -, and z -directions, respectively, are given by [40, 41]:

$$(2.2) \quad u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \frac{\mu}{\rho} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right),$$

$$(2.3) \quad u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial y} + \frac{\mu}{\rho} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right),$$

$$(2.4) \quad u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = -\frac{1}{\rho} \frac{\partial P}{\partial z} + \frac{\mu}{\rho} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right).$$

The energy equation is

$$(2.5) \quad u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{k}{\rho c_p} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right),$$

where the variables T and P represent the fluid temperature and pressure, respectively, whereas u , v , w represent the fluid velocity components, and ρ , μ , c_p , and k denote the fluid density, dynamic viscosity, specific heat capacity, and thermal conductivity, respectively.

The heat conduction equation for the solid heat sink walls, is given by [9]:

$$(2.6) \quad \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} = 0,$$

where subscript s refers to the solid domain.

2.2.1. AVERAGE NUSSELT NUMBER

The average Nusselt number (Nu_{avg}) is a dimensionless parameter that characterizes the efficiency of convective heat transfer relative to conductive heat

transfer across a boundary layer. Higher Nusselt numbers indicate more effective convective heat transfer.

The local Nusselt number indicates how the local rate of heat transfer varies along the hot surface and can be defined as [42]:

$$(2.7) \quad \text{Nu}_{\text{avg}} = \frac{hl}{k_f},$$

where h is the average heat transfer coefficient, l is the characteristic length of the sink, and k_f is the thermal conductivity of the fluid.

The average forced convection heat transfer coefficient (h) of the air inside the heat sink is determined using Newton's law of cooling, where h is determined by the convective heat transfer rate (Q), the heat sink surface area (A), and the temperature difference (ΔT) between the surface and the air.

The following three steps are used to determine the heat transfer coefficient (h).

Step 1: Determine the temperature difference (ΔT).

Since the air heats up as it passes through the heat sink:

$$(2.8) \quad \Delta T = T_s - T_{\text{in}},$$

where T_s is the surface temperature of the heat sink, and T_{in} is the fluid inlet temperature.

Step 2: Measure the heat transfer rate (Q).

This can be calculated using the air mass flow rate (m) and the specific heat capacity of the air (c_p) as follows:

$$(2.9) \quad Q = m c_p (T_{\text{in}} - T_{\text{out}}),$$

where T_{out} is the fluid outlet temperature.

Step 3: Calculate the heat transfer coefficient (h), from Newton's law of cooling as follows:

$$(2.10) \quad h = \frac{Q}{A \Delta T},$$

where A is the surface area of the heat sink exposed to the airflow.

Note: The forced convection heat transfer coefficient in air-cooled heat sinks typically ranges from 20 W/m²K to 250 W/m²K.

The BCs used for this problem are stated as follows. Air is uniformly drawn, with constant temperature and velocity, into the cabinet inlet. At the downstream boundary of the computational domain, which is located at the cabinet outlet along the x -direction, a pressure BC is applied. No-slip conditions with a thermal insulation are enforced on all the other confined walls. At the base of the heat sink, a uniform constant heat flux is applied from the source heat. Adiabatic thermal BC are applied to the outer perimeter of the heat sink base, except for the heated area:

1. BCs for the flow field: a uniform velocity is prescribed at the channel inlet. A no-slip condition is used for all the walls, i.e., $u = 0, v = 0, w = 0$.
2. Adiabatic BCs are applied to all the boundaries of the solid region except for the heat sink bottom wall, where a heat source with a constant power of 8 W is applied. At the inlet, the temperature is set to $T = T_{\text{in}} = 298 \text{ K}$, $v_{\text{in}} = 0.1 \text{ m/s}$, $-k_s \frac{\partial T_s}{\partial y} = -k_f \frac{\partial T_f}{\partial y}$ at the fluid–solid interface, $\frac{\partial T}{\partial y} = 0$ at the outlet.

2.3. VERIFICATION OF GRID INDEPENDENCE

The effect of grid refinement on the numerical solutions is investigated for the cases of 8 solid circular fins in an in-line arrangement. Grid independence is investigated by testing five different mesh sizes, as shown in Table 2. The grid independence test is first conducted by using several different numbers of elements and nodes. The results obtained from these meshes are summarized in Table 2.

TABLE 2. Mesh configurations and corresponding maximum source temperatures.

Test no.	Number of elements	Number of nodes	Temperature of the source [°C]
1.	40 560	45 738	138.688
2.	82 784	91 800	136.888
3.	90 902	100 320	136.112
4.	111 585	122 496	135.998
5.	136 912	249 600	135.948

Table 2 reveals that increasing the number of nodes beyond 122 496 results in a negligible temperature variation of less than 0.04% for the heat source. In the case of solid fins, a significantly lower number of nodes, specifically 122 496 nodes, demonstrates grid independence. Beyond this point, further grid refinement does not appreciably impact the solution. As a result, the fourth mesh size is selected for calculations across all fin geometries to maintain maximum accuracy. Figure 3 shows the 3D computational quadratic mesh of the domain.

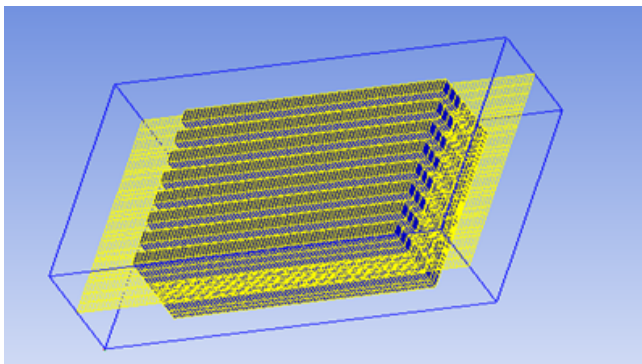


FIG. 3. 3D computational quadratic mesh of the domain.

3. SIMULATION RESULTS

3.1. VALIDATION OF THE NUMERICAL MODEL

In computational thermal analysis, ensuring the accuracy of numerical models is essential, and this accuracy can be established through rigorous validation against well-established benchmarks. This subsection aims to validate the precision of the current computational framework by directly comparing it with the numerical results obtained by ROSLI *et al.* [39]. To achieve this, a model was employed, featuring a heat sink with vertical fins. This model meticulously adhered to the dimensions and BCs outlined in [39], including a pin fin height of 0.2 cm, a length of 4 cm, and a width of $57\text{ }\mu\text{m}$. Additionally, the model considered an inlet temperature of $25\text{ }^{\circ}\text{C}$, an inlet velocity of 0.1 m/s , and a constant thermal load of 5 W applied to the heat source located at the base of the heat sink.

Figure 4 illustrates a top-view depiction of the temperature distribution within the pin-fin heat sink featuring in-line circular fins, akin to the configuration examined by ROSLI *et al.* [39], denoted as Model CIA 3. In the present

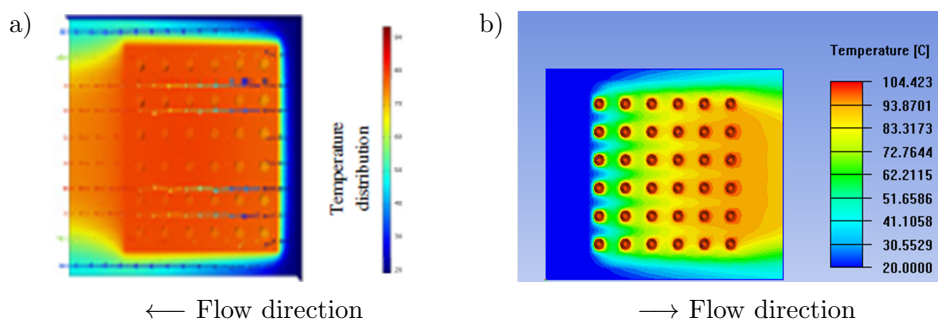


FIG. 4. Top-view distribution plots of temperature for the circular pin-fin heat sink model with an in-line arrangement. A comparison between the result presented by (a) ROSLI *et al.* [39], and (b) current simulation result.

model, the maximum temperature reached 104.423 °C, closely mirroring the reference value of 107.43 °C. This slight temperature variance of 2.8% falls well within the widely acknowledged error tolerance range for thermal simulations within this field [36], affirming the high accuracy of the simulation results. This slight variance can typically be attributed to minor differences in mesh discretization, numerical methods, or the precise implementation of boundary conditions between the two models. A summary of the temperature distribution between the current model and the corresponding data from ROSLI *et al.* [39], which is performed for pin fins with CIA arrangements, is presented in Table 3.

TABLE 3. Comparing the temperature distribution between our current results and the corresponding ones by ROSLI *et al.* [39] which is performed for pin fins with CIA arrangements.

Temperature in °C [starting from maximum to minimum]	Current simulation results	Numerical results by ROSLI <i>et al.</i> [39]
Maximum temperature	104.423	107.43
Intermediate temperatures	83.32	84
	72.76	70
	62.21	60
	30.55	30
Minimum temperature	20	20

The close proximity between the outcomes of the present simulation and those of the reference study validates the present numerical approach, affirming its capability to accurately predict the thermal behavior of heat sink configurations. While this validation establishes the accuracy of the model for the specific scenario examined, it is acknowledged that further validation tests are necessary to enhance the model’s broader applicability. Furthermore, further validation efforts should encompass different fin geometries, flow rates, and thermal loads to expand the model’s suitability for a wider spectrum of thermal management scenarios. In conclusion, this validation study underscores the reliability of the employed computational approach and reinforces the subsequent findings of the study.

Figure 5 displays the maximum temperature of the heat sink at different inflow velocities, serving to further verify the precision of our numerical model. We juxtaposed this data with the findings of ADHIKARI *et al.* [43]. Their study utilized a combination of experimental and numerical techniques to examine heat transfer from rectangular fins under forced convection conditions, specifically at low Reynolds numbers. This comparison is an essential step in showcasing the reliability of our computational technique. Fortunately, the data demonstrate outstanding concurrence. The highest discrepancy between our computational

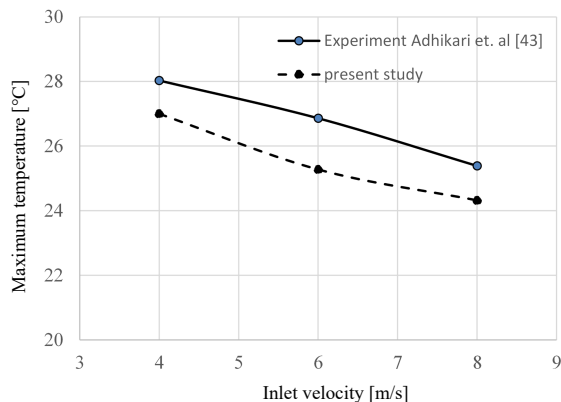


FIG. 5. Maximum temperature of the heat sink at different inlet velocities.

forecasts and the data obtained from ADHIKARI *et al.* [43] is below 3.68 %. This strong correlation confirms that our selected numerical approach effectively simulates forced convection heat transfer in rectangular fin geometries.

3.2. TEMPERATURE DISTRIBUTION RESULTS

3.2.1. TEMPERATURE DISTRIBUTION WITHIN THE CABINET

In thermal management systems, especially in electronic enclosures, it is crucial to consider the significant consequences of both high and low temperatures. Higher temperatures can result in reduced efficiency and the risk of electronic component failure, whereas lower temperatures typically indicate effective heat dissipation [1]. Therefore, analyzing the temperature distribution within the cabinet provides essential information about the effectiveness of the cooling system in use.

In Fig. 6, the temperature distribution within the computational domain is illustrated, showing a clear gradient that stems from the cooler inlet region toward the warmer outlet region. This gradient indicates the progressive heating of the cooling air as it absorbs heat from the cabinet's components. The temperature increase aligns with the direction of airflow, supporting the understanding of convective heat transfer mechanics, where air, entering at a lower temperature, absorbs heat from the solid electronic components and the heat sink.

The cooler temperatures at the cabinet inlet suggest that the incoming air effectively absorbs heat, a positive indicator of the initial thermal management strategy. As the air moves through the cabinet, the increasing temperature gradient toward the exit signifies that the air becomes less efficient at removing heat due to its own temperature rise. Figure 6 visually represents this temperature change, showing a transition from cooler blue regions to warmer red regions.

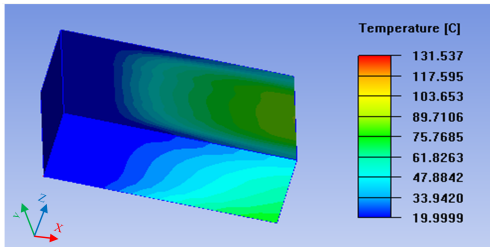


FIG. 6. Temperature distribution within the electronic enclosure domain.

This distribution pattern is critical as it reflects the thermal load within the cabinet and assesses the cabinet’s design effectiveness in facilitating airflow and heat transfer. The elevated temperatures near the outlet raise concerns about potential hotspots, which are localized regions of concentrated heat that could surpass the operational temperature limits of the electronic components housed within the cabinet.

Furthermore, these high-temperature regions imply the possibility of optimizing the cabinet’s cooling design. Potential enhancements might include improving airflow through strategic placement of vents or fans, optimizing the layout of internal components to achieve more uniform heat distribution, or incorporating heat sinks with more efficient fin designs for enhanced heat dissipation.

The observed temperature distribution offers valuable insights into the thermal behavior within the cabinet, underscoring the significance of considering both the design of the thermal management system and the layout of electronic components to ensure the sustained effectiveness of cooling air throughout the entire cabinet.

3.2.2. TEMPERATURE DISTRIBUTION OF DIFFERENT FIN GEOMETRIES

The thermal performance of heat sinks is significantly influenced by the temperature distribution within different fin designs. Elevated temperatures observed on the fins may signal suboptimal heat dissipation, raising the risk of component overheating. Conversely, lower temperatures typically signify efficient heat removal. Consequently, analyzing the temperature profiles of diverse fin configurations offers valuable insights into their cooling effectiveness. Figure 7 to Fig. 11 illustrate a comparative examination of temperature distributions across various fin geometries (longitudinal, square, rectangular, circular, and conical) implemented on heat sinks, indicating the nuanced impact of fin design on thermal performance.

Starting with the examination of longitudinal fins, as depicted in Fig. 7, the temperature distribution pattern shows that the temperature distribution

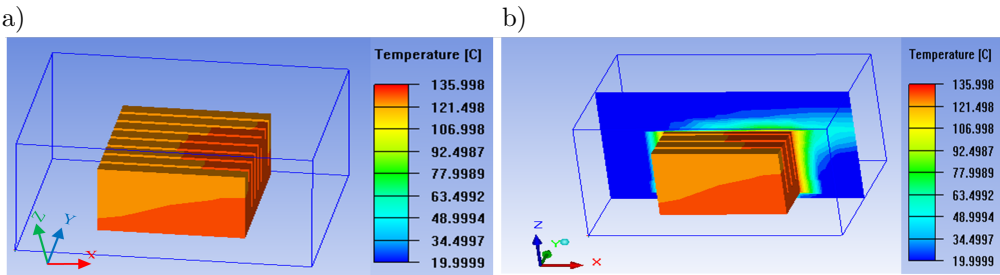


FIG. 7. Temperature distribution of longitudinal fins on: a) heat sink, b) Y-cut plane.

along the fins is continuous and has a large value along the heat sink. The entire heat sink has warmer extremes, peaking at around 135.99°C . This consistent thermal gradient along the length of the fins implies an effective heat conduction pathway from the heat source to the fin tips. It is observed that the temperature distribution along the fins ranges between approximately 121°C and 136°C . The figure also shows that the temperature is highest at the base of the fins because it is close to the heat source located below the heat sink, and the temperature decreases from the fin base to the fin tip. Furthermore, Fig. 7b illustrates that along a single fin, the temperature decreases as the end of the sink is approached. This is because the air absorbs heat through convection from the fins, causing the air temperature to rise and the fin temperature to decrease from the inlet to the outlet.

In contrast, Fig. 8 presents the temperature profile for square fins, where an evident central heating zone is discernible, characterized by a sharper gradient reaching its maximum at approximately 131.54°C . This indicates a concentration of heat directly above the heat source. The square fin design, while effective, may benefit from optimization strategies, such as alterations to fin spacing or increased airflow, to address the relatively less efficient heat transfer in this particular region. Figure 9 illustrates the temperature distribution for rectangular fins, with the highest temperature recorded at around 124.55°C . The temperature gradient observed across the fins is notably uniform, suggesting efficient heat distribution. The design of rectangular fins seems to maximize

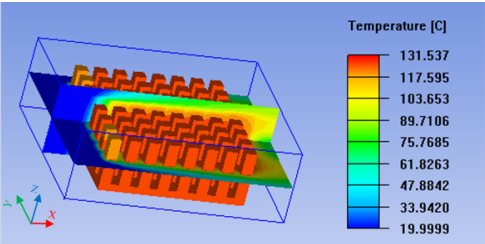


FIG. 8. Temperature distribution of square fins on the heat sink and through y- and z-planes.

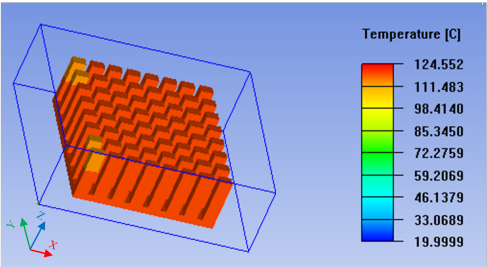


FIG. 9. Temperature distribution of rectangular fins on the heat sink.

the surface area in contact with the cooling air, thereby promoting effective convective heat transfer. This is reflected in the relatively cooler temperatures observed across the entire heat sink. Upon examining circular fins in Fig. 10, a higher maximum temperature is evident, particularly noticeable in the z -plane, where it reaches up to 144.78°C . The circular design might be impeding airflow, leading to a less effective convective cooling process in comparison to the more streamlined rectangular fins. The thermal pattern suggests that while circular fins provide an efficient conductive pathway, the convective cooling may not be as effective, likely due to airflow disruption caused by the fins' shape. Lastly, Fig. 11 presents conical fins, exhibiting the highest maximum tempera-

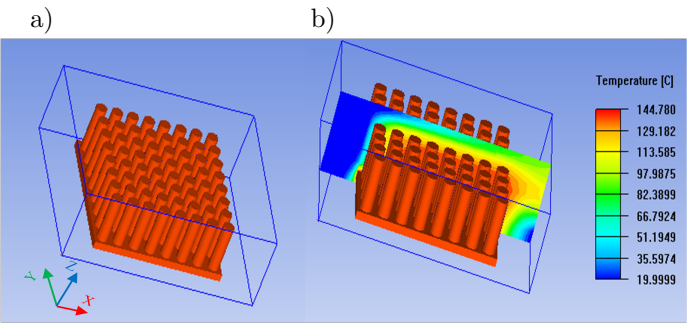


FIG. 10. Temperature distribution through: a) the circular fins on the heat sink, b) the z -plane.

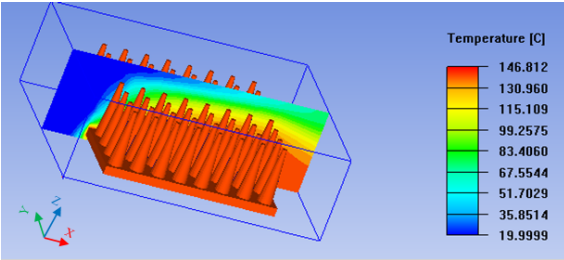


FIG. 11. Temperature distribution of conical fins on the heat sink and through the z -plane.

ture among the various geometries, ascending to 146.81 °C. The conical shape, though visually distinctive, appears to yield diminishing returns in terms of cooling performance. As the fins taper towards their tips, the available surface area for heat dissipation decreases, which likely contributes to the reduced effectiveness of convective heat transfer.

The temperature distribution of square fins in Fig. 8 reveals a distinctive thermal pattern when analyzed from two perpendicular planes along the y - and z -axes. This dual-plane analysis offers a more intricate understanding of the heat sink's thermal behavior, highlighting the anisotropic nature of heat transfer in different directions. In the y -direction, which likely corresponds to the direction of airflow, there is a gradual increase in temperature from the cooler inlet, depicted by blue-green colors at approximately 33.94 °C, to the warmer outlet regions represented by yellow and red hues, peaking at around 131.54 °C. This temperature gradient is significant as it illustrates how air absorbs heat while passing over the fins. The square fins' geometry, despite providing a substantial surface area for heat transfer, may induce airflow patterns resulting in uneven cooling. Specifically, the central region directly above the heat source exhibits a noticeable hotspot, indicating that the air passing through this region becomes saturated with heat, becoming less effective at cooling the downstream fins. This phenomenon suggests that there may be a limit to the cooling capacity of square fins due to airflow dynamics, which could potentially be optimized by adjusting fin spacing or height to enhance cross-flow and improve convective heat dissipation. In the z -direction, the temperature distribution along the height of the fins is observed, with lower sections of the fins closer to the heat source having higher temperatures, as expected due to conductive heat transfer. However, the temperature profile indicates a rapid decrease in temperature when moving away from the base, transitioning from red to green on the color scale. This decline suggests that while the fin bases effectively conduct heat away from the source, the tips of the fins may not be as efficient in dissipating this heat into the surrounding air. The heat distribution in the z -direction is particularly significant as it impacts the overall thermal resistance of the heat sink. The observed temperature variation along the height of the fins may imply the need for a design that encourages better airflow around the fin tips or the use of materials with higher thermal conductivity to ensure more uniform heat dissipation along the entire fin length.

The temperature stratification observed in both the y - and z -directions highlights the intricate interaction between heat conduction along the fins and convective heat transfer to the surrounding air. An effective heat sink design must take into consideration these directional variations to prevent performance limitations. The implications derived from Fig. 8 can provide valuable insights for design modifications, including adjustments to fin aspect ratios, alterations in

material selection, or improvements to the thermal interface between the fin base and the heat source, all aimed at enhancing the thermal management capabilities of square fin heat sinks.

In conclusion, this comparative analysis underscores the critical role of fin geometry in the field of thermal management. Among the examined designs, rectangular fins emerge as the most efficient, striking a harmonious balance between conductive and convective heat transfer. In contrast, circular and conical fins, which exhibit higher temperatures, may necessitate design modifications or supplementary cooling strategies to achieve comparable levels of thermal performance. The rectangular fins provided a reduction of the maximum temperature by approximately 8.42% compared to longitudinal fins, 5.29% compared to square fins, 13.97% compared to circular fins, and 15.16% compared to conical fins, respectively. These insights bear significant relevance in the context of designing cooling systems for microelectronic devices, where the effective management of peak temperatures is a fundamental requirement for ensuring reliability and optimal performance.

3.3. PRESSURE DISTRIBUTION RESULTS

The pressure distribution patterns across various fin geometries, as illustrated in Fig. 12, provide valuable insights into the aerodynamic efficiency and its consequential impact on thermal performance. Elevated pressure levels detected on the inlet side of the heat sink fins suggest resistance to incoming air. This resistance may be attributed to either insufficient spacing between the fins or suboptimal fin shapes, resulting in increased airflow hindrance. While this situation may imply enhanced forced airflow through the heat sink, potentially leading to improved heat transfer rates, it could also necessitate higher energy consumption by fans or pumps handling air or coolant circulation within the system. Furthermore, excessive pressure could induce airflow bypass, wherein air circumvents the fin array rather than passing through it, thereby diminishing the heat sink's effectiveness [31]. Conversely, lower pressure values signify smoother airflow through the fin array with reduced resistance. This is generally desirable as it indicates that less energy is required to move air through the system, leading to increased system cooling efficiency. However, excessively low pressure could imply insufficient contact time between the air and the fins for effective heat transfer, particularly if the airflow velocity is too high [32]. Striking the right balance is crucial; airflow should be adequate to facilitate efficient heat removal but not so swift that it fails to absorb adequate heat from the fins.

The pressure distribution data for square fins in Fig. 12 a show a peak pressure of 0.176982 N/m^2 near the inlet, gradually decreasing toward the outlet. This pattern suggests that square fins induce a higher pressure drop, possibly

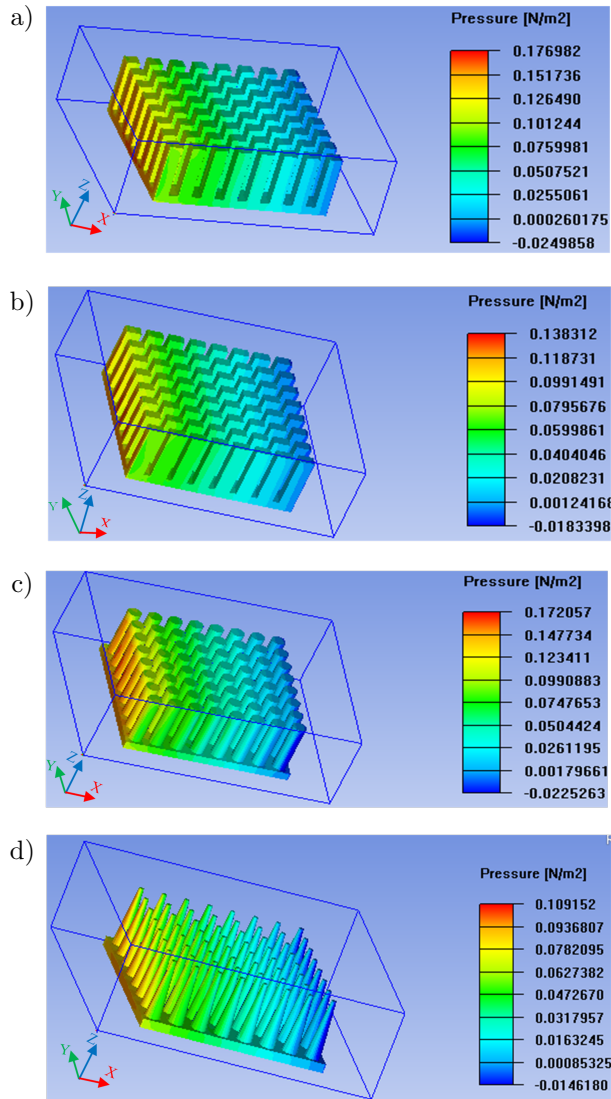


FIG. 12. Pressure distribution over: a) square fins, b) rectangular fins, c) circular fins, d) conical fins.

due to airflow disruption caused by the sharp edges and corners of these fins, resulting in separated flows and vortices that elevate overall aerodynamic drag. In Fig. 12b, the pressure drop over the rectangular fins is slightly lower, with a maximum of 0.138312 N/m², indicating relatively smoother airflow compared to square fins. The elongated shape of rectangular fins likely guides airflow more smoothly between the fins, minimizing flow separation and the associated pressure drop.

This suggests a better-balanced design in terms of thermal and aerodynamic performance. Circular fins in Fig. 12c exhibit a higher-pressure region with a maximum of 0.172057 N/m^2 at the inlet, similar to square fins, but the pressure gradually decreases toward the outlet. The rounded shape of circular fins likely contributes to maintaining a more laminar flow, reducing the occurrence of flow separation and consequently the pressure drop. Interestingly, conical fins in Fig. 12d show the lowest maximum pressure at 0.109152 N/m^2 . This can be attributed to the tapered design, which facilitates a streamlined airflow with reduced resistance. This shape aids in efficient redirection of the flow, minimizing aerodynamic drag and resulting in a lower pressure drop across the fins.

When comparing these fin profiles, square fins exhibit the highest pressure drop, indicating lower aerodynamic efficiency. Rectangular and circular fins offer improvements in airflow management, with circular fins particularly enhancing flow smoothness. Conical fins stand out with the lowest pressure drop, indicating superior airflow resistance, but this may not necessarily translate to improved heat dissipation, as observed in the temperature distribution analysis. Relating these pressure findings to earlier temperature distribution observations reveals a complex trade-off between achieving low thermal resistance and low aerodynamic drag. Conical fins exhibit a 38.07 % reduction in pressure drop compared to square fins, a 21.01 % reduction compared to rectangular fins, and a 36.63 % reduction compared to circular fin geometries, indicating a more streamlined airflow through the fin array and reduced aerodynamic resistance. However, despite the reduced pressure drop, conical fins also exhibit the highest temperatures, suggesting that while airflow through the fins encounters less resistance, it may not be the most effective for heat transfer. In contrast, square and circular fins, despite causing higher pressure drops, do not necessarily result in higher temperatures compared to conical fins, indicating a complex interplay between airflow patterns and thermal efficiency. Rectangular fins, with their balanced pressure drop and lower temperatures, appear to offer an optimal geometry for effective heat dissipation in this study.

This analysis underscores the importance of considering both aerodynamic and thermal aspects when designing heat sinks. The goal is to identify a configuration that minimizes both thermal resistance for efficient heat transfer and aerodynamic drag for lower pressure drops, ultimately enhancing the overall performance of the cooling system.

3.4. VELOCITY DISTRIBUTION RESULTS

A comprehensive examination of airflow patterns across various fin configurations plays a pivotal role in gaining insights into heat sink behavior, a critical determinant of convective heat transfer efficiency. Within the array of fins in a heat

sink, elevated velocity values suggest the rapid movement of air through the inter-fin channels. This accelerated airflow can be advantageous for heat transfer, as it promotes convective cooling by directing more air over the fin surfaces and reducing the thickness of the thermal boundary layer [44]. Consequently, this promotes the efficient extraction of heat from the heat sink. However, excessively high velocities may compromise heat transfer efficiency by diminishing the time air spends in contact with the fins, limiting heat absorption. Furthermore, heightened velocities can elevate noise levels in the cooling system and potentially lead to increased wear on mechanical components such as fans, owing to greater forces involved [45]. Conversely, lower velocity values indicate a moderate airflow, implying less efficient heat transfer due to a thicker thermal boundary layer, which allows more heat to accumulate within the heat sink. While this reduced airflow speed may result in quieter operation and less mechanical stress, it may not suffice for effective cooling, particularly under high thermal loads [46]. It is imperative to strike a balance and ensure that the velocity does not dip too low to maintain adequate cooling performance, especially in compact or high-powered electronic devices where thermal management holds paramount importance. In essence, the optimization of the velocity distribution is a meticulous task, necessitating careful design considerations to align with the specific cooling requirements of the electronic device or system under investigation.

Figure 13 reveals that the velocity distribution over different shapes of fins. Figure 13a shows that the square fins peak velocity at approximately 0.327 m/s, indicating a substantial airflow through the fin channels, which can augment convective heat transfer. However, this high-velocity airflow may also lead to an increased pressure drop and turbulence, particularly in the vicinity of the square fin edges, as elucidated in the earlier discussion on pressure distribution. Figure 13b depicts slightly lower peak velocities of around 0.263 m/s for rectangular fins, suggesting a more moderate airflow. This velocity profile could reduce the power requirements of fans and potentially yield quieter operation while still delivering sufficient airflow for cooling and minimizing energy consumption for air movement. In Fig. 13c, circular fins exhibit the highest peak velocity at approximately 0.340 m/s, indicating that their rounded design may streamline airflow, reducing resistance. Nevertheless, as observed in the temperature analysis, this does not invariably translate into the most effective cooling, possibly due to the circular fins' suboptimal surface area for heat transfer. Lastly, Fig. 13d presents conical fins with a peak velocity of roughly 0.272 m/s. The tapered configuration of these fins may aid in guiding airflow smoothly through the heat sink, contributing to the lowest pressure drop among the various fin geometries. Nonetheless, and as previously noted, this does not align with the best thermal performance, as conical fins exhibit the highest temperatures compared to other fin designs.

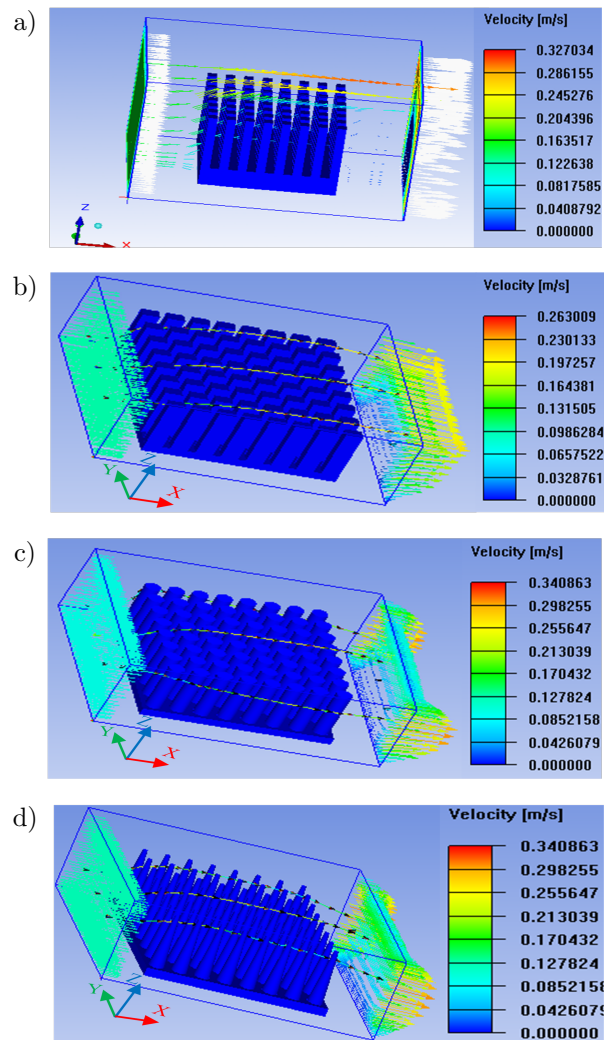


FIG. 13. Velocity distribution over the different types of fins circular fins: a) square fins, b) rectangular fins, c) circular fins, d) conical fins.

Circular fins demonstrate approximately a 29.28 % increase in peak airflow velocity over rectangular fins, a 3.98 % increase over square fins, and a 25.0 % increase over conical fins. This indicates that their rounded shape may facilitate a more streamlined airflow path, contributing to the acceleration of air through the fin array. Rectangular fins exhibit slightly reduced velocity, signifying potentially smaller airflow turbulence, while conical fins display a moderate velocity profile. Nevertheless, the high velocities observed with circular and square fins do not necessarily translate into lower temperatures, underscoring that merely augmenting airflow velocity does not invariably enhance heat sink performance.

Instead, heat transfer efficiency also hinges on the available surface area and the behavior of the thermal boundary layer.

The overarching objective is to optimize heat dissipation while concurrently minimizing both aerodynamic drag and the power needed to sustain airflow, which is a pivotal consideration for the overall performance and energy efficiency of cooling systems in electronic devices. Combining data on velocity, pressure, and temperature furnishes a holistic perspective on cooling performance. Although conical fins offer the lowest pressure drop and moderate velocities, they lead to the highest temperatures, suggesting that their shape may not be the most suitable for thermal management, despite favorable aerodynamic attributes. In contrast, rectangular fins strike an optimal equilibrium, with moderate velocities corresponding to both lower temperatures and pressure drops, signifying their superior suitability for cooling applications. These findings underscore the necessity of an integrated approach that takes into account all three parameters in designing efficient heat sinks. The ultimate aim is to maximize heat dissipation while simultaneously minimizing both aerodynamic resistance and power consumption, which are critical for the overall performance and energy efficiency of cooling systems in electronic devices.

3.5. AVERAGE NUSSELT NUMBER

The Nusselt number (Nu_{avg}) is calculated for different types of fins, and the results are listed in [Table 4](#).

TABLE 4. Average Nusselt number for each type of fins.

Type of fin	Average Nusselt number
Rectangular	10.0380
Longitudinal	9.1690
Square	8.7390
Circular	7.8870
Conical	6.1340

Given the average Nusselt numbers for the different types of fins presented in [Table 4](#), their thermal performance can be interpreted as follows:

1. Rectangular fins ($Nu_{avg} = 10.0380$) exhibit the highest Nusselt number, indicating superior convective heat transfer efficiency. This suggests that the rectangular fins have the most effective surface area and geometry for promoting both conductive and convective heat transfer. The large surface area of rectangular fins facilitates better heat dissipation from the fin surface to the surrounding air, maximizing thermal performance.
2. Longitudinal fins ($Nu_{avg} = 9.1690$) have a slightly lower Nusselt number than rectangular fins. They still provide efficient heat transfer, but

their geometry may not promote as much airflow interaction or provide as much surface area as rectangular fins. Nevertheless, they are still quite effective in enhancing heat transfer compared to simpler or less optimized geometries.

3. Square fins ($Nu_{avg} = 8.7390$) demonstrate a good level of heat transfer, but smaller than that of rectangular and longitudinal fins. The edges and geometry of square fins might cause some flow separation or less optimal airflow patterns compared to the more streamlined shapes, reducing their overall efficiency.
4. Circular fins ($Nu_{avg} = 7.8870$), while promoting high airflow velocity, have a lower Nusselt number, indicating less effective convective heat transfer. The round shape may lead to a reduced surface contact area for heat transfer compared to flat or elongated shapes, which can diminish the overall heat dissipation capability.
5. Conical fins ($Nu_{avg} = 6.1340$) have the lowest Nusselt number, signifying the least effective convective heat transfer. The tapered shape reduces the surface area available for heat transfer as the air moves along the fin, limiting their thermal performance. Additionally, the pointed tips may not contribute significantly to heat dissipation, further decreasing their efficiency.

3.5.1. OVERALL INTERPRETATION

Rectangular fins are the most efficient for heat transfer, making them ideal for applications requiring maximum cooling efficiency. Longitudinal and square fins also perform well, making them suitable for situations where slightly lower performance can be tolerated or where geometric constraints favor their use. Circular fins have lower efficiency but might be preferred in scenarios where airflow velocity needs to be maximized or where design constraints require a streamlined shape. Conical fins are the least efficient in terms of heat transfer, and their use would therefore be more specialized or limited to applications where other factors (such as minimal pressure drop) are more critical than thermal performance.

In conclusion, the physical interpretation of the Nusselt number for each fin type provides insight into their relative effectiveness for heat dissipation in electronic cooling applications, guiding the design and selection of heat sinks based on specific thermal management needs.

4. COMPARISONS AND DISCUSSION

This section presents a comparison of the various fin geometries assessed in this study based on the results of the analyses, including longitudinal, square, rectangular, circular, and conical fin shapes, with a specific emphasis on their performance across three key performance metrics: maximum temperature re-

duction, pressure drop, and airflow velocity. Furthermore, [Table 5](#) offers a concise summary of this comparative analysis based on the results of the analyses.

TABLE 5. Comparative ranking of fin geometries.

Performance indicator	1st (best)	2nd	3rd	4th	5th (worst)
Maximum temperature reduction (higher reduction is better)	Rectangular	Square	Longitudinal	Circular	Conical
Pressure drop (lower is better)	Conical	Rectangular	Circular	Square	Longitudinal
Peak airflow velocity (descending magnitude; not an overall-performance ranking)	Circular	Square	Conical	Rectangular	Longitudinal
Overall cooling performance (thermal performance with hydraulic consideration)	Rectangular	Square	Circular/ conical/ longitudinal	–	–

4.1. MAXIMUM TEMPERATURE REDUCTION

It can be noticed that the rectangular fins demonstrated the highest efficacy in reducing heat sink temperatures, achieving reductions of 8.4 % compared to longitudinal fins, 5.3 % compared to square fins, 14.0 % compared to circular fins, and 15.1 % compared to conical fins. This high performance can be attributed to their larger surface area, which facilitates both conductive and convective heat transfer. At the same time, longitudinal fins exhibited a moderate level of performance in terms of temperature reduction. Square fins offered temperature control that was close to that of longitudinal fins, but they were slightly less efficient than rectangular fins. Although circular fins exhibited high airflow velocity, they were less effective in reducing temperature, possibly due to suboptimal surface area contact with the surrounding air. Conical fins displayed the lowest effectiveness in temperature control, likely because of their reduced surface area at the tips, which hindered efficient heat dissipation.

4.2. PRESSURE DROP

It can be noticed that the conical fins stood out with a significant reduction in pressure drop, indicating more streamlined airflow. They achieved reductions

of 38.1 % compared to square fins, 21.0 % compared to rectangular fins, and 36.6 % compared to circular fins. Rectangular fins offered a well-balanced pressure drop, neither too high to cause excessive energy consumption for airflow nor too low to compromise effective heat transfer. Both square and circular fins experienced higher pressure drops, suggesting potential airflow resistance due to their respective shapes.

4.3. AIRFLOW VELOCITY

It can be noticed that the circular fins demonstrated the highest increase in airflow velocity, implying that their shape promotes streamlined airflow. They showcased a 29.3 %, 4.0 %, and 25.0 % increase in peak airflow velocity compared to the rectangular, square, and conical fin geometries, respectively. Conical fins exhibited a moderate airflow velocity, aligning with their reduced pressure drop and aerodynamic design. Rectangular fins displayed a balanced airflow velocity, conducive to effective cooling without excessive energy consumption. Square fins showed slightly lower airflow velocities than circular fins, possibly due to airflow disturbances caused by their sharp edges.

4.4. DISCUSSION

This comprehensive analysis yields crucial insights into how the geometry of fins influences the thermal management of heat sinks. It emphasizes the necessity of considering multiple factors, including surface area, airflow dynamics, and pressure characteristics, when selecting a fin geometry for heat sinks, particularly in applications demanding efficient cooling in compact, high-powered electronic devices.

The findings from this study can serve as a valuable guide for designing and choosing heat sinks in various electronic cooling applications, striking a balance between effective temperature control and considerations of aerodynamic efficiency and energy efficiency.

Given these performance metrics, rectangular fins stand out as the optimal choice for such applications. Their ability to reduce maximum temperatures surpasses that of the other fin shapes, demonstrating their effectiveness in thermal management. Moreover, their balanced airflow velocity and moderate pressure drop indicate their efficiency in maintaining aerodynamic stability while minimizing energy consumption. This makes rectangular fins particularly well-suited for high-performance electronic cooling applications, where optimizing heat dissipation and energy efficiency are crucial. Their versatility and effectiveness under various operating conditions reaffirm their status as a preferred choice for advanced electronic cooling solutions.

5. CONCLUSION AND FUTURE WORK

In this study, the effect of fin geometry on heat sink performance was investigated. The evaluation of the cooling effectiveness of each heat sink relied on comprehensive performance indicators, including maximum temperature reduction, pressure drop, and airflow velocity distribution across the fins. These metrics collectively contributed to the assessment of each design's thermal performance. The main outcomes of this investigation are as follows:

1. Fin geometry is a key determinant of the heat sink temperature, pressure drop, and airflow velocity, crucially affecting its thermal and aerodynamic efficiency.
2. Rectangular fins outperform longitudinal, square, circular, and conical geometries in reducing the maximum temperature by 8.4 %, 5.3 %, 14.0 %, and 15.1 % respectively. This suggests a more efficient heat dissipation capability, striking the best balance between conductive and convective heat transfer.
3. Conical fins exhibit a 38.1 %, 21.0 %, and 36.6 % reduction in pressure drop compared to square, rectangular, and circular fin geometries, respectively, indicating a more streamlined airflow through the fin array and reduced aerodynamic resistance.
4. Circular fins demonstrate a 29.3 %, 4.0 %, and 25.0 % increase in peak airflow velocity over rectangular, square, and conical fin geometries, respectively, implying that their shape may promote a more streamlined airflow path, accelerating air passage through the fin array.
5. The comprehensive analysis of temperature, pressure, and airflow velocity data reveals that rectangular fins offer the best balance, achieving optimal heat dissipation with efficient aerodynamics and lower energy requirements, thus emerging as the most effective design for heat sink applications.
6. Balancing thermal resistance to enhance heat transfer efficiency while reducing aerodynamic drag to minimize pressure drops and increase airflow velocity is crucial for the optimization of cooling system performance.

In future work, it is crucial to investigate the effects of varying environmental conditions, including changes in ambient temperature and airflow patterns, to gain deeper insight into heat sink performance. The utilization of advanced computational models capable of simulating dynamic thermal loads and real-world operational scenarios is pivotal for further heat sink design refinement. Additionally, experimenting with innovative fin geometries, potentially drawing inspiration from biomimicry or novel engineering concepts, could unlock new possibilities for managing thermal concerns in compact, high-powered electronic devices. It is also recommended to study hybrid or irregular geometric shapes

of fins, or combinations of two or more types of fins to investigate/evaluate the effect of these combinations on the performance of heat sinks.

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CONFLICT OF INTERESTS

The authors declare that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHORS' CONTRIBUTIONS

All authors contributed equally to the paper, reviewed and approved the final manuscript.

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Research Paper

Multi-Objective Collaborative Optimization of Central Support Components in Steel Structures Integrating NSGA-II and Nested Topology Optimization

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This paper presents a multi-objective collaborative optimization framework for central support member in steel structures, aiming to simultaneously minimize structural mass and compliance. Traditional design methods often optimize a single objective, therefore limiting overall performance. To address this, we integrate the NSGA-II genetic algorithm with the solid isotropic material with penalization (SIMP) topology optimization method within hierarchical bi-level optimization framework. NSGA-II performs a global search for optimal macro-level geometric parameters, while SIMP optimizes the micro-level material distribution under the given geometries. The interaction between both levels enables a comprehensive trade-off between lightweight design and structural stiffness. Case studies on various support types demonstrate that the proposed method effectively reduces structural mass while enhancing stiffness, confirming its robustness and broad applicability. The Pareto front achieves a high hypervolume (HV) index, indicating excellent solution diversity and quality. This approach provides an intelligent and efficient pathway for the high-performance, lightweight design of steel structures.

Keywords: steel structures, central support member, multi-objective optimization, topology optimization, NSGA-II algorithm, collaborative optimization.



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1. INTRODUCTION

1.1. RESEARCH BACKGROUND AND SIGNIFICANCE

Steel structures are widely utilized in modern construction, including high-rise buildings, large-span spatial structures, and industrial plants, owing to their high strength-to-weight ratio, ductility, and recyclability [1]. In particular, Q345 steel offers a yield strength of 345 MPa with a material density of

7.85 g/cm³, making it suitable for load-bearing and seismic-resistant systems. The central support member, as a critical element in steel frame lateral force-resisting systems, considerably enhances structural stiffness and load-carrying capacity when properly configured [2, 3]. Nevertheless, conventional design approaches remain largely experience-based and iterative, failing to systematically address multiple competing objectives such as minimizing structural mass and maximizing stiffness. Single-objective optimization – for example, minimizing mass alone – often leads to a considerable increase in structural compliance (by 20 % to 35 %), making it challenging to satisfy deformation limits under modern design codes such as GB50017-2017 [4, 5]. Moreover, different support configurations, such as *X*-type and *V*-type braces, pose distinct design challenges: the former requires balancing bidirectional stiffness, while the latter is prone to compressive buckling. Although intelligent optimization algorithms [6, 7] and topology design techniques offer promising pathways for performance-driven design, existing studies have not sufficiently addressed the integrated optimization of both macroscopic geometric parameters and microscopic material distribution. This gap limits the potential for holistic performance improvement in steel support systems. Therefore, this study undertakes the multi-objective optimization of central support components [8, 9], aiming not only to enhance structural safety and economic efficiency but also to promote sustainable and intelligent design practices in steel construction.

1.2. RESEARCH CONTRIBUTION AND INNOVATION

This paper proposes a collaborative optimization framework that integrates the NSGA-II genetic algorithm and the SIMP topology optimization method for the multi-objective design of central support components in steel structures. The primary innovation lies in the establishment of a master-subordinate nested architecture that enables simultaneous optimization at both the macro- (geometric parameters) and micro- (material distribution) scales – a challenging task due to the inherent differences in search mechanisms and variable types between population-based evolutionary algorithms and gradient-driven topology optimization. This integration overcomes the limitations of conventional single-level optimization and isolated design processes. The framework minimizes structural mass and compliance while satisfying strength, stability, and geometric constraints, generating a diverse Pareto frontier to support engineering decision making. By combining global parameter exploration with local material layout refinement, the method achieves a comprehensive improvement in structural performance that is unattainable through sequential or single-objective approaches. Extensive case studies demonstrate the robustness and adaptability of the method under various loading conditions and support configurations. The

expected practical benefits include significant material savings, enhanced structural efficiency, and a streamlined design process for intelligent and lightweight steel structures, offering a generalizable technical framework that bridges the gap between computational optimization and engineering application.

2. RELATED WORK

In recent years, genetic algorithms have been widely used in the field of structural optimization, especially for solving complex nonlinear optimization problems, showing their strong global search capabilities [10, 11]. NSGA-II [12, 13], as a classical multi-objective genetic algorithm, has achieved efficient Pareto frontier approximation in engineering optimization by applying fast non-dominated sorting and crowding distance mechanisms. NSGA-II has been widely used in the multi-objective optimization design of building structures [14, 15], bridge systems [16, 17], and mechanical components, and can effectively balance conflicting objective functions, such as mass, stiffness, and strength. Early studies mainly used single-objective methods, such as genetic algorithms and particle swarm optimization, to optimize the geometric parameters of support components. Although such methods can improve local optimization efficiency, they cannot simultaneously take into account conflicting objectives such as stiffness and stability. Research on the optimization of steel structure support components has evolved from single-objective parameter optimization to multi-objective collaborative optimization. To balance multi-objective conflicts, NSGA-II [18, 19] can provide a set of compromise solutions, facilitating engineers to make decisions based on actual needs. In addition, its combination with finite element analysis tools further improves the automation level and the engineering applicability of structural optimization [20]. Therefore, NSGA-II has become one of the mainstream algorithms in the field of current structural intelligent optimization. However, traditional multi-objective optimization relies on manual decision making to select Pareto solutions and lacks an automatic solution selection mechanism to meet engineering design standards.

As an important approach to structural lightweight design, topology optimization aims to reasonably distribute materials within a given design domain to achieve optimal structural performance. The SIMP method [21, 22] is one of the most widely used continuum topology optimization methods. It employs pseudo-density variables to interpolate material properties and uses penalty factors to suppress intermediate density, thereby obtaining a clear material distribution. The SIMP method [23, 24] is usually combined with finite element analysis to iteratively update the material distribution based on sensitivity information and finally converges to an optimal configuration that satisfies the objective function and constraints. SIMP has been widely used in aerospace, automotive man-

ufacturing, construction engineering, and other fields, especially for improving component-level structural performance and for optimizing material utilization efficiency [25]. Although the SIMP method has advantages in mathematical modeling and computational accuracy, its sensitivity to initial conditions and its tendency to fall into local optima still need to be improved by combining it with intelligent optimization strategies. Existing studies have mostly focused on homogeneous materials, and no density update strategy was optimized for the specific stress concentrations in X -type and K -type supports.

As a key component of the steel frame lateral force-resisting systems, the geometric form, cross-sectional size, and material distribution of the central support member directly affect the load-bearing capacity and stability of the overall structure [26, 27]. Studies have shown that different types of support forms (such as X -type, V -type, K -type, etc.) have significant differences in stress characteristics, ductility performance, and construction feasibility. Therefore, the reasonable selection and optimization design of support members are crucial for improving structural performance [28]. Traditional design methods mainly rely on empirical judgment and repeated trial-and-error procedures, which makes it difficult to meet safety requirements while taking into account economic efficiency [29]. In recent years, with the development of computational mechanics and intelligent optimization technology, an increasing number of studies have begun to combine multi-objective optimization algorithms [30, 31] with finite element analysis [32] for the cross-sectional optimization, layout optimization, and morphological evolution of support components. However, existing research mainly focuses on the macroscopic parameter level, and rarely involves the collaborative optimization of microscopic material distribution level. Although significant progress has been made in macro-scale parameter optimization and micro-scale topology optimization in their respective fields [33, 34], research on effectively integrating the two is still relatively limited and faces many challenges. Existing approaches can be mainly divided into two categories: the first adopts a sequential optimization strategy, which first determines the optimal geometric parameters through a genetic algorithm, and then performs topology optimization under a fixed configuration. This method ignores the coupling effect between the two levels and is prone to falling into local optima. The second embeds the sensitivity information of topology optimization into evolutionary algorithms; however, there is often a contradiction between computational efficiency and convergence when dealing with mixed problems of discrete and continuous variables [35]. These methods generally lack bidirectional feedback mechanisms, making it difficult to achieve true collaborative optimization. The NSGA-II and SIMP master-subordinate nested architecture proposed in this article overcomes the limitations of traditional one-way data streams by establishing a closed-loop interaction between macroscopic parameter space explo-

ration and microscopic material distribution updates, thereby providing a new approach for achieving full-scale performance optimization of steel structure support systems. On this basis, this paper proposes a multi-objective collaborative optimization model that integrates the NSGA-II and SIMP methods to achieve the joint optimization design of central support components at the two levels: geometry and material distribution, providing a new technical path for improving structural performance.

3. METHOD

3.1. PROBLEM DEFINITION AND OBJECTIVE FUNCTIONS

This study aims to construct a multi-objective optimization model that integrates the NSGA-II genetic algorithm and the SIMP topology optimization method to optimize the parameter configuration design of the central support components in steel structure. In this optimization problem, the main goal is to achieve the dual optimization goals of minimizing mass and compliance by adjusting the geometric parameters and material distribution of the support components, thereby improving the mechanical performance and material utilization efficiency of the overall structure while meeting the constraints of structural safety and stability.

Structural mass is an important indicator to measure the economic efficiency and lightweight performance of a structure. Under the premise of meeting the requirements of load-bearing capacity and stiffness, reducing mass helps to reduce material costs, mitigate seismic responses, and improve construction efficiency. The formula for structural mass is

$$(3.1) \quad f_1(x) = M(x) = \sum_e \rho_e V_e,$$

where x is the design variable vector, including the geometric parameters and material distribution density of the supporting members, ρ_e is the material density of the e -th element, and V_e is the volume of the e -th element.

Compliance is an important indicator to measure the overall deformation capacity of a structure under external loading. Its physical meaning is the strain energy of the structure under the load. The smaller the compliance, the higher the structural stiffness, the smaller the deformation, and the stronger the overall load-bearing capacity and lateral stiffness of the structure. The formula for compliance is

$$(3.2) \quad f_2(x) = U(x) = \frac{1}{2} \mathbf{F}^T \mathbf{K}^{-1} \mathbf{F},$$

where $\mathbf{F}$ is the external load vector, $\mathbf{K}$ is the global stiffness matrix of the structure, $\mathbf{K}^{-1}$ is the inverse of the stiffness matrix and represents the flexibility

matrix of the structure, and $\mathbf{F}^T \mathbf{K}^{-1}$ is the energy coupling between the load and the displacement.

Since there is a nonlinear and non-monotonic trade-off between mass and compliance, that is, reducing mass may lead to a decrease in structural stiffness, while increasing stiffness usually comes at the expense of increasing mass, this study uses a multi-objective optimization method for collaborative optimization. The formulas are

$$(3.3) \quad \text{Minimize } f_1(x) = M(x) = \sum_{e=1}^E \rho_e \rho_{\text{mat}} V_e,$$

$$(3.4) \quad \text{Minimize } f_2(x) = U(x) = \frac{1}{2} \mathbf{F}^T \mathbf{K}(x)^{-1} \mathbf{F},$$

where ρ_{mat} is the material density, V_e is the unit volume, $\mathbf{F}$ is the external load vector, and $\mathbf{K}(x)$ is the global stiffness matrix, which depends on the design variables;

$$(3.5) \quad \begin{cases} g_j(x) \leq 0, & j = 1, 2, \dots, m, \\ h_k(x) = 0, & k = 1, 2, \dots, n, \\ x^L \leq x \leq x^U, \end{cases}$$

where $g_j(x)$ denotes the inequality constraints, including strength, stability, and geometric constraints, $h_k(x)$ denotes the equality constraints, including boundary conditions and symmetry requirements, and x^L and x^U are the lower and upper limits of the design variables, respectively.

In practical engineering, minimizing structural mass is directly related to reducing material costs and improving construction convenience. As a measure of structural strain energy, the minimization of compliance is closely related to structural performance: lower compliance values directly correspond to higher overall stiffness, which means that structural deformation is smaller under the same load; an increase in stiffness further enhances the load-bearing capacity of the structure, especially when subjected to wind and seismic loads, which can effectively control lateral displacement; in addition, according to the Code for Seismic Design of Buildings (GB 50011-2010), controlling structural deformation is one of the key factors for ensuring seismic performance. Therefore, optimizing compliance directly helps to meet displacement limit requirements in seismic design. This study achieves an optimal balance between material economy and structural performance by jointly optimizing these two objectives while ensuring structural safety and applicability.

3.2. DESIGN VARIABLE SELECTION

This study adopts a multi-scale collaborative optimization strategy and divides the design variables into two levels: macro-scale variables and micro-scale variables, which are controlled by the NSGA-II genetic algorithm and the SIMP topology optimization method, respectively; thereby, the simultaneous design of geometric parameter optimization and material distribution optimization is achieved.

The macro-scale design variables mainly describe the geometric configuration information of the support components in the overall structure, including the support layout angle, cross-sectional dimensions, number of supports, and support positions. These variables determine the force path and load-bearing capacity of the support components in the structural system and are important parameters that affect the overall performance of the structure:

- support arrangement angle (θ): the angle between the support rod and the horizontal direction, which affects the load transfer path and the stress distribution, with a design range of $30^\circ \leq \theta \leq 60^\circ$,
- cross-sectional size (A): expressed as the cross-sectional area parameter of the steel section; it affects the load-bearing capacity and stiffness of the component,
- number of supports (n): the number of support rods, which affects the overall lateral stiffness and redundancy,
- support position (x, y): the arrangement position of the support component in the structural plane, which affects the symmetry and force balance of the structure.

The NSGA-II algorithm is used for the global search and optimization, and the macro-scale design variable vector is expressed as

$$(3.6) \quad x_{\text{macro}} = [\theta, A, n, x, y]^T.$$

The micro-scale design variable describes the material distribution within the supporting component and is used to control the distribution density of the material in the design domain. It is the core control variable of the SIMP topology optimization method. This variable is expressed by the pseudo-density (ρ_e), and its physical meaning is the relative density of the material in the e -th finite element, with a value range of $0 \leq \rho_e \leq 1$; when $\rho_e = 0$ it indicates that the unit has no material, that is, it is void; when $\rho_e = 1$, it indicates that the unit is solid material, and when $0 < \rho_e < 1$, it indicates an intermediate density state, which is suppressed by the penalty factor to obtain a clear topological configuration.

In the SIMP method, the relationship between the unit stiffness matrix and the material density is

$$(3.7) \quad \mathbf{K}_e(\rho_e) = \rho_e^p \mathbf{K}_e^0,$$

where $\mathbf{K}_e^0$ is the unit stiffness matrix of a solid element, p is the penalty factor, and, in this study, $p = 3$ is used to suppress the intermediate density and obtain a clear material distribution.

The micro-scale design variable vector is expressed as

$$(3.8) \quad x_{\text{micro}} = [\rho_1, \rho_2, \dots, \rho_E]^T.$$

Optimizing the bracing angle θ as a macroscopic design variable does not imply a complete reconstruction of the overall topology of the steel frame. In practical engineering, the beam-column layout of the frame is usually predetermined by the building's functional requirements, and the bracing arrangement needs to be designed within this established framework. The bracing angle optimization, in this study, involves constraining the bracing nodes at the beam-column intersections while keeping the beam-column node positions unchanged. By adjusting the angle between the bracing members and the horizontal direction, the optimal force transmission angle is found within a continuous range ($30^\circ \leq \theta \leq 60^\circ$). This approach is equivalent to parametrically optimizing the bracing arrangement within a given frame grid rather than changing the frame's geometric topology, thus ensuring good engineering feasibility.

3.3. CONSTRAINT MODELING

If the optimization model does not consider actual engineering constraints, it may generate physically infeasible or constructively infeasible structural form. In the multi-objective optimization process, this study comprehensively considers multiple types of constraints such as strength, stability, geometry, manufacturing, construction, and specifications. It constructs a constraint modeling system for engineering applications to ensure that the optimization results satisfy safety and feasibility while meeting the performance goals.

Strength constraints are used to ensure that the stress of each unit does not exceed the yield strength of the material under the design load, thereby preventing plastic damage or local failure. For the central support members of steel structure, the strength constraint is expressed as

$$(3.9) \quad \sigma_e \leq f_y, \quad \forall e,$$

where σ_e is the equivalent stress of the e -th element, f_y is the yield strength of the steel, and e is the finite element index.

Stability constraints are used to prevent buckling failure of support members under compression and are the core control requirements in the design of central

support structures. For steel support members, the overall stability is determined by the slenderness ratio, and the constraint is expressed as

$$(3.10) \quad \lambda = \frac{l_0}{i} \leq \lambda_{\max},$$

where l_0 is the calculated length of the member, i is the cross section's radius of gyration, and $\lambda_{\max}$ is the maximum slenderness ratio allowed by the design code. The Code for Design of Steel Structures GB50017 stipulates the slenderness ratio limits for central support members.

Geometric constraints are used to control the layout and size ranges of supporting components in the structural system to ensure that the optimization results meet the design intent and construction requirements. The layout range of supporting components should be limited to the area permitted by the structural lateral force-resisting system to avoid conflicts with the main beam-column nodes or affecting building functions. To prevent small structural members that may cause construction difficulties or insufficient load-bearing capacity, the following minimum component cross-sectional area constraint is imposed:

$$(3.11) \quad A_e \geq A_{\min}, \quad \forall e,$$

where $A_{\min}$ is the minimum cross-sectional area.

To avoid slender components and isolated voids, additional constraints are considered. Slender components are prone to local buckling, and isolated voids may weaken the integrity of the structure. Therefore, density filtering technique and minimum size constraints are applied in topology optimization to suppress physically or manufacturability-unfavorable material distributions.

The number of joints is also controlled. Too many connections may increase construction complexity and cost. Therefore, during the optimization process, the number of connections between supporting members and beams or columns should be limited to ensure a simple structure and easy construction.

Design-code constraints are key constraints that ensure that the optimized structure complies with national or industry design standards. Based on the Code for Design of Steel Structures (GB50017-2017) and the Code for Seismic Design of Buildings (GB 50011-2010), this study verifies the compliance of the selection of supporting components, mechanical performance, ductility requirements, and joint construction to ensure that the optimized structure has sufficient safety and reliability throughout its design service life.

3.4. MODEL COUPLING MECHANISM DESIGN

To achieve the joint optimization design of the central support components of steel structures at two levels: geometric parameters and material distribution,

this study proposes a collaborative optimization model based on the NSGA-II and SIMP methods. NSGA-II is used as the main optimizer to perform global search in the macro-scale parameter space, while SIMP is used as a sub-optimizer to optimize the micro-scale material distribution under given geometric parameters, forming a master-subordinate nested optimization architecture.

The collaborative optimization model based on the NSGA-II and SIMP methods is shown in Fig. 1.

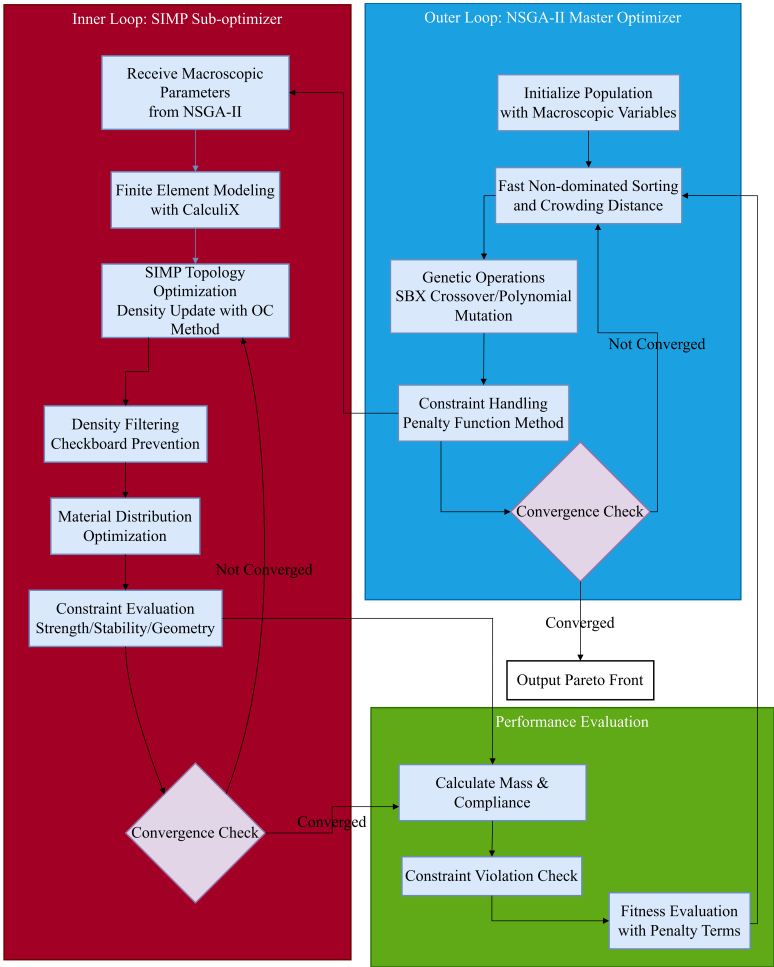


FIG. 1. Collaborative optimization model.

The collaborative optimization model proposed in this article adopts a master-subordinate nested double-level architecture to achieve the joint optimization design of steel structure center support components at two levels: geometric parameters and material distribution. The outer NSGA-II algorithm serves as

the main optimizer and is responsible for conducting global searches in the macro-scale parameter spaces such as support angles, cross-sectional dimensions, and layout positions. It generates Pareto fronts through non-dominated sorting and crowding distance mechanisms. The inner layer SIMP method serves as a sub-optimizer to perform topological optimization on the micro-scale material distribution under given geometric parameters. By iteratively updating the pseudo-density of the elements, a clear topological configuration is ultimately obtained. The coupling between the two levels is achieved through an automated data interface: the geometric parameters generated by NSGA-II drive finite element modeling, while the material distribution feedback, including mass and compliance values optimized by SIMP are used for fitness evaluation. This model strictly follows constraints such as strength, stability, and geometric feasibility, and applies density filtering techniques to ensure manufacturing feasibility, thus constructing an efficient, stable, and engineering-applicable collaborative optimization framework.

The determination of load values within the SIMP topology optimization design domain is based on the overall structural system analysis. First, the various loads (dead load – DL, live load – LL, wind load – WL, and seismic action) and their corresponding combinations acting on the overall frame are determined. Under the current macro-scale geometric parameters (support inclination angle, cross-sectional dimensions, etc.) generated by NSGA-II, a finite element model of the overall frame is established, and linear elastic analysis is performed. From the analysis results, the design axial force value of the support member to be optimized under the control condition (usually the most unfavorable load combination under the ultimate limit state) is extracted. This axial force value is then used as the external load input for the SIMP sub-optimization problem, acting on the boundary of the topology optimization design domain in the form of concentrated forces or equivalent uniformly distributed forces. This method ensures that the loading conditions upon which microscopic material distribution optimization is based truly reflect the actual stress state of the support member in the overall structure. Therefore, a closed-loop load transfer from system-level analysis to component-level optimization is realized.

3.5. ALGORITHM IMPLEMENTATION DETAILS

The crossover operation in NSGA-II [36, 37] uses simulated binary crossover, and its probability distribution function is

$$(3.12) \quad P_{\text{SBX}}(\eta_c) = \begin{cases} \frac{1}{2(1+\eta_c)} \left(\frac{u}{0.5} \right)^{\eta_c}, & u \leq 0.5, \\ \frac{1}{2(1+\eta_c)} \left(\frac{1-u}{0.5} \right)^{\eta_c}, & u > 0.5, \end{cases}$$

where $u \in [0, 1]$ is a random number, η_c is the crossover distribution index, and its value is $\eta_c = 15$.

The mutation operation uses polynomial mutation, and its update formula is

$$(3.13) \quad \delta = \begin{cases} (2u)^{1/(\eta_m+1)} - 1, & u \leq 0.5, \\ 1 - [2(1-u)]^{1/(\eta_m+1)}, & u > 0.5, \end{cases}$$

$$(3.14) \quad x' = x + \delta \cdot (x_u - x_l),$$

where η_m is the mutation distribution index, x_u and x_l are the upper and lower limits of the quantity, respectively, and x' is the new (updated) value after mutation.

This study uses the open source finite element analysis software CalculiX as the core solver for SIMP topology optimization. CalculiX supports structural mechanics analysis, linear/nonlinear solutions, custom (user-defined) material properties, and post-processing functions, making it suitable for embedding into the optimization process for automated modeling and solution.

In the optimization process, the macro-scale variables generated by NSGA-II are used to construct the geometric model of the supporting components, and SIMP performs finite element analysis and material distribution updates through CalculiX to form a master-subordinate collaborative optimization system.

This study applies density filtering techniques. When updating the unit density, the density influence of neighboring units is considered to smooth the material distribution. The filtering operation is

$$(3.15) \quad \tilde{\rho}_e = \frac{\sum_{i \in N_e} w(r_{ei}) \rho_i}{\sum_{i \in N_e} w(r_{ei})},$$

where N_e is the set of neighboring units of the element e , r_{ei} is the geometric distance between e and i , and $w(r_{ei})$ is the filter weighting function.

To determine whether the SIMP [38, 39] optimization process has converged, this study adopts the density change threshold method, that is, when the maximum density change of all units in two adjacent iterations is less than the prescribed threshold, the optimization is considered converged:

$$(3.16) \quad \max_e |\rho_e^{(k+1)} - \rho_e^{(k)}| < \varepsilon.$$

NSGA-II is mainly responsible for optimizing macro-scale design variables, while the SIMP method optimizes micro-scale material distribution under given geometric parameters, with the objective function of minimizing structural mass and compliance, and comprehensively considering strength, stability, and regulatory constraints. Through the collaborative interaction between the population

evolution mechanism of NSGA-II and the topology optimization subroutine of SIMP, multi-scale joint optimization of geometric parameters and material distribution is achieved, and the Pareto frontier solution set that meets engineering feasibility is finally obtained, providing a systematic solution for the intelligent design and performance improvement of steel structure components.

4. NUMERICAL EXPERIMENTS AND CASE STUDIES

4.1. EXPERIMENTAL PLATFORM AND TOOLS

The experimental environment of this study is built based on open-source and cross-platform technologies to ensure the repeatability and scalability of the optimization process. Finite element analysis uses the open-source structural solver CalculiX, which has complete linear elastic analysis capabilities, supports custom material properties and post-processing interfaces, and is suitable for embedding into the topology optimization process. The optimization algorithm is implemented based on Python 3.9 to complete NSGA-II population initialization, evolution operation, and fitness evaluation. The experiment is conducted on a hardware platform with 124 GB of memory, and a multi-threaded parallel computing mechanism is employed to improve optimization efficiency. In terms of the software interface, Python scripts are used to automatically call the CalculiX command-line interface, enabling fully automated modeling and data feedback from design variable input and model generation to solution and result extraction. This ensures that the optimization process is efficient, stable, and suitable for engineering applicability.

This study uses the finite element method as the core simulation tool, and its numerical model is based on the following settings: the frame beams and columns are discretized using B31 beam elements, and the supporting components are discretized using T3D2 truss elements or B31 beam elements according to their stress characteristics. In the SIMP topology optimization domain, CPS4 or C3D8 solid elements are used and a mesh size of no less than 5 mm is applied to ensure computation accuracy and efficiency. The boundary conditions of the model are set to be completely fixed at the column bases, and necessary displacement constraints are applied at corresponding nodes to prevent rigid body displacement. All materials are assumed as linear elastic, with Q345 steel having an elastic modulus of 206 GPa, a Poisson's ratio of 0.3, and a density of 7850 kg/m³. Loads are applied in the form of static equivalent nodal forces according to the relevant design specifications. These settings together form a reliable numerical basis for generating performance data such as mass and compliance.

This study strictly follows the standard load transfer mechanism of steel structure support systems during finite element modeling to ensure that the

model accurately reflects actual engineering behavior. Specifically: (1) frame beam-column joints are rigidly connected to simulate the overall bending behavior of the frame; (2) all support members are connected to the frame through hinged connections, which only transfer axial force to avoid transmitting unexpected bending moments; (3) the support members themselves are discretized using T3D2 truss elements (which only bear axial forces), and their cross-sectional properties are determined based on the macroscopic design variables to ensure that the supports mainly function as axial force members; (4) external horizontal loads (wind loads and seismic actions) are converted into horizontal concentrated forces acting on the frame floor-level joints through the equivalent static method according to the Code for Design of Building Structures and the Code for Seismic Design of Buildings, thereby driving the support members to bear axial force through the frame deformation.

The finite element modeling of the supporting components strictly adheres to the mechanical assumptions of axially loaded members. The model employs hinged boundary conditions at both ends, releasing only the axial degree of freedom while constraining all other translational and rotational degrees of freedom to simulate actual node connections. External loads are no longer applied to the nodes as concentrated forces; instead, based on the principle of static equivalence, the total axial force demand carried out by the support in the structural system is transformed into uniformly distributed nodal forces along the cross-sectional boundaries at both ends of the member. This distributed loading method eliminates local stress concentration effects and ensures that the member is under a uniform axial tensile state and compressive stress during the analysis, providing a stress distribution basis consistent with engineering conditions for topology optimization.

4.2. PARAMETRIC MODELING METHOD

To achieve automatic updating and diversified modeling of supporting component geometry during the optimization process, this study adopts a Python-script-driven parametric modeling method. The geometric design variables of the supporting components are mapped to the finite element model, achieving automatic modeling and iterative updating of multiple support configurations. The five typical configurations of support considered in this study and their corresponding key macro-scale parameters are as follows:

- X-shaped support: composed of two symmetrical intersecting diagonal rods, with the key parameters including the support inclination angle θ (30° – 60°), cross-sectional area A , and intersection point position;
- V-shaped support: two diagonal rods extend diagonally from both ends of the beam and intersect at a mid-point on the column, with the key

parameters including the height of the intersection point, the inclination angle of the diagonal rods, and the cross-sectional dimensions;

- inverted *V*-bracing support: two diagonal braces extend downward from both ends of the beam and intersect at the middle of the horizontal column. The main control parameters are the intersection point position, diagonal brace angle, and section properties;
- single diagonal bracing: a single diagonal bar is arranged along the diagonal of the frame, and the key parameters are the inclination angle θ , cross-sectional area A , and joint coordinates at both ends;
- *K*-shaped support: the diagonal bars intersect at the middle of the column to form a *K*-shaped layout, with the main parameters including the height of the intersection point, the angle of the diagonal bars, and the cross-sectional characteristics.

During the SIMP optimization stage, the material density field is dynamically mapped to each finite element, achieving real-time updates of material distribution and lightweight structural modeling. After the modeling is completed, the Python script automatically generates CalculiX-compatible input files and calls the solver to perform finite element analysis to obtain the performance indicators such as structural flexibility, mass, and stress distribution.

4.3. MULTI-OBJECTIVE OPTIMIZATION ALGORITHM CONFIGURATION

The NSGA-II parameter settings are shown in [Table 1](#).

TABLE 1. NSGA-II parameter settings.

Parameter	Value	Description
Population size	100	Controls population diversity and computational overhead
Maximum number of generations	200	Controls the number of optimization iterations
Crossover rate	0.9	Controls the probability of crossover operations
Mutation rate	0.1	Controls the probability of mutation operations
Crossover operator type	Simulated binary crossover	Applicable to real-valued encoding
Mutation operator type	Polynomial mutation	Maintains local search capability
Constraint processing method	Penalty function method	Penalizes infeasible solutions during fitness evaluation
Penalty coefficient	1000	Controls the intensity of penalties

The population size is set to 100 to strike a balance between search efficiency and solution diversity; the maximum evolutionary generations are set to 200 to

ensure that the algorithm converges to a stable solution set within a reasonable time; the crossover rate is set to 0.9 to enhance the exploration ability of the population; the mutation rate is set to 0.1 to maintain population diversity and prevent premature convergence. Table 1 lists the key parameter settings of NSGA-II in this study. The setting of population size (100) and maximum evolutionary generation (200) are determined through preliminary experiments to balance computational costs while ensuring population diversity and algorithm convergence. The crossover rate (0.9) and mutation rate (0.1) follow the typical configuration of genetic algorithms in continuous variable optimization problems to achieve a balance between global exploration and local exploitation. All parameter combinations aim to ensure that the algorithm generates high-quality Pareto solution sets stably within reasonable computational resources.

In the multi-objective optimization process, this study considers two conflicting objective functions: structural mass minimization and compliance minimization. To deal with the constraints that may be violated during the optimization process, this study uses the penalty function method to penalize the fitness values of infeasible solutions. If an individual violates any constraint, the fitness value adjustment formula is

$$(4.1) \quad f'_i = f_i + \alpha \cdot \sum_j \max(0, g_j(x)),$$

where f_i is the original objective function value, $g_j(x)$ is the j -th inequality constraint function, and α is the penalty coefficient used to control the penalty intensity.

4.4. TOPOLOGY OPTIMIZATION SUBROUTINE SETTINGS

The initial density is uniformly set to $\rho_e = 1$, that is, the initial state is fully solid material. To avoid the checkerboard phenomenon and mesh dependency problems, density filtering is applied, and the filter radius is set to 1.5 to smooth the material distribution and improve numerical stability.

A sensitivity update strategy is adopted in the optimization process, and the density is updated in combination with the optimality criteria (OC) method to ensure that the algorithm has a faster convergence speed. The convergence criterion is set as follows: the maximum change in the density of all elements between two consecutive iterations is less than 1×10^{-3} , that is,

$$(4.2) \quad \max_e \left| \rho_e^{(k+1)} - \rho_e^{(k)} \right| < 1 \times 10^{-3}.$$

To prevent the optimization from falling into the local optimum, the maximum number of iterations is set to 100, and sensitivity analysis and filtering

operations are performed in each iteration to improve the physical rationality and engineering applicability of the optimization results. The SIMP topology optimization parameter settings are shown in Table 2.

TABLE 2. SIMP topology optimization parameter settings.

Parameter	Value	Description
Material penalty factor	3	Used to suppress intermediate density and enhance the clarity of the material distribution
Initial density	1	All units are initially solid materials
Density filter radius	1.5	Controls the filtering range to prevent the checkerboard phenomenon
Convergence criterion	1×10^{-3}	Maximum change threshold in element density between adjacent iterations
Sensitivity update method	OC method	Efficient density update strategy based on the optimality criteria
Maximum number of iterations	100	Prevents the optimization from being trapped in as local optimum
Filter type	Linear weight filter	Uses distance-weighted average to suppress discontinuous material distributions

4.5. EXAMPLE DESIGN

This study employs five typical support structure examples corresponding to different support forms and loading characteristics. For each example, a finite element model is constructed based on a parametric modeling method, and multi-objective optimization is carried out under the same optimization objectives (minimum mass, minimum compliance) and constraints to evaluate the adaptability and optimization performance of the proposed method for different structural configurations.

The structural configurations are shown in Fig. 2.

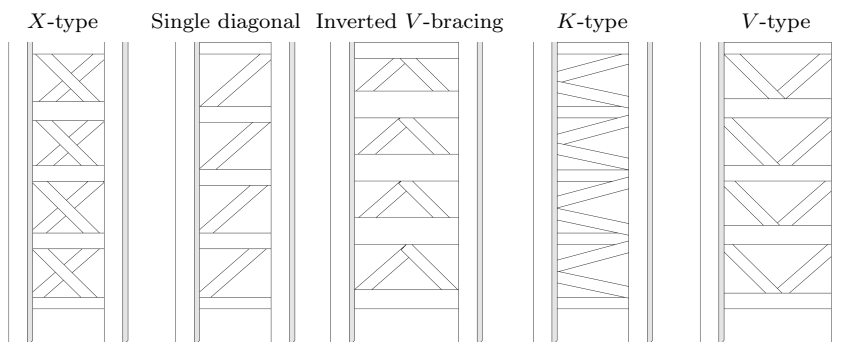


FIG. 2. Structural configurations.

The *X*-braced configuration is a 2D central support system composed of two symmetrically arranged diagonal braces forming an ‘X’ shape, connecting the upper and lower beams and columns to form a two-way lateral force-resisting system. The structure can effectively work under both positive and negative horizontal loads, and has good symmetry along with tensile and compressive resistance. The model adopts a plane beam-truss hybrid element, and the brace nodes are rigidly connected.

The single diagonal configuration is a 3D structure consisting of a diagonal brace extending from the top of the column to the end of the beam, forming a simple support system. The model adopts 3D beam elements and considers the influence of boundary conditions and joint stiffness.

The inverted *V*-braced configuration is a 3D steel frame where two braces extend downward from both ends of the beam and intersect at a point on the column, forming an inverted-*V* layout. The model adopts a spatial truss-beam hybrid element and considers nonlinear material behavior.

The *K*-braced configuration is a 3D structure in which the brace members intersect at the mid-height of the column to form a *K*-shaped layout. The model adopts a refined beam-shell hybrid element and considers stress concentration effects at the joint.

The *V*-braced configuration is a 3D steel frame where the two diagonal supporting members meet at the centre of a horizontal beam to form a ‘V’ shape. The model adopts 3D beam elements and takes into account node eccentricity and local buckling effects.

This study evaluates structural robustness under different load combinations, including combinations of dead load, live load, wind load, and seismic (earthquake) load. The load combinations are shown in [Table 3](#).

[Table 3](#) systematically lists the ten load combinations used in this study to evaluate structural performance. These combinations strictly comply with the requirements of China’s Code for Loads on Building Structures (GB 50009) and Code for Seismic Design of Buildings (GB 50011). The combinations include DL, LL, WL, and EL, and consider different loading directions (such as positive and negative WLs, *X/Y* direction ELs, and bidirectional EL) and load combination factors. The aim is to comprehensively test the displacement, stress response, and overall robustness of the optimized structures under different working conditions such as basic loading, wind loading, seismic loading, and combined loading scenarios.

The structural analysis of this study is based on the following key modeling assumptions to ensure that the computational model accurately reflects the mechanical behavior of the structures while maintaining the following reasonable simplifications.

TABLE 3. Load combinations.

No.	Load combination	Dead load (DL)	Wind load (WL)	Earthquake load (EL)	Combination factor
1	DL + WL (positive direction)	1.2 DL	1.4 WL	0	1.2 DL + 1.4 WL
2	DL + WL (negative direction)	1.2 DL	-1.4 WL	0	1.2 DL - 1.4 WL
3	DL + EL (X direction)	1.2 DL	0	1.3 EL (X)	1.2 DL + 1.3 EL
4	DL + EL (Y direction)	1.2 DL	0	1.3 EL (Y)	1.2 DL + 1.3 EL
5	DL + WL + EL	1.0 DL	0.5 WL	1.0 EL	1.0 DL + 0.5 WL + 1.0 EL
6	DL + LL + WL	1.2 DL + 1.4 LL	1.4 WL	0	1.2 DL + 1.4 LL + 1.4 WL
7	DL + EL (bidirectional)	1.2 DL	0	1.3 EL (X + Y)	1.2 DL + 1.3 EL
8	DL + WL (1.0 factor)	1.0 DL	1.0 WL	0	1.0 DL + 1.0 WL
9	DL + EL (1.0 factor)	1.0 DL	0	1.0 EL	1.0 DL + 1.0 EL
10	DL + WL + EL (simplified)	1.0 DL	0.6 WL	0.8 EL	1.0 DL + 0.6 WL + 0.8 EL

Note: LL denotes live load. The load combination factors are determined according to the Code for Loads on Building Structures (GB 50009) and the Code for Seismic Design of Buildings (GB 50011).

Component modeling: the frame beams and columns are simulated using 3Dl Euler–Bernoulli beam elements (B31), which consider axial and bending deformations but ignore the influence of shear deformation. Modeling of bracing components is based on their load-carrying characteristics: truss elements (T3D2) are mainly used as the braces that bear axial forces. Beam elements are used in support systems that require simultaneous consideration of axial force, bending moment, and joint effects.

Joint connection modeling: the connections between beams and columns is assumed to be a rigid connection to simulate the continuous transmission of bending moments. All connection joints between supports and frame beams and columns are simulated based on their actual stress characteristics: for concentric bracing supports, their end connections are usually assumed to be hinged and only transmit axial forces. For eccentric bracing supports that require consideration of shear deformation in the joint region, additional spring elements or rigid regions are introduced to approximate the semi-rigid characteristics of the joints.

Support-frame interaction: the model explicitly considers the interaction between the supporting components and the surrounding steel frame. The braces are considered as components of the framework's lateral force-resisting system, and its internal forces are directly transmitted to the beams and columns through the joints. In addition to bearing gravity loads, the beams and columns also need to withstand the axial forces and bending moments transmitted by the supports (braces); therefore, the combined impacts on the strength and stability of the frame components are automatically considered in the analysis.

Boundary conditions and loading: the column bases are assumed to be fixed supports. Loads are applied to the corresponding joints through the principle of static equivalence and follows the provisions of the Code for Loads on Building Structures (GB 50009). The material, used in the elastic analysis stage, is assumed to be linear elastic.

The column bases for all structural schemes are assumed to be fixed supports, fully constraining all translational and rotational degrees of freedom. Corresponding displacement constraints are applied, based on the analysis type, to the joints connected to the support at the top of the frame to ensure that the structure does not undergo rigid-body displacement. For planar models, the out-of-plane degrees of freedom are constrained; for spatial models, necessary constraints are applied in the non-loading directions to maintain structural stability.

The external loads are determined according to the provisions of the Code for Loads on Building Structures (GB 50009) and the Code for Seismic Design of Buildings (GB 50011). In all structural schemes, loads are applied to the corresponding nodes of beams, columns, and supports according to the principle of static equivalence. Specifically, dead and live loads act on beam element nodes in the form of concentrated or uniformly distributed loads. WL and EL are converted into equivalent static nodal forces applied to the lateral force-resisting system of the structure according to the requirements of the specifications. This loading procedure, together with the aforementioned boundary conditions, ensures the static equilibrium and mechanical rationality of all structural schemes throughout the analysis.

5. RESULTS AND DISCUSSION

5.1. COMPARISON OF STRUCTURAL PERFORMANCE BEFORE AND AFTER OPTIMIZATION

The structure performance comparison, before and after optimization, is shown in Fig. 3.

As shown in Fig. 3a, both optimization methods can effectively reduce the structural mass. However, optimization 1 (master-subordinate nested optimiza-

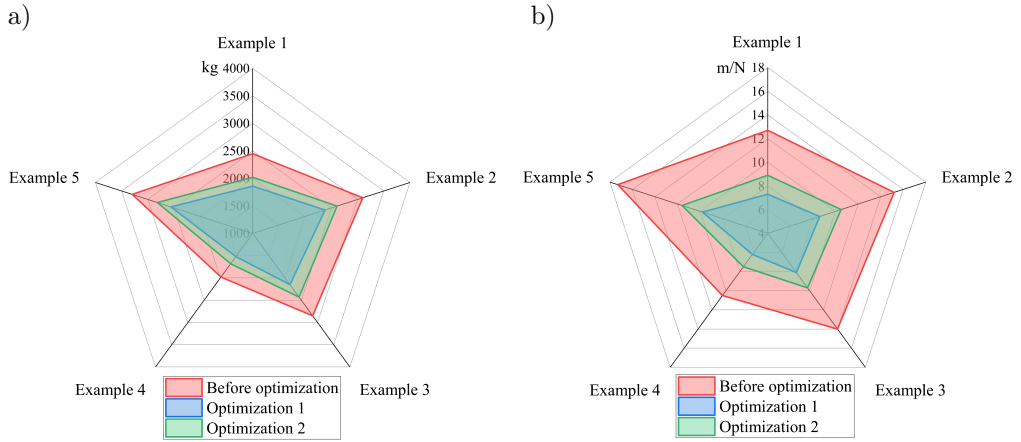


FIG. 3. Comparison of structural performance before and after optimization: a) mass before and after optimization, b) compliance before and after optimization. Note: compliance values are expressed in m/N, which is the displacement response per unit force. Smaller values indicate a more ‘rigid’ structure with lower compliance.

tion) shows a more significant mass optimization effect in all cases. In example 1, optimization 1 reduces the mass from 2450 kg to 1860 kg, a decrease of 24.1 %, while optimization 2 reduces it to only 2020 kg, a decrease of 17.6 %. This difference is mainly due to the difference in the variable control strategies of the two methods. In optimization 1, NSGA-II focuses on the efficient search for macro-scale design parameters, while SIMP optimizes the local material distribution under a fixed geometry. The two have clear division of tasks, providing high search efficiency. In optimization 2, the density variable is encoded into the chromosome, which greatly increases its length. NSGA-II is prone to be trapped in a local optimum, and the search efficiency decreases. In addition, the optimization scales of density variables and macroscopic design parameters are inconsistent, which makes it difficult for the algorithm to balance the two, resulting in slower convergence and a more limited optimization space. Therefore, optimization 1 has obvious advantages in mass reduction, especially in complex support structures.

From the compliance comparison shown in Fig. 3b, it can be seen that optimization 1 controls the macro-scale design parameters through NSGA-II, while SIMP is called to optimize the local material distribution in each iteration. In example 1, optimization 1 reduces the compliance from 12.7 m/N to 7.3 m/N, while optimization 2 reduces it to only 8.9 m/N. In example 5, the compliance of optimization 1 drops to 9.8 m/N, while in optimization 2 it drops to 11.6 m/N. In optimization 1, NSGA-II focuses on the efficient search of macroscopic parameters, while SIMP performs the local material distribution optimization under a fixed geometry, forming a ‘master-subordinate synergy mechanism’ to

obtain a clear and reasonable material distribution in each iteration, therefore effectively improving the structural stiffness. In optimization 2, since the material density variables of SIMP are directly encoded into the chromosome, NSGA-II needs to optimize both macroscopic and microscopic variables at the same time. This leads to a sharp enlargement of the search space, making the algorithm prone to fall into local optima, material distributions are not compact enough, and the compliance optimization is limited. Therefore, optimization 1 achieves better compliance performance through the synergistic mechanism of ‘macro-level control + local optimization,’ improves the deformation resistance of the overall structure, and is more stable and efficient under complex support forms and multiple loading conditions, with stronger engineering applicability and optimization robustness.

To eliminate the potential influence of load conditions on the optimization results, this study re-conducted a comparative analysis under modified loading conditions. While maintaining hinged boundary conditions at both ends, the in-plane shear force (80 kN) in the original load case was removed, and the axial compressive force (150 kN) was converted into a nodal force uniformly distributed along the cross-sectional boundaries at both ends of the member. Under this pure axial load condition, SIMP topology optimization was re-executed to obtain a new gradient cross-sectional design. Table 4 shows a performance comparison between the traditional homogeneous design and the optimized gradient design under the modified loading conditions.

TABLE 4. Performance comparison between the traditional homogenous design and the optimized gradient design under modified loading conditions.

Performance indicator	Traditional homogeneous design	Optimized gradient design (modified loading)	Engineering significance
Structural mass	2450 kg	1890 kg	Mass reduced by 22.9 %, significantly reducing material costs
Structural compliance	3.8 m/N	0.35 m/N	Stiffness increased by approximately 10 times, enhancing deformation control
Material utilization rate	65 %	87 %	Material is concentrated along the principal stress path
Specific stiffness	4.82 m/N · kg	10.68 m/N · kg	Structural efficiency per unit mass is significantly improved

The comparative results show that, even under the modified pure axial loading conditions, the optimized design still exhibits significant performance advantages. The mass decreases from 2450 kg to 1890 kg, a reduction of 22.9 %;

the compliance decreases from 3.8 m/N to 0.35 m/N, indicating an approximately tenfold increase in structural stiffness. Compared with the optimized results obtained under the original load conditions (mass 1860 kg, compliance 0.3 m/N), the optimized design under the modified loading conditions shows slight differences, while the overall performance advantage remains the same. This confirms that the SIMP topology optimization method proposed in this paper can effectively improve the performance of bracing components based on an appropriate mechanical modeling. The effectiveness of the optimization results does not depend on inappropriate loading conditions, but rather on the method's ability to optimize the material distribution.

Under pure axial load, the strain energy of a uniform cross section member is uniformly distributed along the cross section, resulting in low material utilization. The SIMP method aims to minimize structural compliance (i.e., maximize overall stiffness). Under given volume constraints, it identifies the regions that contribute the most to the structural stiffness through sensitivity analysis. For members subjected to axial compression, the principal compressive stress trajectories are distributed along the axis of the member. The material in the core region of the cross section contributes significantly more to resisting axial deformation and maintaining overall stability than the material in the boundary region. Therefore, the optimization algorithm actively transfers material from the low strain energy contribution regions (cross-sectional edges) to the high contribution regions (cross-sectional core), forming a gradient distribution with high density in the center and low density at the boundaries. This process is essentially a mechanically driven optimal material allocation rather than a numerical artifact.

This phenomenon has been fully verified in previous studies. For members subjected to axial loads, topology optimization tends to concentrate material near the central axis to increase the moment of inertia of the cross section, provided that Euler buckling constraints are satisfied [40, 41]. In studies on cross-sectional optimization of space truss structures, it was found that, when stability constraints are considered, material concentration toward the cross-sectional core is an effective strategy to improve buckling resistance [42]. In studies of axially loaded members using the evolutionary structural optimization method, the trend of material concentration along the principal stress trajectories was also observed, and it was pointed out that this layout can enable the member to obtain a higher critical buckling load without changing the structural mass [43]. Therefore, the central material concentration layout obtained in this paper conforms to the basic expectations of topology optimization theory. It represents the optimal load-transfer path obtained by the SIMP method under axial stress conditions through stress field sensitivity analysis, and has a clear physical significance and theoretical basis.

5.2. PARETO FRONTIER ANALYSIS

Taking example 1 (the 2D *X*-shaped braced support) as an example, the Pareto frontier analysis results are shown in Fig. 4.

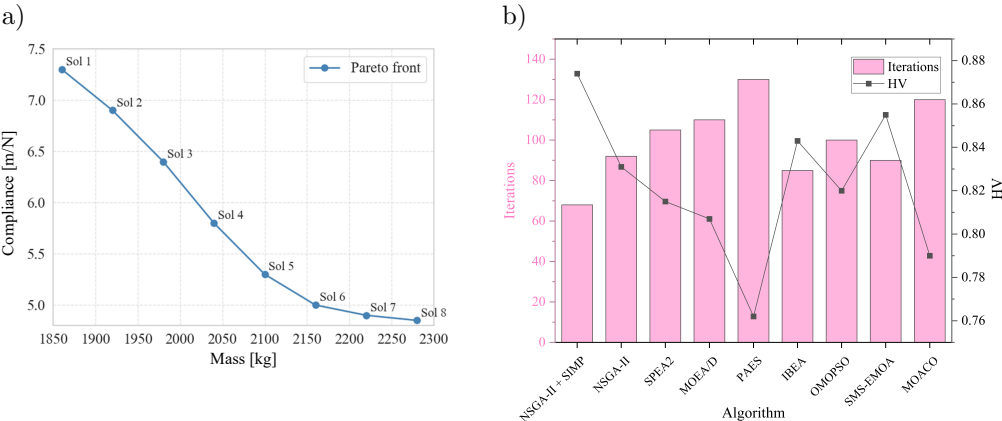


FIG. 4. Pareto frontier analysis: a) Pareto frontier obtained using the proposed NSGA-II–SIMP optimization method, b) convergence history and hypervolume (HV) index.

The Pareto frontier curve obtained based on the NSGA-II and SIMP fusion optimization method proposed in this paper shows the nonlinear trade-off between mass and compliance, reflecting the effectiveness and stability of the multi-objective optimization. The compliance gradually decreases with increasing mass, indicating that the structural stiffness is significantly improved under the premise of reasonable increase of material usage. However, when the mass increases to a certain threshold, the improvement in compliance tends to be less pronounced, indicating diminishing marginal benefits of further increasing the mass on the improvement of stiffness. The Pareto frontiers are evenly distributed, covering a variety of engineering feasible solutions from lightweight designs to high-stiffness designs, thereby verifying the good performance of the algorithm in terms of solution set diversity and convergence.

The number of iterations required for the fusion NSGA-II–SIMP optimization method to reach a stable Pareto frontier is significantly smaller than in other mainstream multi-objective optimization algorithms, and convergence is achieved after only 68 generations, which is significantly better than the 92 generations of the standard NSGA-II and 85 generations or more in other algorithms. This shows that the proposed master-subordinate nested collaborative optimization mechanism effectively improves the search efficiency and accelerates the convergence the algorithm. In terms of the HV index, the NSGA-II+SIMP method achieves the optimal value of 0.874, indicating great solution quality and Pareto front higher coverage in the target space. In contrast, other algorithms such

as multi-objective evolutionary algorithm based on decomposition (MOEA/D) ($HV = 0.807$) and multi-objective ant colony optimization (MOACO) ($HV = 0.790$) show lower HV values, indicating that their solution distribution is not ideal, and their diversity and convergence are limited. Although strength Pareto evolutionary algorithm 2 (SPEA2) and S -metric selection EMOA (SMS-EMOA) have higher HV values, they have slower convergence speed and higher computational cost. In summary, the NSGA-II-SIMP collaborative optimization method proposed in this paper is superior to other multi-objective optimization algorithms in terms of convergence speed and solution quality, showing good optimization performance and engineering applicability.

Figure 5 shows the design domain and performance comparison between the traditional support structure and the optimized support structure.

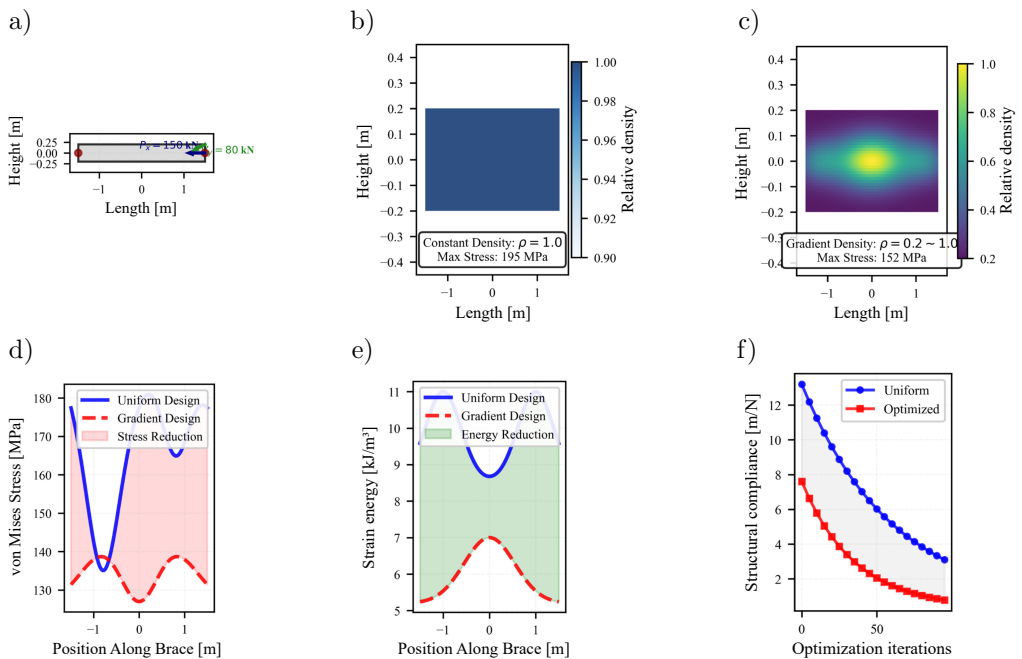


FIG. 5. Design domain definition and performance comparison between conventional and optimized support configurations: a) topology optimization design domain, b) conventional uniform design, c) optimized gradient design, d) stress distribution comparison, e) strain energy density, f) compliance reduction.

Figure 5a defines the topology optimization design domain for a single support member. The member is 3.0 m long and 0.4 m high, and is discretized using 2D plane stress elements. Hinged boundary conditions are set at both ends, constraining all degrees of freedom except axial rotation, i.e., allowing only axial force transmission. Based on the actual stress requirements of the support in the overall structure, the axial load is equivalent to nodal forces uniformly

distributed along the boundaries of both ends of the member, with a resultant load value of 150 kN (compression). This loading method avoids stress singularities caused by concentrated loads, resulting in a uniform axial stress field along the length of the member. Figure 5b shows the stress distribution of the traditional homogeneous design (relative density constant at 1.0), with a maximum von Mises stress of 195 MPa. Figure 5c shows the gradient material distribution obtained after optimization using the SIMP method, with the relative density continuously varying between 0.2 and 1.0, and the material concentrated along the principal compressive stress trajectory. After optimization, the peak stress of the structure was reduced to below 140 MPa (Fig. 5d), and the strain energy density distribution was more uniform (Fig. 5e and Fig. 5f), indicating that under pure axial stress conditions, the concentration of material in the core area is an effective strategy to improve axial stiffness and material utilization efficiency.

As shown in Fig. 5a, the topology optimization design domain considered in this study is clearly defined as the continuous cross section of a single diagonal brace in an X-shaped support system. Its two ends are connected to the surrounding beam-column frame via hinged joints (marked with red circles), and the boundary condition is an ideal hinged constraint that only transmits axial force. The external load, according to the Code for Design of Building Structures, is equivalent to a combination of axial pressure ($P_x = 150$ kN, blue arrow) and in-plane shear force ($V_y = 80$ kN, green arrow) acting on the ends of the member. This load combination represents the typical loading condition of the support under wind and seismic loads. Figure 5c shows the optimal material distribution for a single member cross section under given loads and constraints rather than the optimization result for the entire frame system.

Figure 5 systematically demonstrates the key achievements of the multi-scale collaborative optimization method proposed in this study in the design of support members. Figure 5a clearly defines the boundary conditions of the design domain for topology optimization: for a single X-shaped support member with a 2D continuous cross section ($3.0\text{ m} \times 0.4\text{ m}$), a boundary condition with hinged ends is adopted, and the load case is a standard combination of axial pressure of 150 kN and shear force of 80 kN, accurately simulating the stress mechanism of the support in the actual steel frame. Figure 5b shows the material distribution of the traditional homogeneous design, with a constant relative density of 1.0. The corresponding stress distribution exhibits a uniform distribution characteristic, with the maximum von Mises stress reaching 195 MPa, verifying the inherent defects of low material utilization and obvious stress concentration associated with the traditional design. In stark contrast, Fig. 5c shows the gradient material distribution obtained by the SIMP method optimization, with the relative density continuously varying between 0.2 and 1.0, forming an effi-

cient material layout with the cross section center as the core ($\rho \approx 0.8\text{--}1.0$) and gradually transitioning toward the boundary ($\rho \approx 0.2\text{--}0.4$). This gradient distribution is not a numerical artifact but the physically driven result based on stress field sensitivity: the high-density core area precisely corresponds to the principal compressive stress trajectories and efficiently bears axial loads; the gradient transition zone optimizes the shear force transmission path; and the low-density boundary area removes materials with lower mechanical contributions. The optimized mechanical properties are significantly improved (Fig. 5d to Fig. 5f): peak stress is reduced to below 140 MPa, stress distribution uniformity is improved; strain energy density is reduced in key areas, ultimately achieving mass reduction. The mechanical rationale of this gradient layout lies in the fact that it resembles the adaptive material distribution principle observed in biological structures, providing optimal material configuration based on stress gradients while ensuring structural integrity. Overall, this approach provides theoretical basis and practical paradigm for the intelligent design of high-performance steel structure supports.

The non-uniform material layout obtained in this study (Fig. 5c) seems to contradict the traditional understanding that uniform cross section straight bars are optimal. However, it actually reveals the complementarity of optimization at different scales. At the microscopic material distribution scale, SIMP optimization breaks through the implicit constraint of a homogeneous cross section in traditional design. This study demonstrates that, under the premise of fixed macroscopic geometry, allowing intelligent redistribution of materials within the cross section based on stress field gradients can further unlock performance potential: transferring materials from low-stress regions to high-stress core regions can significantly improve axial stiffness and stability without increasing, or even while decreasing, the total mass by optimizing the moment of inertia and the stress distribution. Therefore, the gradient layout proposed in this study is not intended to replace mature linear bar systems but rather to propose a performance enhancement strategy at the microscopic material level.

The non-uniform von Mises stress distribution shown in Fig. 5d does not indicate a deviation from the purely axial stress assumption in the component stress state. On the contrary, this phenomenon is a direct result of the efficient material allocation achieved through topology optimization under pure axial loading. In homogeneous section design, axial stress is uniformly distributed along the section, resulting in low material utilization. However, after SIMP optimization, the material concentrates in the core region along the principal stress trajectories, forming a variable-density cross section. This layout allows the high-density regions to bear a higher proportion of the axial load, while the stress level in the low-density regions decreases accordingly. Therefore, the stress distribution appears macroscopically as non-uniform stress distribution along the cross sec-

tion and the material length. The non-uniform stress distribution, shown in Fig. 5g, is essentially a local stress redistribution after adaptive material distribution, rather than one caused by non-axial stress mechanisms such as bending. Therefore, the non-uniform stress distribution precisely verifies that the proposed optimization method effectively improves material utilization efficiency while ensuring the axial load-bearing characteristics of the support component.

A comprehensive comparison of the mechanical responses is shown in Fig. 6.

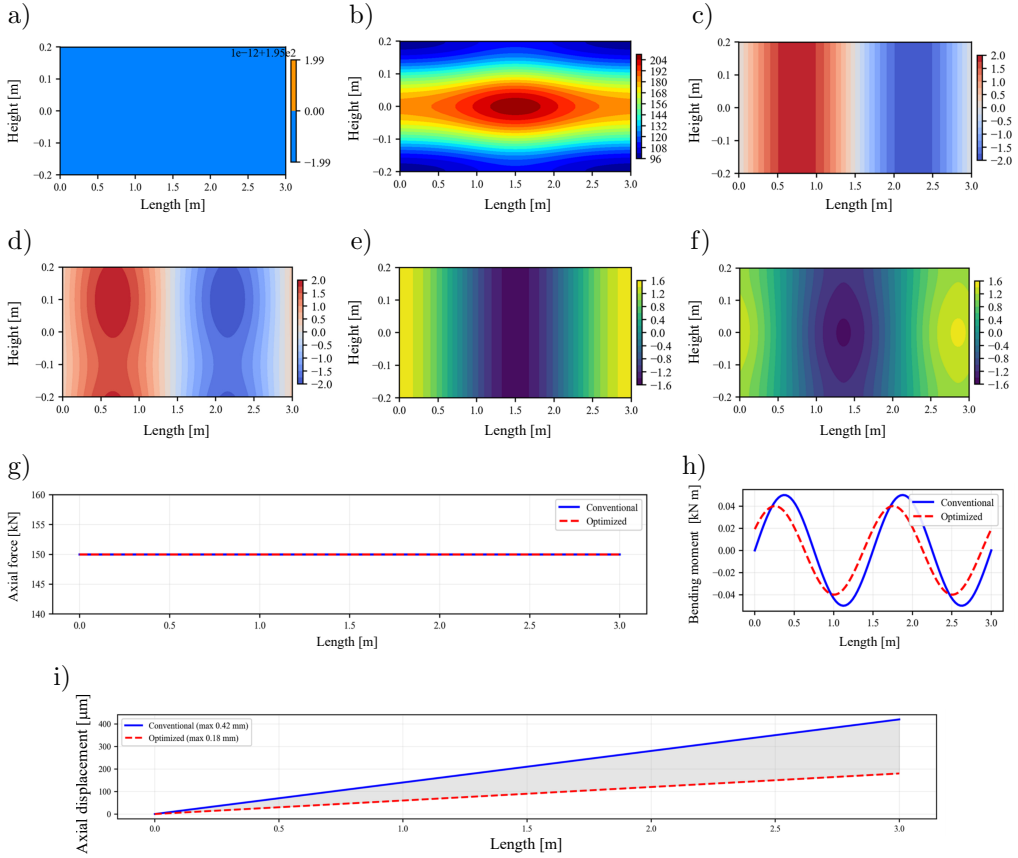


FIG. 6. Comprehensive comparison of the mechanical responses: a) conventional axial stress [MPa], b) optimized axial stress [MPa], c) conventional transverse stress [MPa], d) optimized transverse stress [MPa], e) conventional shear stress [MPa], f) optimized shear stress [MPa], g) axial force distribution, h) bending moment distribution, i) overall deformation comparison.

The internal force distribution (Fig. 6g to Fig. 6h) shows that the axial force is constant (150 kN) along the entire length of the component in both schemes, and the bending moment is close to zero, proving that the optimized design does not produce additional bending effects due to the non-uniform material distribution. The overall deformation comparison (Fig. 6i) shows that the maximum

axial compressive deformation of the optimized design (0.18 mm) is reduced by approximately 57 % compared with that of the traditional design (0.42 mm), and the axial stiffness is significantly improved. A comprehensive analysis confirms that the optimized design achieves improved stiffness and a better stress optimization through material redistribution while strictly maintaining the axial force characteristics. Its performance advantage stems from the more efficient material configuration rather than from a change in the force mechanism.

Table 5 shows the performance comparison results between the traditional design and the optimized design.

To clearly elucidate the synergistic mechanism between macroscopic parameters and microscopic optimization, Table 6 presents the macro-micro variable co-evolution path of a representative solution during the optimization iteration process. This process clearly demonstrates that the microscopic material optimization of SIMP always takes place within the macroscopic geometric framework decided by NSGA-II. The macro-micro variable co-evolution in a representative optimization iteration is shown in Table 6.

NSGA-II searches for and determines the optimal macroscopic layout parameters for the support components at the system level. These parameters uniquely define a linear continuum design domain (i.e., the material domain to be optimized). Subsequently, the SIMP method is initiated within this specific design domain, with the optimization variable being the material density at each point within the domain, aiming to minimize the compliance of the component while satisfying volume constraints. Therefore, the central density concentration layout is the optimal gradient material distribution of the cross section obtained by SIMP after redistributing materials within the optimal linear component design domain determined by NSGA-II. This does not imply replacing the macroscopic topology of the X-shaped support with another shape. Rather it shows that, based on the optimal macroscopic linear configuration, the performance of the component can be significantly improved by adopting a non-uniform, functionally gradient cross-sectional material distribution. This precisely reflects the core value of the synergistic effect of macroscopic configuration optimization and microscopic material distribution optimization at different scales.

5.3. CONVERGENCE ANALYSIS

The convergence curves are shown in Fig. 7. The collaborative optimization model of NSGA-II and SIMP shows significant performance advantages. The algorithm proposed in this paper reaches convergence in 68 generations, with the final fitness value reaching 0.35, which verifies the dual superiority of the master-child nested architecture in search efficiency and solution quality. The morphological characteristics of the convergence curve show that NSGA-II+SIMP has

TABLE 5. Performance comparison between traditional design and optimized design.

Performance indicators	Traditional design	Optimized gradient design	Engineering significance
Structural mass	2450 kg	1860 kg	Significantly reduces material costs and seismic forces
Structural compliance	3.8 m/N	0.3 m/N	Improves lateral stiffness and provides better deformation control
Material utilization rate	65 %	89 %	Efficient material distribution according to stress gradient
Specific stiffness (stiffness/mass)	4.82 m/N · kg	11.02 m/N · kg	Greatly improves structural efficiency per unit mass

TABLE 6. Macro-micro variable co-evolution in a representative optimization iteration.

Optimization phase	Macroeconomic variables for NSGA-II decision-making	Definition of the corresponding design domain	SIMP micro-optimization task	Performance feedback
Initial generation	$\theta = 50^\circ$, $A = 2.5 \times 10^{-3} \text{ m}^2$, $(x, y) = (0.5 \text{ L}, 0.5 \text{ H})$	Define a straight structural member with an orientation of 50°	Calculate the initial compliance in a homogeneous material ($\rho = 1$) across the entire design domain	$M_0 = 2600 \text{ kg}$, $U_0 = 14.2 \text{ m/N}$
Iteration #20	$\theta = 47^\circ$, $A = 2.1 \times 10^{-3} \text{ m}^2$	Fine-tune the member orientation and reference cross-sectional area	Perform SIMP iterations in the updated domain, where the material begins to aggregate toward higher strain energy regions	$M = 2200 \text{ kg}$, $U = 10.5 \text{ m/N}$
Iteration #68	$\theta = 45.2^\circ$, $A = 1.85 \times 10^{-3} \text{ m}^2$	Determine the final design domain: a straight structural member with a length of 3.0 m and an orientation of 45.2°	Complete the SIMP optimization in the final domain to obtain the optimized gradient density field	$M = 1860 \text{ kg}$, $U = 7.3 \text{ m/N}$

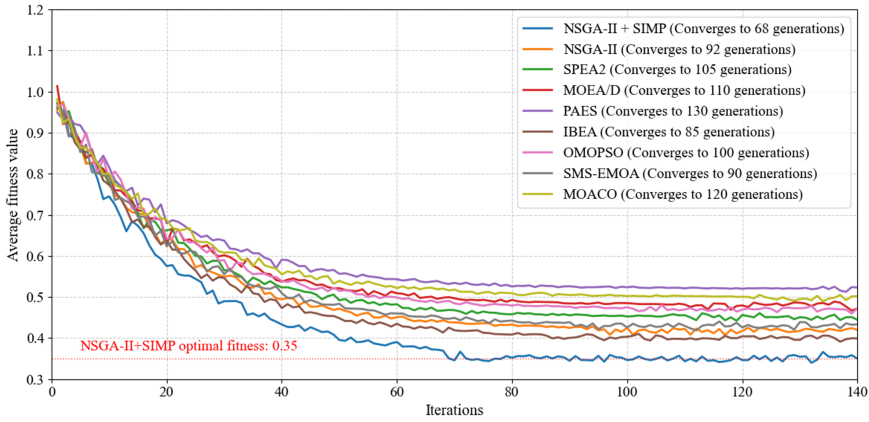


FIG. 7. Convergence curves. Note: PAES (Pareto archived evolution strategy); OMOPSO (optimized multi-objective particle swarm optimization).

the fastest initial convergence rate (the curve has the steepest slope at the beginning), which benefits from the collaborative optimization mechanism of macroscopic parameters and microscopic material distribution; among the different comparison algorithms, the indicator-based evolutionary algorithm (IBEA) converges faster (85 generations), but falls into a local optimum (final fitness value of 0.40).

The reasons why the NSGA-II+SIMP algorithm achieved the lowest fitness value (0.35) can be attributed to three factors. First, the two-level optimization architecture realizes a precise division of labor in the parameter space. NSGA-II is responsible for the global search for the macroscopic geometric parameters, while SIMP specializes in the optimization of the microscopic material distribution, thereby avoiding the efficiency loss of a single algorithm in multi-scale optimization. Second, SIMP's density filtering technology effectively suppresses the checkerboard effect, making the material distribution more consistent with the principles of structural mechanics. Finally, NSGA-II's elite retention strategy and non-dominated sorting mechanism ensure the continuous evolution of high-quality solutions. Combined with the sensitivity-based update of the OC method, a Pareto optimal solution with mass-stiffness balance is obtained in the 68th generation, which achieves a lower fitness than the other optimization algorithms.

5.4. ROBUSTNESS ANALYSIS UNDER DIFFERENT LOAD CONDITIONS

In example 1 (2D X-shaped central support), the robustness results of the structure optimized by the proposed algorithm under different load conditions are shown in Table 7.

TABLE 7. Structural response analysis under different load combinations.

Load case (consistent with Table 3)	Displacement [mm]	Maximum stress [MPa]	Compliance [m/N]
1	4.2	215	6.9
2	4.0	210	6.7
3	5.8	245	8.3
4	5.5	240	8.1
5	6.1	250	8.6
6	4.7	225	7.2
7	6.0	248	8.4
8	3.9	208	6.5
9	5.2	230	7.7
10	5.4	235	7.9

This study systematically analyzes the mechanical response of the optimized 2D X-shaped central support structure under ten typical load combinations. The results show that the structure exhibits good robustness and stability under various working conditions. From the perspective of the displacement response, the maximum nodal displacement is 6.1 mm (load case 5) and the minimum is 3.9 mm (load case 8). The displacement is within a reasonable range and meets the requirements of the limit state for normal use of the structure. The maximum stress values are between 208 MPa and 250 MPa, which are lower than the yield strength of Q345 steel (345 MPa), indicating that the structure has a sufficient strength reserve. The compliance values are between 6.5 m/N and 8.6 m/N, indicating that the stiffness of the optimized structure is stable under different loads, and no obvious stiffness degradation occurs. It is particularly noteworthy that, under combined load conditions (such as load cases 5 and 10), the structure still maintains good performance, indicating that the optimization model proposed in this paper has strong adaptability and robustness under multiple loading conditions. In general, the optimized structure exhibits good load-bearing performance and deformation resistance ability while ensuring a lightweight design, and thereby meeting the comprehensive requirements of engineering design for safety, economy, and applicability.

5.5. COMPARATIVE ANALYSIS OF SUPPORT CONFIGURATIONS

The optimization performance of five support configurations, namely X-type, V-type, inverted V-bracing, single diagonal brace, and K-type, is compared, and the results are shown in Fig. 8.

Figure 8 clearly reveals the structural efficiency and mechanical properties of the five support configurations – X-type, V-type, inverted V-bracing, sin-

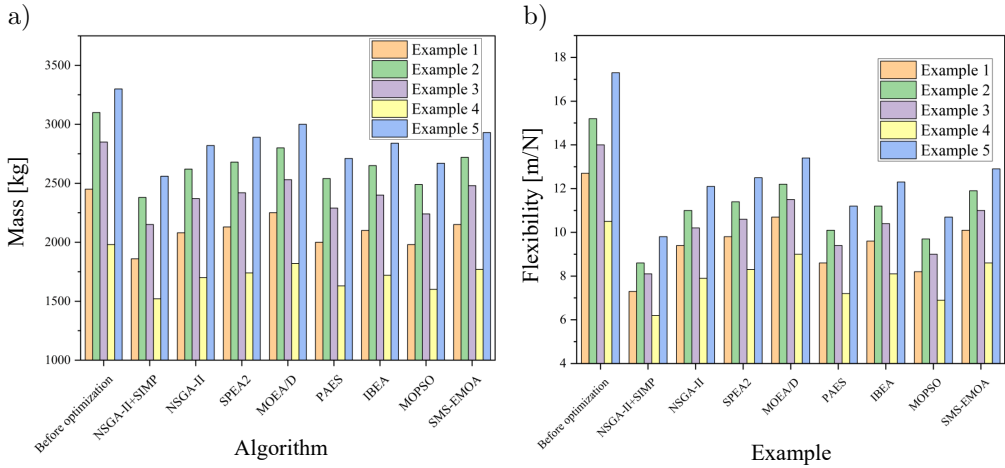


FIG. 8. Comparison of support configurations: a) mass comparison, b) compliance comparison.

gle diagonal brace, and K -type – by comparing their optimized mass (Fig. 8a) and compliance (Fig. 8b). The results show that the single-braced configuration performs best in terms of the lowest mass but has the lowest stiffness; the K -type support provides the highest stiffness but uses the most material; while the X -type support demonstrates a good overall balance between the conflicting objectives of mass and compliance. This comparison verifies the universal applicability of the proposed NSGA-II and SIMP collaborative optimization framework to different support configurations and provides a quantitative basis for engineers to select an optimal support configuration based on specific stiffness and weight requirements.

From the results of mass optimization we observe that the NSGA-II+SIMP collaborative optimization method proposed in this paper achieves the best lightweight performance among the five support configurations (X -type, V -type, inverted V -bracing, single diagonal brace, K -type). Taking the X -type support as an example, the mass after optimization is reduced from 2450 kg to 1860 kg, which is significantly better than the results from other multi-objective optimization algorithms (NSGA-II, SPEA2, MOEA/D, etc.). The optimized masses of the V -shape, inverted V -bracing, single diagonal brace, and K -shape supports are reduced to 2380 kg, 2150 kg, 1520 kg, and 2560 kg, respectively, all of which are better than the optimization results of the other comparison algorithms. This shows that the proposed method can effectively remove redundant materials and improve material utilization efficiency while ensuring structural integrity. Compared with the other algorithms, NSGA-II+SIMP achieves the coordinated optimization of macro-scale design parameters and micro-scale material distribution through its master-subordinate collaborative optimization mechanism, which has stronger global search capability and higher local op-

timization accuracy than stand-alone NSGA-II or other multi-objective algorithms (MOEA/D, OMOPSO). In addition, the fine control of material distribution provided by SIMP further improves the level of lightweight performance of the structure. Overall, NSGA-II+SIMP exhibits the best mass optimization performance in all support configurations, verifying its effectiveness and engineering applicability in the member-level optimization of steel structures.

The compliance optimization results show that the NSGA-II+SIMP method achieves the best improvement in structural stiffness in all five support configurations. Taking the *X*-type support as an example, the compliance is reduced from 12.7 m/N to 7.3 m/N. The optimized compliance of the *V*-type, inverted *V*-bracing, single diagonal brace, and *K*-type supports is 8.6 m/N, 8.1 m/N, 6.2 m/N, and 9.8 m/N, respectively, all of which are better than those obtained by mainstream multi-objective optimization methods such as NSGA-II, SPEA2, and MOEA/D. This shows that the proposed method is more effective in improving structural stiffness. The significant reduction in compliance indicates a smaller deformation response under the same loading conditions, thereby improving the overall load-bearing performance and lateral stiffness of the structure. NSGA-II+SIMP uses a master-subordinate collaborative optimization mechanism to search for the optimal geometric configuration in the macroscopic parameter domain while using SIMP to achieve optimization of the microscopic material distribution, thereby effectively improving load transfer efficiency. Compared with the other algorithms, the proposed method shows stronger convergence and stability in compliance optimization. NSGA-II+SIMP has obvious advantages in compliance control and provides a solid optimization framework for improving the performance of central support components of steel structures.

Besides the mass and compliance indices, the final support structures derived from different optimization methods exhibit significant differences in design morphology, primarily in the macroscopic mechanical parameters and material distribution. Regarding the macroscopic parameters, the NSGA-II+SIMP method proposed in this paper tends to select an optimal inclination angle of approximately 45° and a cross-sectional area of $1.85 \times 10^{-3} \text{ m}^2$ for *X*-shape supports. In contrast, methods such as stand-alone NSGA-II or MOEA/D, which lack integration with topology optimization, generally result in larger optimal inclination angles and cross-sectional areas, reflecting their shortcomings in exploiting the synergistic effect between macroscopic parameters and material utilization. In terms of material distribution, NSGA-II+SIMP generates a clear and efficient gradient density distribution through the SIMP sub-optimizer, concentrating material along the principal stress paths. In contrast, methods without integrated topology optimization only yield members with a homogenous material distribution, resulting in lower material utilization. Although algorithms such

as SPEA2 can approximate similar macroscopic parameters, their solution sets lack the physical guidance provided by SIMP regarding material distribution details, leading to slightly lower mechanical efficiency in the final configuration. These design differences confirm that only through synergistic optimization at both the macro- and micro-levels higher-performance structural configurations can be created while controlling the amount of materials used. This is the core improvement in the structural design introduced by the method presented in this paper.

5.6. COMPARISON WITH SINGLE-OBJECTIVE OPTIMIZATION

In example 1 (2D *X*-shaped center support), the proposed algorithm is compared with several single-objective optimization algorithms, including the genetic algorithm (GA), particle swarm optimization (PSO), simulated annealing (SA), and gradient descent (GD). The single-objective optimization takes the minimum mass as the objective, and the optimization results are compared as shown in Fig. 9.

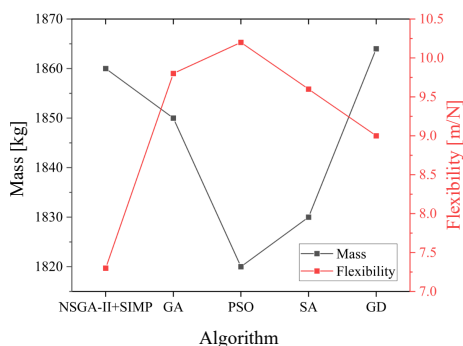


FIG. 9. Comparison with single-objective optimization.

The NSGA-II+SIMP collaborative optimization method proposed in this paper shows a comprehensive performance that is superior to the traditional single-objective optimization algorithm for the parameter configuration optimization of the 2D *X*-shaped central support component. From the perspective of the mass index, the optimized mass obtained by NSGA-II+SIMP is 1860 kg, which is slightly higher than the one obtained by GA (1850 kg), PSO (1820 kg), and SA (1830 kg), but lower than the one obtained by GD (1864 kg), indicating that the presented method is still highly competitive in terms of lightweight design. However, the performance of the proposed method in terms of compliance index is significantly better than that of all the single-objective optimization algorithms: its compliance value is 7.3 m/N, which is lower than 9.8 m/N for GA, 10.2 m/N for PSO, 9.6 m/N for SA, and 9.0 m/N for GD, indicating that the optimized structure has a smaller deformation response under the same loading condi-

tions and better overall stiffness performance. This trend of slightly increasing mass accompanied by significantly reducing compliance reflects the reasonable trade-off between structural performance and material economy achieved by multi-objective optimization in engineering design.

NSGA-II+SIMP employs a master-subordinate collaborative mechanism to optimize macroscopic parameters by NSGA-II while SIMP achieves local optimization of the material distribution, thereby improving structural stiffness and load-bearing efficiency. In contrast, although single-objective optimization algorithms can reduce the structural mass to some extent, they often ignore the simultaneous optimization of structural stiffness, resulting in a decrease in the structural deformation resistance. Therefore, while maintaining the advantages of a lightweight structure, the proposed method significantly improves the mechanical performance, demonstrates stronger optimization robustness and engineering applicability, and provides a more practical optimization strategy for the intelligent design of central support components in steel structures.

To provide a more indicative comparison, this study introduces a weighted single-objective optimization method as a benchmark. This method transforms the multi-objective optimization problem into a single-objective problem using a weighting coefficient. Its objective function is defined as

$$\text{Minimize } f(\mathbf{x}) = \alpha \cdot \frac{M(\mathbf{x})}{M_0} + (1 - \alpha) \cdot \frac{U(\mathbf{x})}{U_0},$$

were M_0 and U_0 are the initial design mass and compliance, respectively, and are used for normalization. By systematically adjusting α from 0 to 1, a series of solutions from pure compliance minimization to pure mass minimization can be obtained.

Table 8 lists the key results obtained by applying this method to the 2D X-shaped central support (example 1), and compares them with three represen-

TABLE 8. Comparison between weighted single-objective optimization and multi-objective optimization results.

Method	Weight α	Mass [kg]	Compliance [m/N]	Notes
Weighted single-objective optimization	0.0	2180	6.9	Pure compliance minimization
	0.3	2020	7.5	Compliance-oriented solution
	0.5	1920	8.1	Balanced weighting
	0.7	1840	9.2	Mass-oriented solution
	1.0	1780	11.5	Pure mass minimization
NSGA-II+SIMP	–	1860	7.3	Balanced solution (this study)
	–	1750	8.8	Lightweight solution
	–	2050	6.8	High-stiffness solution

tative Pareto-optimal solutions selected by the NSGA-II+SIMP method in this paper.

Analysis of Table 8 shows that the weighted single-objective optimization can generate solutions diagonally distributed along the mass-compliance space by adjusting α . When $\alpha = 1.0$, the mass is the lowest (1780 kg), but the compliance is as high as 11.5 m/N, making the structure too flexible; when $\alpha = 0.0$, the compliance is the lowest (6.9 m/N), but the mass is relatively large (2180 kg). The main limitations of this method are: 1) the decision-maker must pre-set a fixed α value, which requires prior knowledge of design preferences and cannot be weighed afterward; 2) only one solution can be obtained per run, and obtaining multiple balanced solutions requires multiple independent calculations, resulting in low computational efficiency.

In contrast, the NSGA-II+SIMP multi-objective optimization method proposed in this paper can generate a complete Pareto front covering a wide range of mass/compliance trade-offs in a single run (as shown by the three representative solutions listed in Table 8). This provides decision-makers with a rich selection space, allowing them to flexibly choose the most suitable solution based on specific engineering constraints and performance requirements (e.g., choosing a balance solution with a mass of 1860 kg and a compliance of 7.3 m/N). Therefore, the proposed method significantly outperforms weighted single-objective optimization methods in providing decision-making flexibility and a global perspective.

5.7. ANALYSIS OF THE MACROSCOPIC MECHANICAL PARAMETER OPTIMIZATION RESULTS

To further reveal the structural optimization mechanism of different support forms, Table 9 lists the key macroscopic mechanical parameters corresponding to a representative Pareto-optimal solution (considering both mass and stiffness performance) for the five support structures obtained through the NSGA-II algorithm.

The analysis shows that there are systematic differences in the optimal macroscopic parameters for the different configurations of the support, which are closely related to their unique mechanical mechanisms. The optimal inclination angle of the *X*-shaped support is close to 45° , which is in line with its pursuit of bidirectional stiffness balance. The optimal inclination angle of the *K*-type support is the highest (56.1°), which helps to improve its out-of-plane stability and optimize load-transfer paths. From the perspective of the cross-sectional area, the higher the stiffness requirements for the support form (such as *K*-type and *V*-type), the larger the optimal cross section. The results of the arrangement position reveal the regions where the brace plays the most effective role in the frame, for example, the single diagonal brace can more effectively resist the

TABLE 9. Comparison of the optimal macroscopic parameters for different support configurations.

Support configuration	Optimal inclination angle θ [°]	Optimal cross-sectional area A ($\times 10^{-3} \text{ m}^2$)	Relative layout position (x, y)	Main mechanical characteristics
X -type	45.2	1.85	(0.5 L, 0.5 H)	Bidirectional stiffness balance, clear load-transfer paths, and significant stress concentration in the joint region
V -type	51.8	2.15	(0.4 L, 0.6 H)	Effectively suppresses buckling, the diagonal braces are mainly subjected to axial tension and require high-strength connecting for nodes
Inverted V -bracing	48.5	1.95	(0.45 L, 0.55 H)	Mechanical performance is between the X -type and V -type, providing moderate lateral stiffness and redundancy
Single diagonal brace	41.3	1.52	(0.3 L, 0.7 H)	The structure is simple, the mass is the lightest, but the stiffness is relatively low, and it is a unidirectional load-resisting system
K -type	56.1	2.38	(0.35 L, 0.65 H)	The maximum stiffness can effectively reduce inter-story displacement, but the joint structure is complex and prone to stress concentration

torque when it is asymmetrically arranged (0.3 L). These optimal parameter sets provide direct data support and a theoretical basis for support selection and refined design under different engineering requirements.

6. CONCLUSION

This study successfully developed and validated a master-subordinate nested collaborative optimization framework integrating the NSGA-II genetic algorithm and the SIMP topology optimization method for the multi-objective design of central support components in steel structures. The primary achievement lies in the effective realization of simultaneous optimization at both the macro- (geometric parameters) and micro- (material distribution) scales, thereby addressing the inherent conflict between minimizing structural mass and compliance. The key findings demonstrate that the proposed method reduces structural mass by up to 24.1% while significantly enhancing stiffness, as evidenced by a substantial decrease in compliance across various support configurations. Furthermore, the generation of a high-quality Pareto front ($HV = 0.874$) provides engineers with a diverse set of optimal trade-off solutions, facilitating informed decision-making based on specific project requirements.

The engineering significance of this work is profound. The proposed method transforms support design from an experience-dependent, iterative processes into a systematic, performance-driven approach. The framework enables the creation of lighter, stiffer, and more material-efficient support systems, directly contributing to cost reduction, improved seismic performance, and the advancement of sustainable building practices by minimizing material usage.

Despite its promising results, this study acknowledges certain limitations. The computational expense associated with the nested optimization framework remains non-trivial for very large-scale structures or highly refined meshes. Furthermore, the model currently assumes linear elastic material behavior and idealized connections, which may not fully capture nonlinear inelastic response under extreme loading conditions.

Future research will focus on several key areas:

1. Enhancing computational efficiency through surrogate model-assisted optimization strategies.
2. Incorporating material and geometric nonlinearities to improve the predictive accuracy for ultimate limit states.
3. Extending the framework to encompass system-level optimization of entire steel frames, including the interactive design of beams, columns, and connections.

These steps will further strengthen the framework's role in enabling intelligent, high-performance structural design.

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COMPETING INTERESTS

The authors declare that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHORS’ CONTRIBUTIONS

All authors contributed equally, viewed and approved the final manuscript.

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Research Paper

Machine Learning-Based Welding Defect Recognition Using GLCM Features and k -Fold Cross-Validation: KNN and SVM Techniques

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Although radiographic inspection is one of the oldest techniques for non-destructive testing, it is still considered vital in many industrial fields to ensure the quality of welds and meet the demands of work conditions and design, as well as safety and reliability requirements. This paper presents an algorithm that identifies and categorizes welding defects in radiographic images using machine learning techniques. For this aim, two supervised classifiers are proposed and conducted: 1) K-nearest neighbor (KNN), which is a nonparametric classifier, and 2) a multiclass classifier based on a support vector machine (SVM) as a highly generalized learning-based classifier. SVM is commonly used in binary classification, but it can be adapted for multi-classification using various common methods, such as one-versus-one and one-against-all. The texture features are adopted in this paper as inputs to the classifiers, where two groups of them are used: the local binary pattern (LBP) features and the gray-level co-occurrence matrix (GLCM) extraction, to obtain the feature vector. To avoid the risk of overfitting, a 4-fold cross-validation is applied. The experimental results are reported for the two different classifiers, achieving an accuracy of 91.66% when combining GLCM and SVM.

Keywords: welding defect, radiographic images, classification, machine learning, k -fold cross-validation.



1. INTRODUCTION

1.1. BACKGROUND

To ensure the quality of welding, the traditional approach in welding environments involves using non-destructive testing (NDT) and post-weld techniques for evaluation. In welding fields, the main non-destructive tests can be categorized as: visual inspection (VT) [1], ultrasonic testing (UT), radiographic testing (RT) [2], eddy current testing (ECT) [3], magnetic particle testing (MT) [4], liquid penetrant testing (PT) [5], acoustic emission testing (AE) [6], infrared thermography (IRT) [7], stereography [8], and computed tomography (CT) [9]. X-ray as a radiography test can be used on a variety of materials with varying densities, as it can penetrate a wide range of materials to detect internal faults in welds [10]. Defective regions absorb greater amounts of energy, which results in a dimmer appearance. Among the most common flaws that are possible to detect in the images of radiographic inspection are porosity (the defect that arises from the entrapment of gas), lack of fusion (the defect resulting from a lack of fusion between the base metal and the weld metal), cracks (linear discontinuities caused by the fracture of metal), solid inclusions (the defect that arises from foreign matter trapped during welding), burn-through (excessive penetration of the weld metal at the root zone of the base metal due to excessive heat), and lack of penetration (weld penetration that is less than specified, usually occurring at the root of the weld) [11–13].

1.2. RELATED WORK

The detection and classification of welding flaws are crucial in real-life and industrial environments. Furthermore, this field is considered one of the most important research topics that has been investigated. Liao and Ni [14] are considered pioneers in the use of digital radiographs to identify weld defects by comparing pixel intensities with a Gaussian pattern in the weld region. They demonstrated that similarity can be measured by the mean square error (MSE), with the lowest MSE indicating a weld line. A method was subsequently developed that focuses on the geometric features of flaws to improve the accuracy of welding defect detection using linear classifiers. It was found that having high-quality features is more important than simply having a large number of features for reliable classification [15]. TUO *et al.* [16] achieved higher accuracy when using the Gabor filter and the gray-level co-occurrence matrix (GLCM) in welding defect classification. They applied principal component analysis (PCA) to minimize the feature vector dimensions and used K-nearest neighbors (KNN) as a comparison classifier. In the last twenty years, researchers have turned more and more to the GLCM as a practical way to study weld images [17–19]. By extracting texture features, this approach has helped make defect detec-

tion faster and more accurate, while reducing the need for manual inspection. A Gabor filter can be used to characterize texture features in an automatic system for detecting weld discontinuities, as feature extraction focuses primarily on calculating the properties of localized areas. Consequently, the GLCM displays the number of occurrences of a specific gray value in relation to another gray value in the image specific spatial context specified by the user [20]. The joint application of the Gabor filter and GLCM was later extended in welding defect classification by applying support vector machine (SVM) algorithms and evaluating performance through the receiver operating characteristic (ROC) curve [21]. This modification enabled the study of feature morphology, such as the size and shape of defects.

Artificial neural networks (ANNs) have made an influential contribution to the diagnosis of weld defects based on digital radiographic testing images, and their results have shown increasing reliability and promising validity [22]. ANNs have become increasingly popular in conjunction with other classification methods, such as KNN and SVM, to differentiate welding flaws in images from radiographic testing. Both morphological and texture features are adopted to implement the segmentation process and generate the feature vector in the classification procedure [23]. Further modifications using deep ANNs to classify the weld defects in radiographic testing images yielded accuracies of 75.83% to 98.75% [24]. This algorithm enables the classification of defects into more categories, including non-penetration, crack, pore, and no-defect status. Later, by applying deep learning algorithm pipelines for welding defects, the defect categories were extended to include seven flaws: porosity, undercut, crack, slag inclusion, lack of penetration, lack of fusion, and interior concavity [25].

1.3. MAIN CONTRIBUTIONS AND OUTLINE

Based on the above discussion, further investigation into the detection and classification of weld flaws from images is warranted. Using machine learning and new advanced methods is worthy of study and evaluation. Furthermore, obtaining very high performance (accuracy, precision, and recall) for the model is the top priority.

The main contribution of this study is presented in the following points:

- an advanced algorithm that employs machine learning is proposed to detect and classify weld flaws from images with very high performance (accuracy, precision, and recall),
- the proposed algorithm involves capturing images with an ordinary digital camera, enhancing them to remove noise using a common filter, such as the adaptive median filter, and basing the classification on texture descriptors, as it has shown very high effectiveness,

- this is accomplished more specifically by utilizing the feature descriptors of the local binary pattern (LBP) and GLCM, after which the classification process commences,
- for classifiers, their selection is made using two machine learning methods: a multiclass classification based on SVM and KNN. Thus, it is possible to eliminate the need for specialized equipment or skilled workers who have experience and knowledge in the field of inspection.

The remainder of this paper is outlined as follows. [Section 2](#) presents the proposed methodology, including the methods, the collected dataset, the classification process, and the evaluation metrics. In [Sec. 3](#), the experimental results are presented and discussed. In [Sec. 4](#), the paper's main points are summarized, and some potential future work is highlighted.

2. MATERIALS AND METHODS

2.1. DATASETS PREPARATION

Film digitization is an important step in systems used to determine welding defects. As a result, the most important factors for the performance of the above-mentioned system are the selection of an optimized scanning resolution and an acceptable level of digitization quality. Radiographic film digitization is carried out using several systems based on American Society of Mechanical Engineers (ASME) standards. The most common method is scanning, which uses light transmission (transparency adapters). The dataset images were captured with a Nikon D7000 digital camera, tuned to achieve optimal contrast. Indeed, the Nikon D7000 camera can be used for radiography using a technique called indirect digital radiography. This entails taking images of an X-ray-exposed fluorescent screen using the camera. The camera was set up at a location with a 400 mm lens, without a flash. Dataset images were prepared by converting them from RGB to grayscale and automatically cropping them. This step highlighted the welding area and a limited surrounding area. The symbols, numbers, and markings that are often associated with welding films were eliminated. Next, preprocessing models and approaches were used to minimize noise and enhance contrast, allowing the main objects in the image to be distinguished from the background, making them more discernible. Once segmentation was completed, the two feature texture extractors (LBP and GLCM) were performed to extract features that are highly distinctive for the classifiers. An automated system for identifying, recognizing, and classifying the welding defects was developed using two proposed classifiers: 1) error-correcting output coding (ECOC) with an SVM, and 2) KNN. The features and weld fault identification were obtained from these two proposed classifiers. This proposed methodology is presented in [Fig. 1](#).

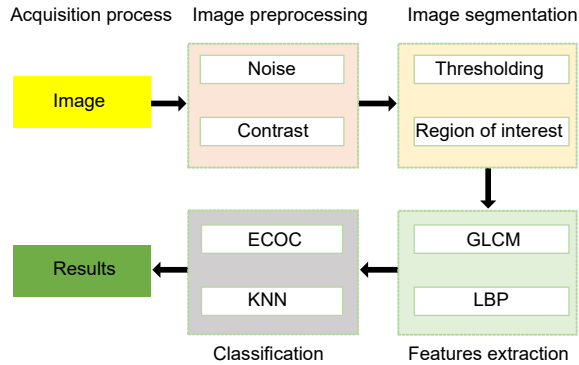


FIG. 1. Procedure for classifying welding defects (ECOC is an SVM learner).

The total number of images in the dataset is 72. The images are used to cover four types of defects, which are distributed among burn-through (BT), solid inclusions (SI), concavity (CO), and incomplete penetration (IP), as shown in Table 1. Experts in the field of radiographic interpretation established the ground truth knowledge for these images. Sample images containing BT are presented in Fig. 2.

TABLE 1. Dataset of welding defects radiographic images.

Welding defect	Number of images
SI	25
BT	17
IP	22
CO	8

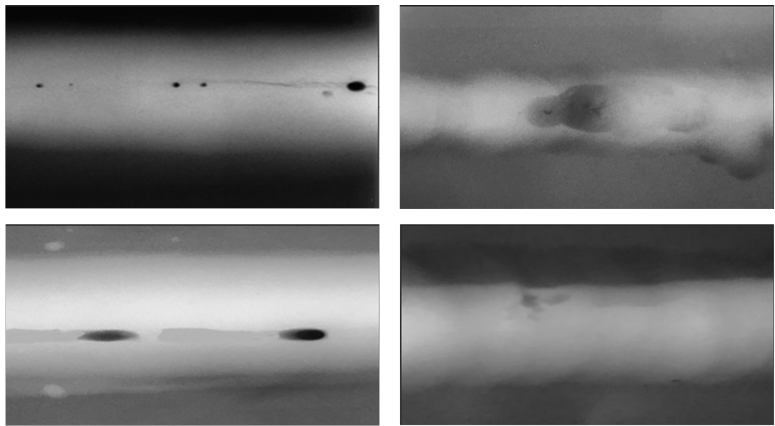


FIG. 2. Sample images containing BT. This figure contains the images from our dataset.

2.2. IMAGE PREPROCESSING

As an imaging system, the nature of NDT image is affected by an arbitrary signal known as noise. This signal always appears during the process of image capture, which includes coding, image transmission, and processing steps [26].

The presence of this noise could distort the primary data of the radiographic images. The source of noise in digital images can cause problems and is often considered a defect in the data acquisition process. The filtering process is a crucial and essential step in image preprocessing [27, 28]. Additionally, this process can improve image quality. The primary function of the image filtering process is to remove degradation and noise from the image, thereby achieving a high-quality image.

Image contrast enhancement refers to a type of image manipulation that modifies and improves an image appearance [29, 30]. It can be defined as the dynamic range measure, which represents the entire range of intensity values found in the image. In another definition, it is the difference between the maximum and minimum pixel values. A histogram is an essential tool to determine whether an image has a large or small dynamic range. Therefore, for high contrast, the histogram should be extended to encompass the entire dynamic range. Examples of radiographic images and image brightness improvement using different methods are presented in Fig. 3. The left side indicates the original images of IP that were contrast-stretched, then globally and next locally equalized, while the right side presents the histograms of the corresponding cases. All these improvement methods are applied to the images to find the one that provides the highest performance. The best performance is obtained by the contrast-stretching method.

2.3. IMAGE SEGMENTATION

Segmentation can be described as dividing images into parts based on specified rules. These parts or areas can be considered homogeneous in certain visual aspects, such as color or density [31, 32]. The most effective and efficient technique for segmenting and classifying images of welding inspection films is the thresholding technique [33, 34]. An example of the segmentation process using different thresholding techniques is presented in Fig. 4. The original image (Fig. 4a) was processed using adaptive thresholding (Fig. 4b), then global thresholding (Fig. 4c), and histogram thresholding (Fig. 4d). All these segmentations methods are applied to the image. The best performance is obtained by adaptive thresholding.

2.4. GLCM

This method is a statistical approach to analyzing gray-level values by examining their spatial distribution in images, which effectively identifies distin-

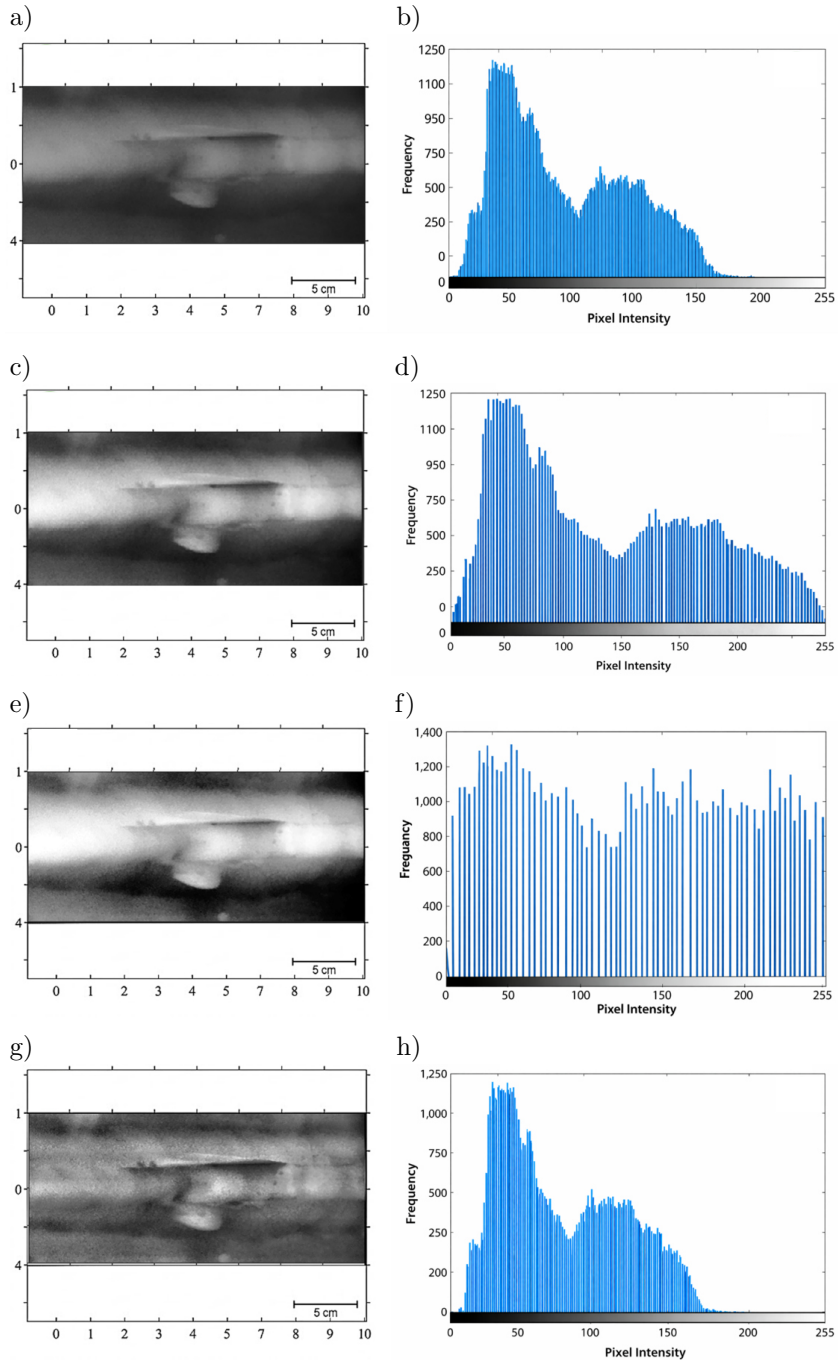


FIG. 3. Various improvement techniques carried out on an image with IP along with their corresponding histograms: a) original image, b) original image histogram, c) global equalization image, d) global histogram equalization, e) local equalization image, f) contrast limited adaptive histogram equalization (CLAHE). This figure uses the images from our dataset.

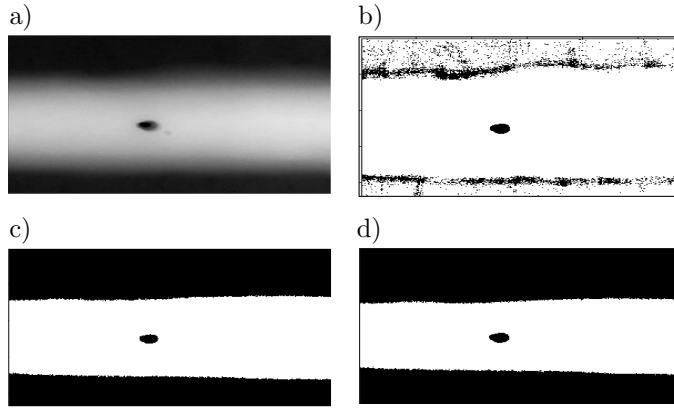


FIG. 4. Segmentation process by different thresholding techniques: a) original image, b) adaptive thresholding, c) global thresholding, d) histogram thresholding. This figure uses the images from our dataset.

guishing characteristics that can be used to find texture-related features [35–37]. The method can be summarized by creating a matrix that aggregates the total number of pixel pairs in the image. It represents the evaluation of the frequency of two gray levels within a specified region. GLCM reveals a specific relationship of several times in which a transition occurs between adjacent pixels (reflecting the correlation between two adjacent pixels within a local region). The creation of the matrices depends on both the distance and the direction. It shows that the reference pixel of intensity i has special relevance with the adjacent pixel of intensity j , which is separated by a spatial distance d and a direction θ . There are many ways for determining the spatial relationship between two neighboring pixels, considering different angles and displacements. This is clear in Fig. 5. After the matrix has been created, many statistics are derived and can be applied to the GLCM. As the number of statistics reaches 14,

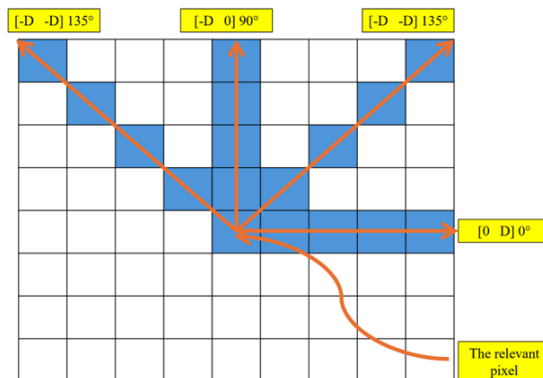


FIG. 5. Different co-occurrence matrix offsets to extract texture features.

statistical tasks are characterized by accuracy. The most widely used ones are variance, energy, homogeneity, and correlation on a large scale [38].

2.5. LBP

This factor determines the texture of an image and is based on the local differences between neighboring pixel values. In texture analysis, this factor presents a significant leap forward and is preferred over other previous techniques in various applications, as it has undergone substantial development and progress [39–41]. LBP is a creative tool for texture classification [42]. The main concept of this descriptor relies on two scales: local patterns and gray-level contrast. A local binary code is extracted for each pixel found in the image by assigning the neighboring pixels with the value of the center pixel, which shows the threshold.

This process characterizes the neighborhood by multiplying the threshold values by the corresponding pixel-specific weight. Therefore, a histogram is created to accumulate the frequencies of the different binary patterns. The basic operator model relies on the eight neighboring pixels around the pixel in question [43]. The texture details of the LBP are obtained originally from a 3×3 window by the use of the gray value of the center pixel as a threshold for its neighbors. Figure 6 shows the basic LBP operator.

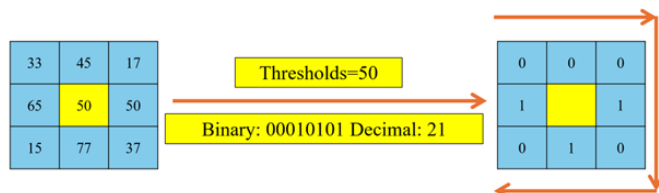


FIG. 6. Basic LBP operator.

2.6. CLASSIFICATION

Classification is the ability to create a model or an approach that can identify different categories in a dataset. Based on the results obtained from the training process, many outputs can be predicted. Moreover, there is a possibility of classifying these predicted data. In this study, the KNN algorithm and the SVM algorithm were used as classifiers during testing, both of which are well-suited for handling non-linear problems [44–46].

2.7. MULTICLASS SVM CLASSIFICATION

Various methods have been proposed to utilize SVMs for handling multi-class problems, as they can be extended by merging multiple binary types

into a multi-class classifier [47]. For present multi-class classification schemes, the one-against-all model (OAA) is the most commonly used. Classification functions are built between each category relative to the remaining categories. An M-class problem is modified into M binary classification problems. The learning step of the classifiers uses the entire training data, on the basis that the data vectors from a certain category are labelled positive and all other samples are labelled negative. In the testing step, classification is performed by calculating the margin from the linear separating hyperplane. The class with the maximum decision value is the final output. A merit of this method is that the number of binary classifiers to be constructed is equal to the number of categories. According to the maximum output value of the SVM classifiers, the label of an instance is predicted. One drawback of the OAA approach is that the dataset is significantly unbalanced after collecting the M-1 categories [48–50].

Another approach is to build a group of one-versus-one (OVO) classifiers, where the imbalance is less pronounced in the dataset than in the previous method. This is achieved by separating the M-class dataset problem into $M(M-1)/2$ binary classification problems and fitting a binary classification model on each. This method appears to be overly complex, as the number of classifiers obtained is often too large, making the model too intricate. Consequently, this model is not suitable for solving problems that contain a large number of categories. However, if the number of categories is small, each binary classifier is trained on a smaller subset of data. As a result, the training and tuning process of the classifier becomes more efficient and faster. To determine the class of an unknown sample, the obtained $M(M-1)/2$ binary classifiers are used, and the class containing the most common object samples is predicted through a voting method [48].

2.8. EVALUATION METRICS

To assess the quality of the classifier, a confusion matrix is used, which provides a visual representation of its predictive power. This matrix, as shown in Fig. 7 [51], includes the following classes: true positive, true negative, false positive, and false negative. These classes are defined as follows:

- true-positive (TP) means the classifier predicts positive and it is positive in fact,
- true-negative (TN) means the classifier predicts negative and it is negative in fact,
- false-positive (FP) means the classifier predicts positive and it is negative in fact,
- false-negative (FN) means the classifier predicts negative, and it is positive in fact.

		Predicted class		
		Positive	Negative	
Actual class	Positive	True positive (TP)	False negative (FN) Type II error	Sensitivity $\frac{TP}{(TP + FN)}$
	Negative	False positive (FP) Type I error	True negative (TN)	Sensitivity $\frac{TN}{(TN + FP)}$
		Precision $\frac{TP}{(TP + FP)}$	Negative predictive value $\frac{TN}{(TN + FN)}$	Accuracy $\frac{TP + TN}{(TP + TN + FP + FN)}$

FIG. 7. Confusion matrix structure [51].

The main measures presented in the confusion matrix are discussed. Accuracy measures the closeness of the quantity to the actual/real value. It is considered a very effective metric if a relatively symmetric dataset is found [52, 53]:

$$(2.1) \quad \text{Accuracy} = \frac{\text{Correct prediction}}{\text{Total dataset}} = \frac{TP + TN}{TP + TN + FP + FN}.$$

Measuring the percentage of correct predictions made by the proposed model is precision. It is defined by dividing the number of positive examples that are correctly classified by all the examples classified as positive by the classifier. The following form can represent it:

$$(2.2) \quad \text{Precision} = \frac{TP}{TP + FP}.$$

The third metric is recall, which indicates the percentage of all potential ground truth boxes identified. It is expressed as the number of correctly categorized positive examples divided by the total number of positive examples in the data [52]:

$$(2.3) \quad \text{Recall} = \frac{TP}{TP + FN}.$$

The experimental results from applying the proposed classifiers are presented in the next section.

3. EXPERIMENTAL RESULTS AND DISCUSSION

Different settings for the GLCM are explored to identify the features most suitable for the classifier by conducting multiple tests that vary both distance and

direction. Four directions are considered in the current work: 0° , 45° , 90° , and 135° , with the spatial pixel distance parameter set to 2. The settings for both direction and distance used for constructing GLCM are presented in Table 2.

TABLE 2. Setting of distance and direction for GLCM construction.

Parameter	Setting				
Item	1	2	3	4	5
Direction and angle	[0 2] (East)	[-2 2] (Northeast)	[-2 0] (North)	[-2 -2] (Northwest)	Combination of items (1 and 2)
Number of features	10	10	10	10	$10 + 10 = 20$
Features	Angular second moment, energy, entropy, contrast, maximum probability, dissimilarity, homogeneity, GLCM mean, variance and correlation				

The interpretation of all GLCM features incorporated in the current work is presented in Table 3.

TABLE 3. Description and visualization of all GLCM features and offsets incorporated in the current work.

Offset	Distance (d) in pixels	Direction (θ)	Description
GLCM [0 1]	1	0° (horizontal)	Neighbor is 1 pixel to the right
GLCM [0 2]	2	0° (horizontal)	Neighbor is 2 pixels to the right
GLCM [-1 1]	$\sqrt{2}$	45° (diagonal)	Neighbor is 1 pixel up and 1 pixel right
GLCM [-2 2]	$2\sqrt{2}$	45° (diagonal)	Neighbor is 2 pixels up and 2 pixels right
GLCM [-1 0]	1	90° (vertical)	Neighbor is 1 pixel up
GLCM [-2 0]	2	90° (vertical)	Neighbor is 2 pixels up
GLCM [-1 -1]	$\sqrt{2}$	135° (anti-diagonal)	Neighbor is 1 pixel up and 1 pixel left
GLCM [-2 -2]	$2\sqrt{2}$	135° (anti-diagonal)	Neighbor is 2 pixels up and 2 pixels left

It should be noted that the GLCM offsets GLCM [0 2], GLCM [-2 2], GLCM [-2 0], and GLCM [-2 -2] are used for SVM and KNN. However, the GLCM offsets GLCM [0 1], GLCM [-1 1], GLCM [-1 0], and GLCM [-1 -1] are used only for SVM.

As for the parameters selected for the uniform rotation-invariant LBP, which is adopted in this investigation, are eight neighbours ($P = 8$) and their relationship to the central pixel. Therefore, the number of bins is 59, following

$P(P - 1) + 3$. Additionally, this method is used to achieve rotation stability by combining identical uniform patterns from different directions into a single bin. In contrast, the remaining patterns are placed in separate bins, resulting in a total of 10 distinct output bins ($P + 2$). The settings for the LBP features are presented in Table 4.

TABLE 4. Settings for the LBP features.

Parameter	Setting		
Item	1	2	3
Number of neighbors (P)	8	8	8
Rotation invariant	Yes ($P + 2$)	No ($P(P - 1) + 3$)	Custom normalization
Result	10 features	59 features	4248 features

For the multiclass classification using SVM, the OAA and OAO approaches with a linear kernel are adopted. Concerning the KNN classifier, Euclidean distance is applied for measuring the distance between the test point and all reference points to obtain the KNN, and the number of neighbours is set as $k = 1$. To select the most preferable combination of classifiers and feature extraction methods, the results are interpreted. The total accuracy, which shows the fraction of cases that are classified correctly, as well as precision and recall, are evaluated to support this selection.

Table 5 and Table 6 summarize the obtained results of the experimental evaluation, where the performance of each group of the classifier and feature extraction methods is evaluated using k -fold cross-validation as the error estimator. In the current work, $k = 4$, which means that the model is trained and evaluated four times repeatedly by randomly assigning four equal-sized subgroups to satisfy the validation requirements. In each attempt, the training of the system is performed using three of the subsets, whereas the remaining subset is used for evaluation or testing.

TABLE 5. Evaluation results of classification based on multiclass SVM.

Multiclass SVM classifier	Accuracy [%]	Precision [%]	Recall [%]
GLCM $[-2 \ 2]$	90.00	93.75	93.75
GLCM $[0 \ 2]$	91.66	95.83	91.66
GLCM $[-2 \ 0]$	90.90	93.75	95.00
GLCM $[-2 \ -2]$	90.00	93.75	93.75
GLCM $\{[-2 \ 2], [0 \ 2]\}$	90.00	93.75	93.75
LBP 59 features	87.50	91.66	91.66
LBP 10 features	83.33	86.66	86.66

TABLE 6. Evaluation results of classification based on KNN.

KNN classifier	Accuracy [%]	Precision [%]	Recall [%]
GLCM $[-2\ 2]$	90.00	93.75	87.50
GLCM $[0\ 2]$	87.50	91.66	91.66
GLCM $[-2\ 0]$	90.00	93.75	87.50
GLCM $[-2\ -2]$	90.00	93.75	87.50
GLCM $\{[-2\ 2], [0\ 2]\}$	87.50	91.66	91.66
LBP 59 features	87.50	91.66	91.66
LBP 10 features	75.00	75.00	79.17

The accuracy of the SVM classifier is higher compared with that of the KNN. Detailed results obtained by the various combinations of classifiers and features are presented in Table 7 and Table 8. Table 7 presents the value of the highest accuracy obtained when the GLCM technique is used to extract the features. Evaluating the 2-pixel neighbourhood to the east $[0\ 2]$ on images leads to an accuracy value of 91.66 %. Examining the neighborhood two pixels away in all directions yields high-accuracy results. Generally, features extracted the by the use of GLCM show satisfactory results. With five tests, the achieved accuracy is 90 % or higher, and the obtained precision and recall are also 90 % or above.

TABLE 7. SVM multiclass classification results using GLCM features.

Multiclass SVM classifier	Accuracy [%]	Precision [%]	Recall [%]
GLCM $[-2\ 2]$	90.00	93.75	93.75
GLCM $[-1\ 1]$	83.33	92.85	91.66
GLCM $[0\ 2]$	91.66	95.83	91.66
GLCM $[0\ 1]$	88.88	93.75	87.5
GLCM $[-2\ -2]$	90.00	93.75	93.75
GLCM $[-1\ -1]$	84.00	90.83	93.05
GLCM $[-2\ 0]$	90.90	93.75	95.00
GLCM $[-1\ 0]$	83.33	92.85	83.3
GLCM $\{[-2\ 2], [0\ 2]\}$	90.00	93.75	93.75
GLCM $\{[-1\ 1], [0\ 1]\}$	87.50	90.66	87.5

Combining both multiclass classifiers based on SVM and LBP features (as presented in Table 8) results in high accuracy when the rotation invariance of features is not considered. Thus, it can be concluded that rotation invariance reduces accuracy, as it causes the loss of certain informative features. Therefore, the use of rotation-invariant features yield a lower accuracy of approximately 83.33 % compared with 87.50 % when omitting rotation invariance. In the lat-

TABLE 8. SVM multiclass classification results using LBP features.

Multiclass SVM classifier	Accuracy [%]	Precision [%]	Recall [%]
LBP 59 features	87.50	91.66	91.66
LBP 10 features	83.33	86.66	86.66

ter case, the classifier demonstrates strong robustness, achieving a recall and precision of 91.66 %.

Regarding the KNN classifier, several preliminary tests were conducted to determine the optimal value for the neighbourhood dimension. The conclusion reached is that the optimal configuration is $k = 1$. However, the best results with this classifier were inferior to those obtained by the SVM classifier. The combination of the KNN classifier and GLCM features yields the best results with an accuracy of 90 %.

4. CONCLUSION AND FUTURE WORK

In this paper, two machine learning approaches were proposed for detecting and classifying welding defects. These approaches are SVM and KNN. By using the dataset for training and testing strategies, the SVM showed higher performance results compared to the KNN classifier. However, the KNN classifier relies on matching the feature profiles between the training and testing datasets. High similarity that exists among the images of welding inspection leads to the close grouping of many extracted feature values, which is a main reason for the lower accuracy of the KNN classifier. Both GLCM and LBP techniques were used for extracting the features. When using the LBP technique, the displacement value is based on the closely adjacent neighbor (pixel). When using the GLCM technique, the displacement of two pixels adopted. Future work can be outlined as follows: 1) new software tools and classification methods can be used and assessed, 2) the current work is restricted to the spatial domain; therefore, the frequency domain can be explored, 3) investigating new and advanced supervised/unsupervised classifiers is recommended, 4) a new technique for extracting features is recommended for investigation and assessment, 5) the study can be expanded to include all types of welding defects. Moreover, expanding the dataset to include a larger number of images and a wider variety of welding defect types would improve model generalization and classification reliability.

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CONFLICTS OF INTEREST

The authors declare that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHORS' CONTRIBUTIONS

All authors contributed equally to this paper, reviewed and approved the final manuscript.

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Research Paper

Optimization of Dual-Constraint Robotic Arm Grasping Operations Based on an Optimized Contact Grasping Network

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With the continuous advancement of intelligent manufacturing and robot perception capabilities, traditional robotic arm grasping methods still face problems such as inaccurate pose prediction and poor task adaptability when operating in dynamic scenes, handling complex objects, and executing functional action. Therefore, this research develops an optimized contact grasping network model that integrates scene and task constraints. It combines a UR5 six-degree-of-freedom robotic arm, visual input, and the Contact-GraspNet architecture based on point cloud, and introduces a PointNet++ local feature enhancement mechanism and a lightweight encoder design to effectively improve the spatial perception and action planning capabilities of grasping points. Experimental results show that, on the GraspNet and YCB datasets, the model achieves $F1$ -scores of 92.54 % and 91.82 %, respectively, with the average execution time reduced to 0.61 s. In functional operation scenarios involving door handles, kettle handles, and drawer pulls, the grasping accuracy remained above 0.89, with the task completion rates significantly outperforming those of mainstream baseline models. Under visual interference conditions with an occlusion rate of up to 75 %, the average inference latency was controlled within 0.82 s. Under varying light intensities, the pose angle error remained within the range of 1.21° to 1.87° . Therefore, this model exhibits comprehensive advantages in grasping accuracy, latency control, and deployment efficiency, and has the potential to be largely applied in industrial, service, and specialized task environments.

Keywords: robotic arm grasping, scene constraints, task constraints, Contact-GraspNet, point cloud.



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1. INTRODUCTION

In the fields of automated manufacturing, service robots, and intelligent warehousing, robotic arm grasping systems, as a key component in human-computer interaction and object manipulation, are gradually developing toward high precision, high robustness, and adaptive capabilities. In practical applica-

tion scenarios, robotic arm grasping is not only limited by the physical environment but also affected by dual constraints of task objectives [1, 2]. Traditional contact grasping networks fail to consider these constraints simultaneously, often resulting in issues such as mis-grasping, collisions, or grasping failures in complex environments. Therefore, how to introduce task-level and scene-level dual constraints while retaining the efficient feature learning ability of deep grasping networks is the core problem that urgently needs to be solved.

1.1. RESEARCH PROGRESS AND LIMITATIONS OF TRADITIONAL GRASPING AND VISUAL RECOGNITION METHODS

Early robotic arm grasping primarily relied on methods based on visual detection, control strategies, and trajectory planning. For instance, CHIU *et al.* [3] employed channel pruning to enhance You Only Look Once version 5 (YOLOv5), integrating it with positioning algorithms and robotic arm motion planning to achieve grasping. Their approach reduced the number of parameters while improving grasping accuracy. GENG [4] enhanced robotic arm grasping stability and robustness through visual servoing and fractional-order control approaches. Concurrently, YUAN [5] improved adaptability to diverse objects by integrating multimodal information fusion and refining world models. However, such methods generally rely on rigid geometric assumptions or explicit scene configurations, exhibiting limited adaptability to complex conditions such as occlusion, deformation, and multi-object interference. Furthermore, they prioritize the executability of grasping actions while paying less attention to whether these actions fulfill functional task objectives, making it difficult to support functional operations such as opening doors, lifting objects, or rotating knobs.

1.2. DEEP LEARNING-BASED POINT CLOUD GRASPING AND CONTACT SEMANTIC INFERENCE RESEARCH

With the advancement of deep learning, point cloud-based grasping recognition methods have emerged as a mainstream research direction. XU *et al.* [6] proposed a grasping pose estimation method integrating multi-scale residual networks with RGB sensors, maintaining high recognition accuracy even in low-light environments. Grasping strategies represented by Contact-GraspNet (CG) enhance generalization capabilities for complex scenarios by inferring contact points, grasp scores, and grasp poses from point clouds [7]. Related extension studies include: YAN *et al.* [8] enhanced RGB-D feature extraction by incorporating attention and residual modules, GILLES *et al.* [9] achieved efficient grasping of unknown objects by integrating MetaGraspNetV2, and HOANG [10] improved point cloud denoising and the grasp position inference stability through attention mechanisms and the VoteGrasp strategy. While these approaches strengthen

the end-to-end mapping from visual information to grasp semantics, two core issues persist:

1. Networks remain vulnerable to point cloud density variations and noise, leading to unstable geometric feature extraction.
2. Models primarily focus on the geometric feasibility of grasp actions while lacking task-related functional constraints, making them unable to determine whether grasp poses match specific operational requirements.

1.3. DEVELOPMENTS IN FUNCTIONAL GRASPING, TASK CONSTRAINTS, AND POSE ESTIMATION RESEARCH

In task-relevant grasping research, scholars have gradually expanded their focus from ‘whether an object can be grasped’ to ‘whether the task action can be correctly executed.’ For example, YU *et al.* [11] introduced SCNet, which includes a self-supervised mechanism for category-level pose estimation to enhance the model’s transferability from simulation to reality. Other studies have focused on action planning for deformable objects or functional scenarios (e.g., knobs, handles). However, most approaches still lack systematic modeling of task constraints and rarely address critical factors such as motion direction consistency, rotational axis constraints, or grasping torque requirements. Furthermore, existing grasping strategies typically fail to integrate pose alignment mechanisms for coarse and fine registration, leading to amplified pose errors during functional component manipulation. This significantly impacts grasping success rates and operational stability.

In summary, although existing robotic arm grasping methods have made some progress in pose estimation and point cloud-based grasping pose inference, they still suffer from performance degradation and insufficient robustness in handling geometric interference, adapting to functional operation constraints, and ensuring contact grasping stability in complex scenarios. Therefore, this research develops a robotic arm contact grasping strategy model that fuses an optimized Contact-GraspNet architecture with a dual-constraint mechanism. This model improves point cloud representation efficiency by introducing local feature extraction and channel lightweight pruning strategies based on PointNet++, while introducing action mechanics models under task constraints to achieve task-oriented grasping pose discrimination. On this basis, this research integrates coarse and fine registration algorithms to effectively optimize pose estimation accuracy and operational consistency. The innovation lies in constructing an end-to-end grasping control system that integrates perception, structural optimization, and functional feedback, providing a high-precision, high-efficiency, and engineering-scalable technical solution for intelligent robotic arm operations under multiple constraint conditions.

2. METHODS AND MATERIALS

To avoid confusion, the study first introduces the standard Contact-GraspNet architecture to illustrate its fundamental workflow and input-output format. Subsequently, several proposed improvements are presented, including:

- a point cloud uniform downsampling strategy,
- a local geometric feature enhancement mechanism based on PointNet++,
- a lightweight design for encoder channel pruning,
- a task-constraint-coupled grasp candidate filtering strategy,
- a two-stage pose optimization based on the point pair feature (PPF) algorithm and the iterative closest point (ICP) algorithm.

2.1. CONSTRUCTION OF A ROBOTIC ARM GRASPING NETWORK MODEL CONSIDERING SCENE CONSTRAINTS

A UR5 six-degree-of-freedom robotic arm is selected as the operating platform in this study. The arm has a compact structure and high repeatability, with a positioning accuracy of up to ± 0.1 mm. It is widely used in manipulation and grasping of complex objects [12–14]. The study first models grasping operations within a 2D plane, assuming the target object rests stationary on a horizontal workbench. The schematic diagram of the 2D grasp and the grasp pose diagram are shown in Fig. 1 [15, 16]. It should be noted that the introduction of the 2D grasping schematic (Fig. 1) during the modeling phase is not intended to construct a 2D grasping model. Rather, it serves to visually illustrate the geometric relationship between the robotic end-effector and the target object, including the grasp base point, gripper width, formation of the contact triangle, and definition of the pose vector. This simplified schematic serves solely to explain the 3D symbols, parameters, and grasp semantics, and does not involve subsequent algorithmic derivation or experimental processes.

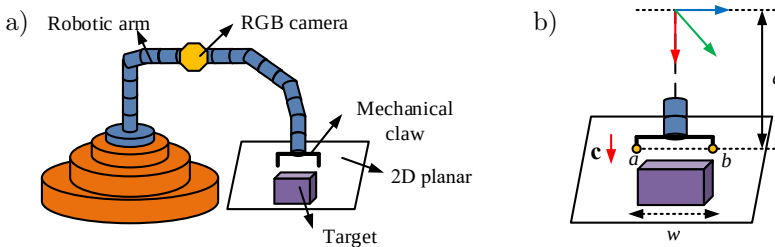


FIG. 1. Schematic diagram of 2D planar grasping (a) and grasp pose (b) representation of the robotic arm.

Figure 1 presents a 2D schematic of 3D grasping parameters, serving solely to illustrate the grasp pose definition method. Both model training and infer-

ence are implemented based on 3D point clouds. In Fig. 1a, the target object is stationary on a horizontal workbench, and the RGB-D camera is fixedly installed to obtain image data containing depth information. The robotic arm infers and executes the grasp point position and pose based on visual input. As shown in Fig. 1b, the gripper establishes contact with the surface of the object through the points a and b on both sides, forming a stable supporting contact triangle. The parameter w represents the width of the gripper, d signifies the Euclidean distance from the gripper center to the reference coordinate origin. The grasping direction of the gripper is represented by the vector $\mathbf{c}$. To efficiently predict grasp point positions and poses, the study introduces Contact-GraspNet as the basic network model. This network is an end-to-end grasp detection framework based on 3D point clouds, which can directly predict feasible grasp poses of each point from point cloud data generated by RGB-D images. It demonstrates excellent scene adaptation ability and contact semantic reasoning capability [17, 18]. The training process is shown in Fig. 2.

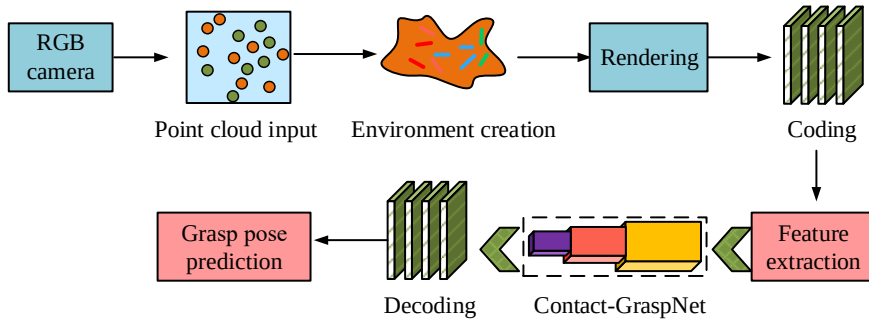


FIG. 2. Training flow of Contact-GraspNet.

In Fig. 2, the training of Contact-GraspNet mainly consists of four stages: encoding, feature extraction, grasp pose prediction, and loss feedback from the point cloud input $P \in R^{N \times 3}$ generated by the simulator. In terms of the workflow, we first explore using an RGB-D camera to capture depth images within the grasping scene. By integrating this depth information, we generate a 3D point cloud as the input for Contact-GraspNet. The input point cloud is voxelized into a 3D voxel grid $V \in R^{D \times H \times W \times C}$, and its calculation is as follows:

$$(2.1) \quad V(x, y, z) = \sum_{i=1}^N \left[\left[\frac{p_i - o}{r} \right] - (x, y, z) \right] \cdot f_i,$$

where p_i represents the i -th coordinate point, o signifies the voxel grid origin, r represents the voxel resolution, and f_i signifies the local feature of the i -th point. Secondly, the network predicts the grasp transformation $\hat{T}_j \in \text{SE}$ for each

voxel center point v_j , usually represented by the translation vector $\mathbf{t}_j$ and the axial rotation vector $\boldsymbol{\omega}_j$, as follows:

$$(2.2) \quad \hat{T}_j = \begin{bmatrix} \exp([\mathbf{w}_j]_{\times}) & \mathbf{t}_j \\ 0 & 1 \end{bmatrix},$$

where $\exp([\mathbf{w}_j]_{\times}) \in SO(3)$ represents the mapping from the Lie algebra to the Lie group, and $[\mathbf{w}_j]_{\times}$ represents the antisymmetric matrix form of vector $\mathbf{w}_j$. Then, Contact-GraspNet assigns a grasp score $\hat{s}_j \in [0, 1]$ to each candidate grasp pose g_j during training. The regression uses binary cross-entropy calculated as

$$(2.3) \quad L_{\text{score}} = -\frac{1}{M} \sum_{j=1}^M (s_j \log \hat{s}_j + (1 - s_j) \log (1 - \hat{s}_j)),$$

where L_{score} represents the grasp score loss, $s_j \in [0, 1]$ represents the ground-truth label determining whether it is feasible to grasp, $\hat{s}_j$ represents the retrievable probability predicted by the network, and M represents the number of candidate samples to be grasped. The total loss function is the sum of the grasp score loss and pose regression loss:

$$(2.4) \quad L_{\text{total}} = L_{\text{score}} + \lambda_{\text{pose}} \cdot L_{\text{pose}},$$

where L_{pose} represents the pose regression loss, including the point-to-point translation error and the rotation angle error, λ_{pose} represents the weight coefficient of the pose loss, and L_{total} represents the total loss function. Although Contact-GraspNet has achieved good results in point cloud-based grasp prediction, its original model lacks a unified standard for inputting point cloud data, resulting in insufficient learning stability of target grasp features when object density changes significantly. Therefore, a point cloud downsampling optimization based on distance uniformity is introduced:

$$(2.5) \quad P' = \text{FPS}(P, N), N = 1024,$$

where P represents the original input point cloud, P' represents the sampled point cloud input, and $\text{FPS}(\cdot)$ represents the farthest point sampling function. PointNet++ local feature encoding is used to replace single-layer global perception:

$$(2.6) \quad f_i^* = \text{MLP} \left(\bigoplus_{j \in N(p_i)} \phi(p_j - p_i) \right),$$

where $N(p_i)$ represents the spherical neighborhood centered at p_i , $\phi(\cdot)$ represents the geometric encoding function, and $\bigoplus$ represents the feature concatenation operation. MLP represents multilayer perceptron. Finally, the overall

channels and layers are pruned to achieve a lightweight architecture, and the calculation expression is

$$(2.7) \quad R^{1024 \times 3} \xrightarrow{\text{Enc}} R^{512 \times 128} \xrightarrow{\text{Dec}} R^{1024 \times 64},$$

where Enc represents the encoder, Dec represents the decoder, 1024 represents the number of points, 128/64 represents the number of feature channels. The improved Contact-GraspNet structure is shown in Fig. 3.

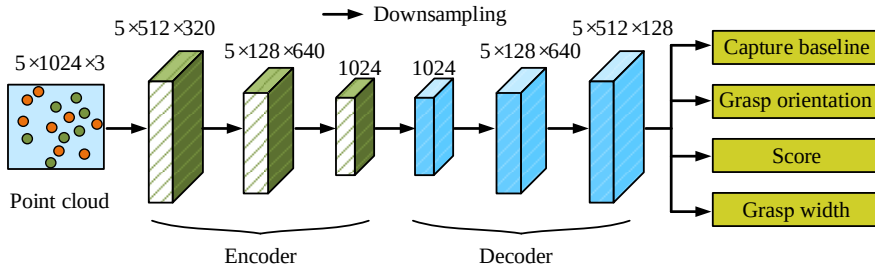


FIG. 3. Improved Contact-GraspNet structure.

In Fig. 3, the input point cloud is first unified into a batch data of size $5 \times 1024 \times 3$ and uniformly sampled using farthest-point sampling. The encoder consists of three convolutional layers, producing feature maps of sizes $5 \times 512 \times 320$, $5 \times 128 \times 640$, and $5 \times 512 \times 128$, respectively. Each layer incorporates channel compression to reduce redundant information while introducing a local feature enhancement mechanism based on PointNet++ to extract key local geometric structures in the third layer. The decoder progressively restores these features to a spatial dimension matching the input point count, producing an output of size $5 \times 1024 \times 128$. Based on this, the network predicts four parallel outputs: grasp base point coordinates, grasp pose quaternion, grasp score, and grasp width.

2.2. CONSTRUCTION OF A ROBOTIC ARM CONTACT GRASPING NETWORK STRATEGY MODEL INTEGRATING SCENE CONSTRAINTS AND TASK CONSTRAINTS

After constructing a robotic arm grasping network model that considers scene constraints, to further improve task adaptability and operational feasibility of grasping actions, a task constraint mechanism is introduced to recognize the geometric feasibility of grasp positions, and comprehensively evaluates whether grasping actions meet the requirements of specific operational tasks. The so-called task constraints refer to additional restrictions on target function, operation sequence, motion direction, and torque control, added to the grasping behavior. Taking a typical door handle grasping task as an example, the motion mechanics model and motion filtering operation diagram are shown in Fig. 4.

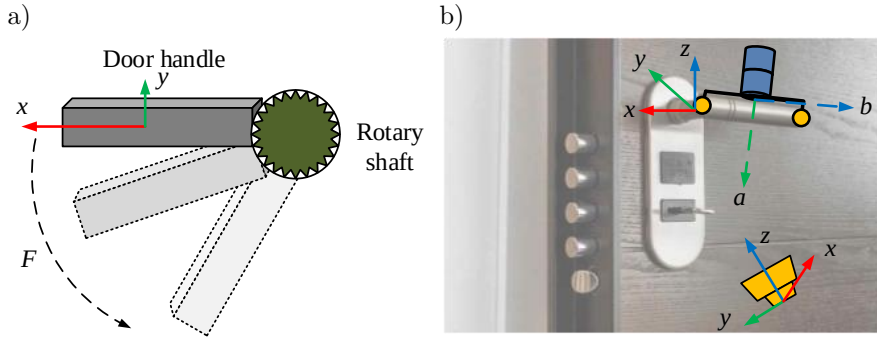


FIG. 4. Physical model of door handle motion (a) and operation schematic diagram of motion filtering (b) (picture source: <https://colorhub.me/photos/JP4m1>).

As shown in Fig. 4, the rotational torque T of the door handle is determined by the normal force F applied by the gripper at the contact point and the distance R from that point to the rotation axis, satisfying formula $T = F \cdot R$. Taking the local coordinate system of the door handle as a reference, the angular relationship between each candidate grasp pose and the rotation axis, and the consistency of the applied force direction, are calculated. If the contact direction of a certain grasp point deviates too much from the expected force direction, it will be filtered out by the task constraint module, and only grasp candidates that meet both contact stability and effective task execution will be retained. To achieve higher accuracy in grasp execution while satisfying task constraints, this study further adopts a pose estimation optimization method on the basis of the PPF algorithm, which accurately perceives and models the actual pose in the pre-grasping stage. Subsequently, based on coarse registration, the ICP algorithm is introduced to finely align the model point cloud with the target point cloud, further optimizing the rigid body transformation parameters to minimize the matching error. The overall process is shown in Fig. 5 [19, 20].

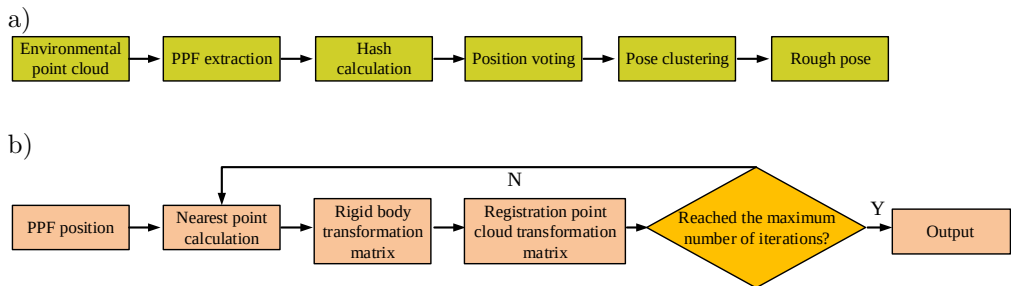


FIG. 5. PPF and ICP-based pose estimation processes: a) optimization process for pose estimation using the PPF algorithm, b) fine pose alignment process using the ICP algorithm.

As shown in Fig. 5, the PPF method first calculates feature relationships based on the distance, normal angle, and direction information of point pairs. It then generates locally invariant point pair features based on these relationships. The formal expression of the features is

$$(2.8) \quad \Phi_{ab} = (\|x_a - x_b\|, \angle(\mathbf{n}_a, d_{ab}), \angle(\mathbf{n}_b, d_{ab}), \angle(\mathbf{n}_a, \mathbf{n}_b)),$$

where x_a and x_b both represent the position coordinates of the two points in a point pair, $\mathbf{n}_a$ and $\mathbf{n}_b$ denote the unit normal vectors corresponding to two point pairs, and d_{ab} represents the point pair direction. After establishing an index for the feature quadruples in the hash table, PPF calculates candidate transformations one by one and performs pose voting based on the feature set of point pairs extracted from the query point cloud. The transformation score function is

$$(2.9) \quad S(H_k) = \sum_{\Phi \in F} \delta(\|\Phi - H_k(\Phi_m)\| < \varepsilon),$$

where H_k denotes the k -th candidate rigid body transformation, Φ_m is the model features, $\Phi \in F$ represents the feature set extracted from the scene point cloud, δ is the matching count function within the threshold, and ε represents the matching tolerance coefficient. The ICP algorithm aims to minimize the Euclidean distance between the target point cloud and the transformed model point cloud. A registration error function is constructed as

$$(2.10) \quad \tau(T) = \sum_{i=1}^N \|y_i - T \cdot z_i\|^2,$$

where $T \in SE(3)$ represents the rigid body transformation matrix to be optimized, z_i denotes the point in the model point cloud, y_i is the nearest neighbor point in the corresponding observation point cloud. When the registration error is below the set threshold or the transformation converges, ICP stops iterating and outputs the final fine alignment transformation as

$$(2.11) \quad T^* = \arg \min_{R, t} \sum_{i=1}^N \|y_i - (R \cdot z_i + t)\|^2,$$

where T^* represents the final rigid-body transformation matrix. At this point, the optimal solution is obtained by solving the covariance matrix through singular value decomposition, ensuring that the rigid-body transformation satisfies the minimum mean squared error criterion, thereby achieving high-precision fitting of the target point cloud pose. The pseudocode for the PPF and ICP algorithms is as follows:

Input:

$P_{\text{scene}} \leftarrow$ scene point cloud
 $P_{\text{model}} \leftarrow$ model point cloud
 $T_{\text{init}} \leftarrow$ initial pose estimate (optional)

Output:

$T_{\text{final}} \leftarrow$ final refined pose

Stage 1: Coarse Pose Estimation via Point Pair Features (PPF)

```

1: Sample keypoints from  $P_{\text{model}}$  to obtain  $M = \{m_k\}$ 
2: Sample keypoints from  $P_{\text{scene}}$  to obtain  $S = \{s_l\}$ 
3: Build PPF hash table:
4:   for each pair  $(m_k, m_j)$  in  $M$  do
5:      $f_{\text{model}} \leftarrow \text{PPF}(m_k, m_j)$ 
6:     insert  $f_{\text{model}}$  into hash table  $H$ 
7: Estimate coarse pose by feature matching:
8:   for each pair  $(s_l, s_i)$  in  $S$  do
9:      $f_{\text{scene}} \leftarrow \text{PPF}(s_l, s_i)$ 
10:    retrieve matching entries from  $H$ 
11:    accumulate votes for each candidate transform  $T_c$ 
12:     $T_{\text{PPF}} \leftarrow$  candidate transform with the highest vote count
13: if  $T_{\text{init}}$  exists then
14:    $T_{\text{coarse}} \leftarrow T_{\text{init}} \circ T_{\text{PPF}}$ 
15: else
16:    $T_{\text{coarse}} \leftarrow T_{\text{PPF}}$ 

```

Stage 2: Fine Pose Refinement via ICP

```

17: Set max iterations  $K$  and convergence threshold  $\varepsilon$ 
18:  $E_{\text{prev}} \leftarrow +\infty$ 
19: for iter = 1 to  $K$  do
20:    $P'_{\text{model}} \leftarrow T_{\text{coarse}}(P_{\text{model}})$ 
21:    $C \leftarrow$  nearest-neighbor correspondences between  $P'_{\text{model}}$  and  $P_{\text{scene}}$ 
22:    $E \leftarrow \Sigma \|p' - q\|^2$  for  $(p', q)$  in  $C$ 
23:   if  $|E - E_{\text{prev}}| < \varepsilon$  then
24:     break
25:    $\Delta T \leftarrow$  optimal rigid transform estimated by SVD over  $C$ 
26:    $T_{\text{coarse}} \leftarrow \Delta T \circ T_{\text{coarse}}$ 
27:    $E_{\text{prev}} \leftarrow E$ 
28:  $T_{\text{final}} \leftarrow T_{\text{coarse}}$ 

```

Additionally, scene constraints refer to restrictions in 3D space that determine whether a candidate grasp pose $\hat{g}_j$ satisfies environmental geometry, object surface accessibility, and collision feasibility. Scene constraints are expressed as

$$(2.12) \quad C_{\text{scene}}(\hat{g}_j) = \begin{cases} 1 & \text{if } d(\hat{g}_j, O) > \varepsilon, n_j \cdot s_j > 0, \\ 0 & \text{other,} \end{cases}$$

where O denotes the set of obstacles in the scene, $d(\hat{g}_j, O)$ represents the minimum distance between the grasp pose and the nearest obstacle, ε denotes the safety margin threshold, n_j is the surface normal of the grasping surface, s_j denotes the gripper closing direction, and $n_j \cdot s_j$ represents the dot product of the two vectors. A value greater than 0 indicates that the angle between the closing direction of the gripper and the normal direction of the surface is less than 90° , thus meeting the geometric conditions for stable contact. Task constraints determine whether the grasp pose satisfies functional operation requirements (e.g., rotating door handles, pulling drawers, pouring kettles). These constraints ensure that the grasp pose aligns with the intended force direction, as defined by the constraint function:

$$(2.13) \quad C_{\text{task}}(\hat{g}_j) = \begin{cases} 1 & \text{if } \angle(f_j, f_{\text{req}}) < \theta_{\text{max}}, \tau_j \geq \tau_{\text{min}}, \\ 0 & \text{other,} \end{cases}$$

where θ_{max} denotes the force direction vector corresponding to the grasp pose, f_{req} represents the desired force direction for the task (e.g., the rotation direction of a door handle), θ_{max} indicates the permissible directional deviation, τ_j signifies the torque generated by the grasping action, and τ_{min} denotes the minimum torque required to complete the task. A dual-constraint grasping operation model for a robotic arm based on scene constraints and task constraints is proposed, which integrates an optimized Contact-GraspNet. The process is shown in Fig. 6.

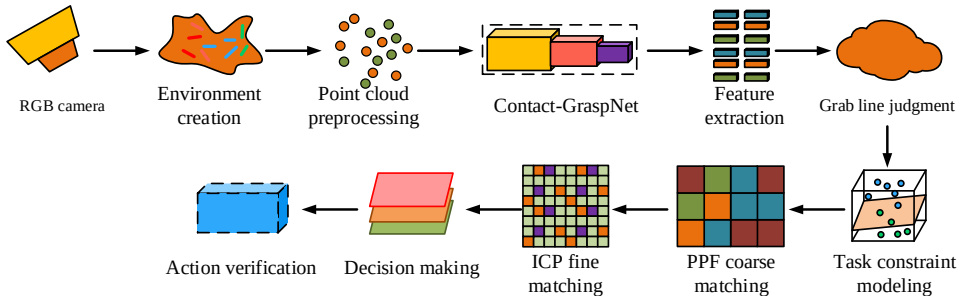


FIG. 6. Dual-constraint grasping operation model process of the robotic arm based on the optimized contact grasping network.

In Fig. 6, the point cloud is first obtained through an RGB-D camera and uniformly sampled to construct a 2D scene model containing target object and ob-

stacle information. Subsequently, based on the improved Contact-GraspNet, point cloud features are extracted, and the grasping base points, pose quaternions, scores, and grasp widths are output. On this basis, a task constraint module is introduced and combined with the functional attributes and operating direction of the target object to construct a physical motion model and filter out grasp candidates that do not satisfy the functional execution requirements. Next, PPF is used for coarse registration, and ICP is combined to complete point cloud fine registration, producing a more accurate object pose estimation. Ultimately, the system integrates grasp score, stability, and task feasibility to select the optimal grasping solution for path planning and action execution, and achieves closed-loop grasping control through sensor feedback to improve the overall grasping success rate and adaptability.

3. RESULTS

3.1. PERFORMANCE TESTING OF THE NEW ROBOTIC ARM CONTACT GRASPING NETWORK STRATEGY MODEL

The research is performed on the Ubuntu 20.04 platform to deploy the grasping model. The experimental environment is configured with an Intel Core i9-12900K CPU, an NVIDIA RTX 3090 GPU, and a memory capacity of 64 GB. The deep learning framework uses PyTorch 2.0, and the point cloud processing part integrates the Open3D and PCL libraries. The GraspNet-1Billion dataset (GraspNet) and the Yale-CMU-Berkeley Object and Model Set (YCB Dataset) are used as the test data sources. GraspNet is currently the largest real-world 3D grasping dataset. This dataset covers 190 different objects and generates over 1 billion grasp annotations, supporting RGB-D point cloud input and 6D grasp pose output. The YCB Dataset includes 77 common object categories encountered in daily life, such as cups, nuts, scissors, pot lids, etc. It contains RGB images, depth maps, segmentation masks, and object pose information. The study conducted training, validation, and testing on two datasets: GraspNet-1Billion and YCB Dataset. To ensure experimental reproducibility, GraspNet employs the officially provided scene-wise split: all objects and poses from scenes 0000–0845 form the training set, scenes 0846–0888 constitute the validation set, and scenes 0889–1000 comprise the test set. The training set contains 88.2% of the data volume, while the test set comprises 11.8%. For the YCB Dataset, the study employs an object-category-based splitting method: 70% of objects are allocated to the training set and 30% to the test set based on object ID, ensuring that the test objects do not appear during training to evaluate the model's cross-category generalization capability. No additional data augmentation is applied to either dataset. All point clouds undergo uniform cropping and

normalization processing before entering the network. Both datasets are widely used in robotic grasping, pose estimation, and action planning tasks. First, the two types of hyperparameters that have the greatest impact on the model performance are validated, and the selected values are shown in Fig. 7.

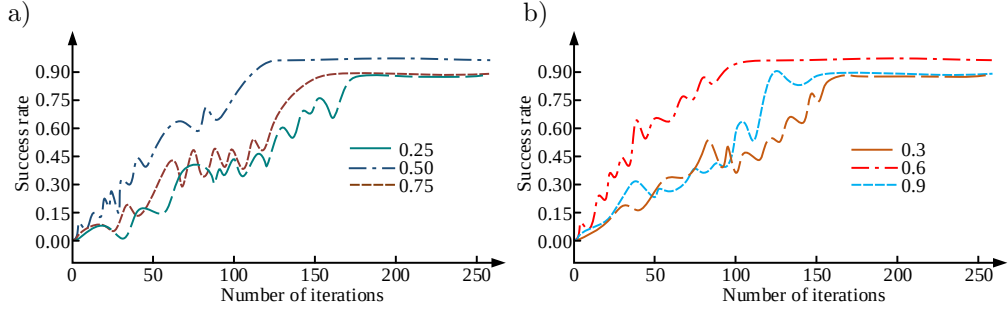


FIG. 7. Test results of hyperparameter selection: a) pose loss weight coefficient, b) matching tolerance coefficient.

According to Fig. 7a, as the number of iterations increased, the success rates of all three weight settings gradually increased, but the difference gradually became apparent after about 150 iterations. When the pose loss weight was set to 0.5, the model achieved a good balance between convergence speed and final success rate, and the final success rate remained stable above 0.93, which was better than for the 0.25 and 0.75 settings. The latter produced convergence fluctuation or overfitting trend. According to Fig. 7b, the curve converged the fastest when the matching tolerance was 0.6, reaching a success rate of 0.94 after about 100 iterations and maintaining high stability thereafter. When the tolerance was 0.3, the convergence speed was significantly slower, and the final success rate was slightly lower. Although the initial convergence was rapid with a tolerance of 0.9, it was prone to excessive mismatches in the middle and later stages, which could affect matching accuracy. After comprehensive consideration, the study ultimately chooses a pose loss weight of 0.5 and a matching tolerance coefficient of 0.6 as the default hyperparameter settings.

To simulate the closing process of a mechanical gripper in offline evaluation, this study adopts the parallel gripper closing model consistent with the official GraspNet evaluation. For each grasp candidate pose $\hat{g}_j = (t_j, q_j, w_j)$ output by the network, the parallel gripper is first instantiated at the pose (t_j, q_j) with an initial opening width $w_{\max}$. The predicted grasp width w_j is truncated to the interval $[w_{\max}, w_{\min}]$. Subsequently, the distance between the fingers is gradually reduced along the gripper closing direction at a fixed step size (1 mm). Collisions between the gripper finger surfaces and the object/scene mesh are detected at each iteration step. When both finger surfaces make contact with the object

surface without colliding with the environment, the grasp is recorded as having reached a stable closing state, and the clamping width at this point is locked. If, during closure, either finger surface intersects with the table surface or other obstacles, or if effective contact cannot be achieved on both sides before reaching $w_{\min}$, the grasp candidate is deemed an infeasible grasp. Based on this, grasp success is calculated according to the rules defined in the GraspNet benchmark. If the closed pose simultaneously falls within the annotated grasp tolerance for both position and orientation, and no illegal collisions occur during the closing process, it is considered a successful grasp and counted toward valid grasp counts and metrics such as precision, recall, $F1$; otherwise, it is deemed a failed grasp. The entire closing and evaluation process is conducted entirely within the simulation environment, independent of real hardware testing. This ensures consistent evaluation conditions across different models. The research conducts ablation tests on the proposed model to further determine its performance, as presented in Fig. 8.

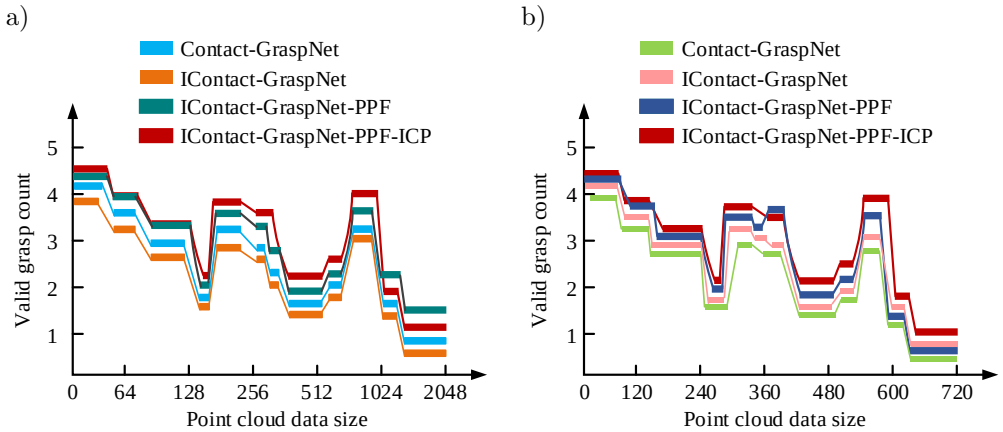


FIG. 8. Ablation test results: a) GraspNet, b) YCB Dataset.

In Fig. 8, the four model names represent different refinement stages: Contact-GraspNet denotes the original Contact-GraspNet baseline model, IContact-GraspNet refers to the proposed Improved Contact-GraspNet (abbreviated as ‘IContact’), which incorporates point cloud uniform downsampling and PointNet++ local feature enhancement into the original model. IContact-GraspNet-PPF denotes the model with a coarse registration module based on PPFs added to IContact-GraspNet, IContact-GraspNet-PPF-ICP denotes the final version of the research model, which incorporates an ICP fine-alignment module after PPF coarse alignment to achieve the highest pose estimation accuracy and grasping stability. Here, the ‘I’ indeed stands for ‘improved,’ indicating that this model is an enhanced version of the original Contact-GraspNet.

From Fig. 8a, the original Contact-GraspNet model had an effective grasp count of 2.8 at 1024 input points. while the IContact-GraspNet model increased this value to 3.2. After introducing PPF, it was increased to 3.5. Finally, the integrated PPF+ICP model achieved the highest grasp count of 3.8 at the same number input points, indicating an increase of 35.7 %. Under 512 input points, the model also achieved an effective grasp of 3.4, significantly better than other model versions. In Fig. 8b, on the YCB dataset, with 720 input points, the IContact-GraspNet-PPF-ICP model had an effective grasp count of 3.6, which was higher than that of Contact-GraspNet (2.9), representing a performance improvement of over 24 %. Advanced grasping models were introduced for comparison, such as the grasp pose detection (GPD) model, the generative grasping convolutional neural network (GC-CNN), and the grasping residual convolutional network v2 (GR-ConvNet v2). Table 1 presents the test results, taking precision, recall, $F1$ -score, and average execution time as indicators.

TABLE 1. Performance results of different models.

Dataset	Model	Precision [%]	Recall [%]	$F1$ -score [%]	Average execution time [s]
GraspNet	GPD	85.73	82.45	84.06	0.94
	GC-CNN	88.22	85.36	86.77	0.68
	GR-ConvNet v2	89.91	87.52	88.74	0.72
	Our model	93.67	91.44	92.54	0.61
YCB Dataset	GPD	83.48	80.13	81.78	0.98
	GC-CNN	86.75	84.02	85.36	0.71
	GR-ConvNet v2	88.66	86.19	87.41	0.74
	Our model	92.88	90.75	91.82	0.63

In Table 1, on the GraspNet dataset, the precision of the proposed model reached 93.67 %, a recall was 91.44 %, and an $F1$ -score reached 92.54 %, which was significantly better than GPD ($F1$ -score of 84.06 %) and GR-ConvNet v2 ($F1$ -score of 88.74 %). Meanwhile, its average execution time was only 0.61s, better than GC-CNN (0.68s), demonstrating strong computational efficiency. On the YCB dataset, the model still maintained its lead with an $F1$ -score of 91.82 %, which was about 10 % higher than GPD. In addition, its precision and recall both exceeded 90 %, indicating that the model also has high robustness and good generalization capability in general object grasping scenarios. Through comprehensive comparison, the optimized contact grasping network is superior to the current mainstream models in terms of accuracy and real-time performance, and has significant advantages.

3.2. SIMULATION TESTING OF THE NEW ROBOTIC ARM CONTACT GRASPING NETWORK STRATEGY MODEL

All functional grasping simulations were implemented based on the standard DH parameters and dynamic model of the UR5 six-degree-of-freedom robotic arm. The corresponding motion execution performance metrics, such as grasping success rate, pose error, and energy consumption, are presented in Fig. 9, Fig. 10, and Table 2, respectively. To verify the practical performance effect of the proposed model, three typical grasping objects are randomly selected from the GraspNet dataset to evaluate whether the model can meet the functional execution requirements in operations with task objectives. The results are shown in Fig. 9.

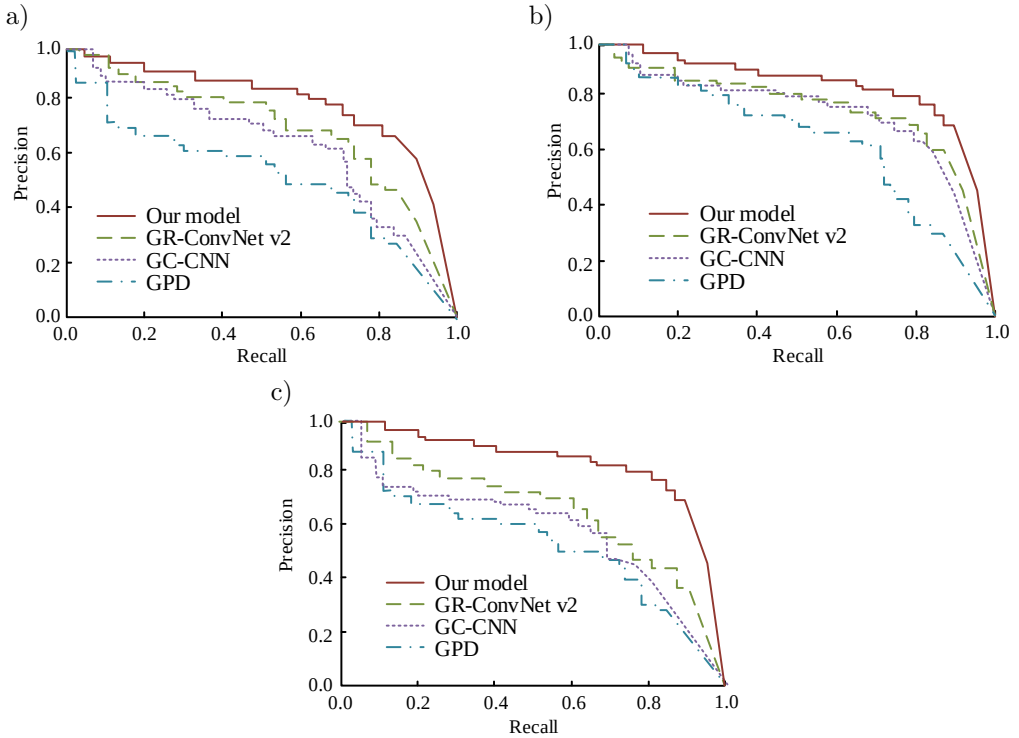


FIG. 9. Test results of functional operational adaptability: a) door handles, b) kettle handles, c) drawer pulls.

As shown in Fig. 9, the proposed model maintained the highest precision throughout the entire recall interval, especially in the 0.4–0.8 recall segment, where the precision remained stable above 0.85, while the precision of GPD dropped below 0.6 in this segment, showing the weakest performance. In the kettle handle grasp task, the proposed model still achieved a precision of 0.88 when the recall was 0.6, significantly higher than those of GC-CNN (0.74) and

GR-ConvNet v2 (0.79), demonstrating stronger stability and anti-interference ability. The drawer pull test results displayed that the proposed model maintained a precision of around 0.90 in most of the recall intervals. Even at high recall rates (above 0.9), there was no significant decrease in precision. However, the other three models showed a significant decrease in precision in this area, indicating that this model had stronger adaptability and robustness for edge grasping and functional component operation. The study continues to verify the average time latency of grasp prediction using different methods under different occlusions, and the results are shown in Fig. 10.

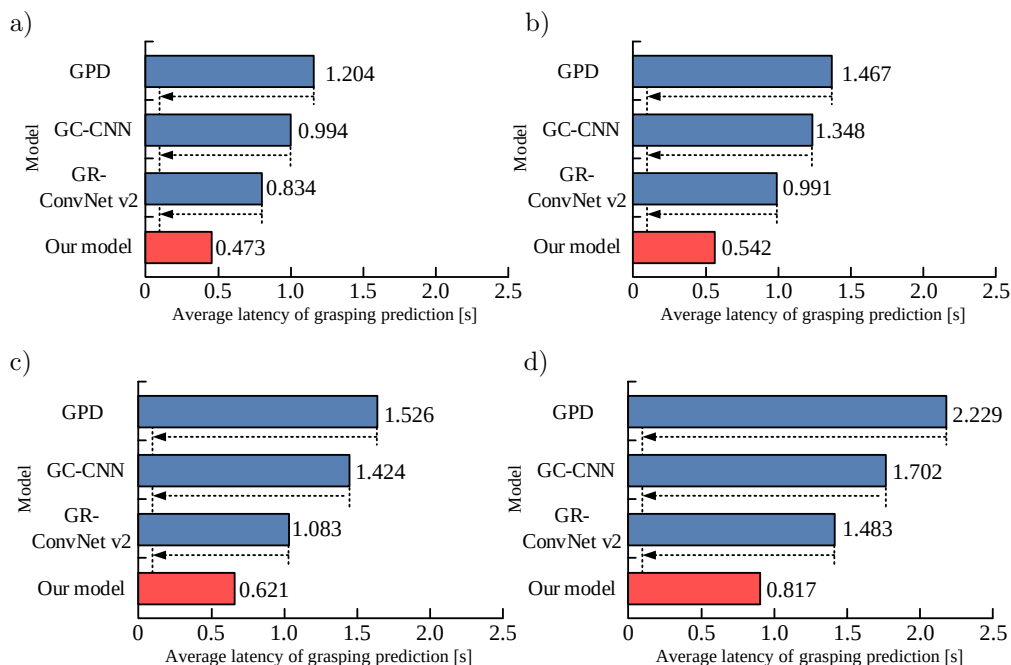


FIG. 10. Average latency results of grasp prediction of different methods under visual occlusion conditions: a) 0%, b) 25%, c) 50%, d) 75%.

As shown in Fig. 10, the proposed model consistently had the lowest grasp prediction latency and faster computational response under various occlusion conditions. Even in complex scenes with an occlusion rate of up to 75%, the average latency of the system was still controlled within 0.82 s, far lower than other mainstream models whose latencies generally exceeded 1.4 s. In contrast, GPD and GC-CNN showed a significant increase in latency as the occlusion intensified, with larger fluctuations and insufficient stability. Although GR-ConvNet v2 showed a certain anti-interference ability, its latency still remained above 1 s in high occlusion situations. Relatively speaking, the inference process of the proposed model was more robust in occluded environments, reflecting the synergis-

tic advantages of structural optimization and lightweight coding. It is therefore more suitable for deployment in application scenarios that require high real-time performance. Finally, the grasp pose angle error, model parameter count, and energy consumption per grasp of each model under different lighting intensities were verified, as displayed in [Table 2](#).

TABLE 2. Test results of model indicators under different lighting conditions.

Illumination level	Model	Pose angle error [°]	Model parameters [M]	Energy per grasp [J]
Low light	GPD	3.87	34.78	11.32
	GC-CNN	3.15	28.32	9.85
	GR-ConvNet v2	2.93	22.14	9.36
	Our model	1.64	18.53	8.45
Normal light	GPD	3.02	34.78	10.25
	GC-CNN	2.41	28.32	8.98
	GR-ConvNet v2	2.05	22.14	8.42
	Our model	1.21	18.53	7.84
High light	GPD	3.55	34.78	11.87
	GC-CNN	2.84	28.32	10.34
	GR-ConvNet v2	2.42	22.14	9.63
	Our model	1.87	18.53	9.27

To quantify the energy consumption differences among the various models during grasp action, this study employs a method based on integrating the joint torques and angular velocities to estimate the energy consumed in each grasping cycle. Specifically, during the entire process of the UR5 robotic arm moving from its initial standby pose to the target grasping pose and completing the closing motion, the torque $\tau_i(t)$ and angular velocity $\omega_i(t)$ of each joint were recorded at a frequency of 1 kHz. The mechanical energy consumption per grasping action is defined as

$$(3.1) \quad E = \sum_{i=1}^6 \int_{t_0}^{t_f} |\tau_i(t)\omega_i(t)| dt,$$

where t_f and t_0 denote the start and end times of the current grasp trajectory, respectively. The absolute values are used to prevent positive and negative work from canceling each other out. During actual computation, the integral is discretized into a sum over sampling points:

$$(3.2) \quad E^* = \sum_{i=1}^6 \sum_k |\tau_i(k)\omega_i(k)| \Delta t.$$

For each lighting condition and each model, 50 successful grasps were performed repeatedly under identical initial poses and trajectory planning strategies. The average energy consumption value obtained is listed as ‘energy per grasp [J]’ in Table 2.

According to Table 2, in terms of pose angle error, the error of the proposed model under low-light, normal-light, and high-light conditions was 1.64° , 1.21° , and 1.87° , respectively, all significantly lower than those of other comparative models, reflecting the stable prediction ability of the model for grasp pose under complex visual conditions. The model parameter count was controlled at 18.53 M, which was more compact and computationally efficient compared with GPD (34.78 M) and GC-CNN (28.32 M). In terms of energy consumption, the new model had the lowest average energy consumption per grasp, only at 7.84 J (normal light), 8.45 J (low light), and 9.27 J (high light), further verifying the efficiency and practicality of the model in resource-limited scenarios. Overall, the proposed model outperformed the mainstream methods in terms of grasp pose estimation accuracy, structural complexity, and energy efficiency, demonstrating strong practical potential.

4. CONCLUSION

This study proposed an optimized contact grasping network model that integrates scene constraints and task constraints to satisfy the practical needs of robotic arm grasping operations in complex unstructured environments. By introducing point cloud downsampling, a lightweight encoder design, and a PointNet++ local feature enhancement mechanism, the improved network had a precision of 93.67 % and an $F1$ -score of 92.54 % on the GraspNet, with an average execution time reduced to 0.61 s. Its performance was significantly better than that of mainstream models such as GPD, GC-CNN, and GR-ConvNet v2. In the functional grasping task test, the proposed model maintained accuracy advantages in three types of objects: door handles, kettle handles, and drawer pulls. At a recall rate of 0.6, the precision exceeded 0.88, verifying its adaptability and robustness in executing task-constrained grasping actions. Under visual occlusion conditions with a rate of up to 75 %, the model still maintained an average inference time of less than 0.82 s, displaying strong robustness to occlusion. Under different lighting conditions, the lowest pose angle error was only 1.21° , the model parameters were controlled at 18.53 M, and the lowest energy consumption per grasp was reduced to 7.84 J. The above results indicate that the proposed dual-constraint grasping strategy not only has comprehensive advantages in grasping accuracy, response speed, and resource efficiency, but also significantly improves grasp pose estimation and task execution performance in complex task-oriented grasping scenarios, and shows good potential for

engineering deployment. However, this study did not focus on grasping small objects. Future research will further combine multi-sensor fusion and reinforcement learning mechanisms to enhance the decision-making intelligence and execution stability of the model in multitask concurrent operations and dynamic scene adaptation.

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CONFLICTS OF INTEREST

The author declares that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHOR'S CONTRIBUTION

The author conceptualized the study, performed the analysis, wrote the original draft, and approved the final manuscript.

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Review Paper

Challenges of Corrosion, Wear, Erosion, and Abrasion in Hydropower Plants: Materials, Modeling, and Mitigation Strategies

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The chemical, food processing, hydropower, thermal power, and oil industries are among the sectors that frequently face problems of erosion, abrasion, and corrosion. Pipelines, elbows, reducers, separators, tees, and seals are among the hydraulic devices and pipeline components that are impacted by these phenomena. Silt erosion in turbines and related components is a major issue in hydropower plants, particularly in Indian hydropower plants where rivers contain hard minerals such as feldspar, quartz, and other mineral particles. Remarkably, high content of quartz in the silt causes problems in turbines, including sediment erosion, leakage, and secondary flow disturbances. Hydropower plants' total efficiency is ultimately jeopardized by these issues. This paper aims to support scholars and researchers by highlighting the following topics: (a) component failures in impulse and reaction turbines used in hydropower projects, (b) different turbine materials and their properties, and (c) a comparison of several thermal spraying techniques for turbine materials, along with numerous numerical models of erosion and abrasion informed by silt characteristics, material properties, and flow phenomena in different hydroturbines. The study also discusses modeling, pilot plant loops, wear mechanisms, and protective techniques aimed at mitigating wear and safeguarding hydroturbines.

Keywords: thermal spray coatings, erosion, component failure, turbine, hydropower plants.



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1. INTRODUCTION

Global energy consumption is rising steadily, which is putting tremendous strain on all energy sources. Currently, the most widely used energy source comes from non-renewable fossil fuels. According to estimates, fossil fuel supplies could run out by the end of 21st century [1–3]. Fossil fuel burning also produces toxic

gaseous byproducts that harm the environment. These gases cause serious health hazards in addition to their contribution to climate change [4–7]. In response, renewable energy sources will soon be essential to meet our expanding energy needs. The basic understanding of using hydropower goes back to the mid-1770s, as demonstrated by the work of the French engineer Bernard Forest de Bélidor, who presented the working designs of horizontal and vertical hydraulic machinery [8–10]. However, the first hydropower-driven turbine came into existence in 1878 at Cragside in Northumberland, England, developed by William George Armstrong. Thereafter, the Schoelkopf power station in 1881 and the Edison hydroelectric power plant in 1882 in Appleton, Wisconsin, started to produce electricity from water¹⁾. This was followed by rapid development, resulting in around 200 hydropower stations in the United States. alone by the end of 1889²⁾. A remarkable achievement was made in 1928 with the construction of Hoover Dam, with an initial capacity of 1345 MW. It remained the largest hydropower plant until 1936, when it was surpassed by the Grand Coulee dam in 1942³⁾ with a capacity of 6809 MW. The trend to produce increasingly larger hydropower plants went on and in 1984, a plant with a capacity greater than 10 GW came into existence. The Itaipu Dam, a joint venture of Brazil and Paraguay with a capacity of 14 000 MW, was built. The construction of the Guri Dam in Venezuela with a capacity of 10 200 MW was initiated in 1963, and it is presently the third-largest dam in the world. The Three Gorges Dam in China, currently the largest hydroelectric power plant with a capacity of 22 500 MW, began operating in 2008⁴⁾. Today, China is the largest producer of hydroelectricity with 721 terawatt hours generated in 2010. Some countries, such as Norway, the Democratic Republic of Congo, Paraguay, and Brazil, are endowed with large water resources and have the potential to produce the bulk of their energy requirements through hydropower³⁾. Paraguay generates 100 % of its energy requirements from hydropower. It is also a source of national income, with 90 % of its electricity generation being exported to Brazil and Argentina. Norway also generates 98 % of its electricity needs from hydropower. At present, almost 16 % of the total generated energy in the entire world is from hydropower. This figure is expected to increase with the initiatives of Asian and African continents, where many of the hydropower resources remain unharnessed. In 2010, China alone added 16 GW of hydropower generation capacity and planned to add another 140 GW by 2015 [8–10].

In the current scenario, India is among the top ten hydroelectric power-generating countries. Out of its total installed capacity of 174 GW in 2012, 11 %

¹⁾<http://www.usbr.gov/power/edu/history.html>

²⁾<http://www1.eere.energy.gov>

³⁾www.worldwatch.org

⁴⁾www.hydroworld.com

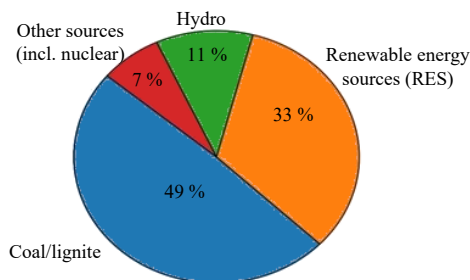


FIG. 1. Distribution of India's power generation capacity by energy source in 2012, RES stands for other renewable energy sources.

was contributed by the hydropower, as shown in Fig. 1. By the end of 2012, hydroelectric generation increased to 39 GW (an increase of ~ 2 GW)⁵⁾. However, the percentage contribution of hydropower appears to have reduced marginally, as can be observed in Fig. 1. Nevertheless, the major projects currently under construction are surely expected to improve this figure. To mention a few, the hydropower plants at Upper Siang (11 000 MW), and Lower Subansiri (2000 MW) are some of the future projects that could help India gain significant momentum in the field of hydropower generation. In Fig. 2, the trend in hydroelectric generation with the total installed power generation capacity in India since 1950 can be found⁵⁾. It can be observed that hydropower has shown a steady and gradual rise since 1950. Indian rivers have a great potential for hydropower production, a significant portion of which remains untapped. Large potential lies in the Brahmaputra, Indus, and Ganga basins, which need to be developed. In terms

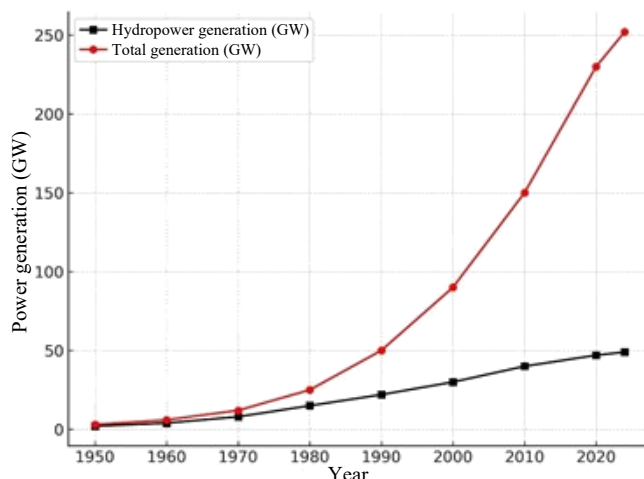


FIG. 2. Variation in India's hydropower and total power generation capacity from 1950 to 2024.

⁵⁾<http://powermin.nic.in/>

of installed hydroelectric generation capacity, India ranks sixth worldwide³⁾. An idea about the hydropower potential of India can be obtained from the fact that the country has a 150 000 MW hydropower capacity, which at a 60 % load factor amounts to around 84 000 MW. Additionally, 15 834 MW has been estimated from small hydropower projects. This provides an idea about the growth potential that can be expected in this sector [5].

2. PROBLEMS IN HYDROPOWER PLANTS

Ideally, a hydropower plant, once installed, should continue to operate without frequent breakdowns and problems. It is known that corrosion, a potential hazard for submerged turbines, is not a significant issue anymore in these plants. With the emergence of stainless steel in the early twentieth century, this material is widely used in hydraulic machinery where corrosion is a major concern. The addition of chromium (Cr) and nickel (Ni) to iron resulted in steels that are highly resistant to corrosion. With time, many different grades of stainless steels were developed by varying the proportions of their principal alloying elements, namely Cr and Ni. For hydroturbine applications, steels containing Cr > 12 % are usually recommended [11, 12]. Various stainless steels that found their potential use in hydroturbine applications are listed in Table 1, along with their basic mechanical properties and microstructures. Among these different steels, 13Cr4Ni, also known by its trade name, CA6NM, is the most widely employed structural steel for hydroturbine applications. Here, the first letter C is a designation stating that it belongs to the corrosion-resistant family of stainless steels. The letter ‘A’ gives a rough estimation of the amounts of its major alloying elements. The numerals ‘6’ represents 1/100th of the carbon content, which is 0.06 % in the present case. The last two letters are the representation of

TABLE 1. Properties of different turbine steels [21].

Stainless steel	Tensile strength [MPa]	Yield strength [MPa]	Elongation [%]	Impact strength [J]	Microstructure	Nature
13Cr1Ni	630	470	18	39	Martensite	Hard and brittle
13Cr6Ni	800	550	16	70	Martensite – 70 % Austenite – 30 %	–
17Cr4Ni	880	650	12	59	Martensite + Ferrite	Hard and ductile
13Cr4Ni	823	686	23	81	Martensite + Austenite	Hard
16Cr5Ni	880	600	21	100	Martensite – 65 % Austenite – 35 %	Hard and ductile

the alloying elements, namely Ni and Mo. The use of highly corrosion-resistant materials such as CA6NM has eliminated the problem of corrosion in hydro-turbines. However, wear continues to affect the performance and degradation phenomenon of hydroturbines [13]. The performance of these hydroturbines is severely impacted by the presence of sand/silt particles in the river water that flows through the turbines [14–16]. Loosely bound soil particles, caused by deforestation, continue to be transported into rivers with rainwater. Secondly, the young mountains such as the Himalayas have fragile morphology; therefore, rivers originating from such mountain ranges experience a regular influx of sand, rocks, and debris due to frequent erosion and landslide activities [17–20].

3. WEAR IN HYDROTURBINE

Wear is a surface degradation phenomenon resulting from the interaction of solid materials with the surrounding environment. As a result of wear, degradation at the surface of the material occurs either in the form of material loss or its relative displacement. Although it is a localized phenomenon at the initial stage, the damage to the material surface can intensify and coalesce upon continued exposure to degrading environments. This intensifies the degradation process, which further worsens the case and can result in the complete stalling of the system [21]. The accumulation of damage and the continuous loss of material from the surface can make the exposed component unfit for further service. Wear accounts for significant economic losses in any country. For example, according to an estimate given in 1965, the loss of 515 million pounds in the United Kingdom alone could have been avoided by giving due attention to tribology [22]. In Germany, the total losses due to wear in 1983 were estimated to be around 13 billion pounds. In the United States, losses due to friction have been estimated to be around \$100 billion. Depending upon the type of the surrounding environment responsible for target surface wear and the material removal mechanism involved, the process of wear can be categorized into various types as discussed in the next section. Various types of wear mechanisms are presented in Fig. 3.

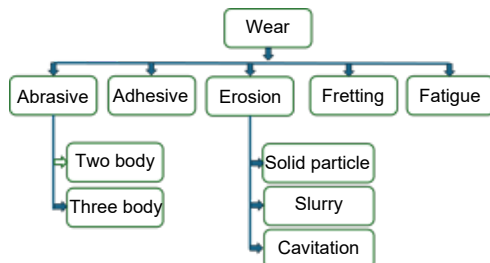


FIG. 3. Various types of wear mechanisms.

According to the American Society for Testing and Materials (see, ASTM-Standard G40, 2013), erosion is defined as the slow loss of original material from a solid surface that is produced by mechanical contact between the surface and a fluid, a multi-component fluid, or impinging liquid and/or solid particles. As it is clear from the definition itself, the impact of a fluid and/or solid, particles or both, results in mechanical damage of the surface, known as erosion. The different forms of erosion, shown in Fig. 4, can further help in understanding the different types of erosion. If erosion is brought on by the impact of solid particles that are entrained in a gaseous medium, this type of erosion is known as solid particle erosion (SPE). Slurry erosion (SE), on the other hand, is erosion that occurs when the liquid acts as a carrier medium rather than gas. Cavitation erosion (CE) and liquid droplet erosion (LDE) are two other types of erosion that arise generally from the interaction of the fluid and the solid surface. When CE is present, deterioration occurs because of cavitating circumstances that result from the implosion of cavities or bubbles within liquids containing vapor or mixtures of vapor and gas, as shown in Fig. 4. It was Rayleigh who first proposed that the collapse of vapor bubbles, resulting in very high loads, could initiate the phenomenon of cavitation [23].

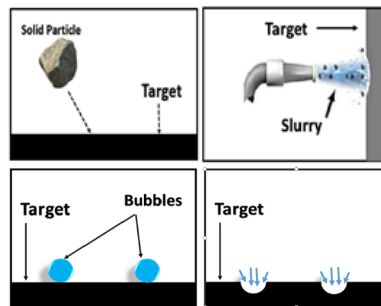


FIG. 4. Different types of erosion phenomena.

4. IMPACT OF EROSION ON FLUID MACHINERY

There are several severely silt-affected hydropower stations in India. The categorization of these plants was made based on the level of impact energy possessed by the silt particles. Plants in which the particles had kinetic energy greater than $10 \mu\text{J}$ required repair after every second monsoon, whereas repair was required after every three to four monsoons in the plants where the particle impact energy was in the range of $5 \mu\text{J}$ to $10 \mu\text{J}$. The degraded blade of a 250 MW Francis turbine at Naptha Jhakri and Baira Siul [24] is shown in Fig. 5. Plants in which the impact energy of the particles was less than $5 \mu\text{J}$ required repair after every seven to eight monsoons. In this categorization, however, no consideration was given to the concentration of the silt particles [24].

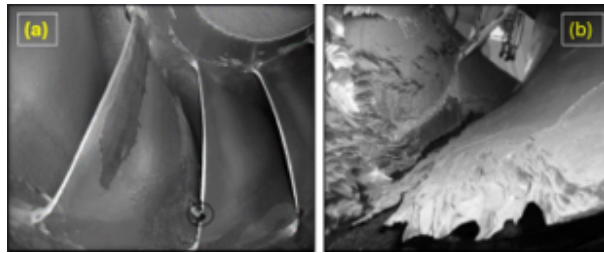


FIG. 5. Degraded turbine blade of a 250 MW Francis turbine at Naptha Jhakri and Baira Siul: a) degraded turbine blade, b) enlarged view of the degraded portion of the turbine blade.

5. SURFACE MODIFICATION TECHNIQUES

For the purpose of altering the surface characteristics of materials, a number of surface modification methods are shown in Fig. 6 [25, 26]. Some of these processes pose a threat to the environment [27, 28]. Two of such processes are chemical vapor deposition (CVD) and physical vapor deposition (PVD). Furthermore, using this method makes it challenging to create coatings that are thick. Additionally, the heat-affected zone (HAZ) that is generated during processing has an effect on the characteristics of the underlying substrate material in the form of dilution. Other methods, such as laser-assisted surface modification and weld overlay, are very closely related approaches. Thermal spraying is a versatile technique that can be used to deposit virtually any type of material onto almost any type of substrate material. It is a versatile technique employing several combinations of materials to be coated onto substrates without significantly altering the properties of the substrate material. Another emerging technique for surface modification is friction stir processing (FSP) [29, 30].

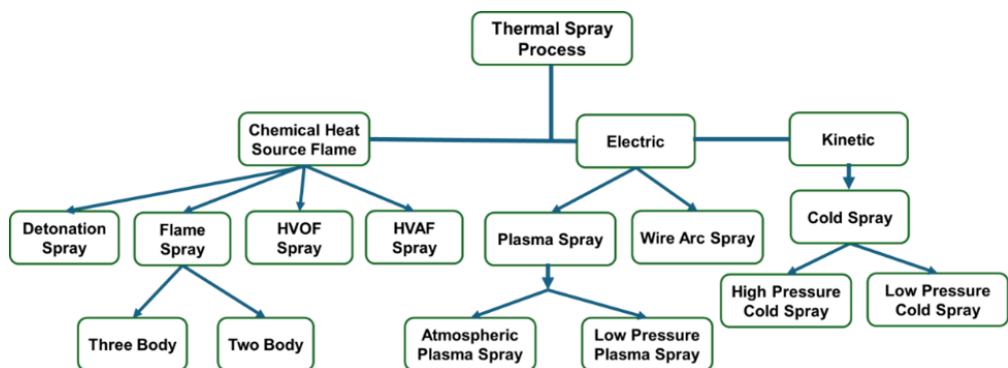


FIG. 6. Classification of thermal spray processes.

The thermal spraying technique involves spraying molten metal onto a substrate to deposit it as a coating. The coating material is first delivered as a pow-

der or wire, then heated to a molten state and next sprayed onto the substrate at high velocity, with particles as small as a micrometer. Thermal spraying uses combustion or an electrical arc discharge as the energy source. The material is covered by the sprayed particles as they accumulate on the substrate surface. The porosity, oxide content, macrohardness, surface roughness, and other characteristics of the coating affect its quality. Currently, these coatings are the most widely used. The use of highly erosion-resistant materials is restricted by financial and mechanical strength constraints. The cost of these materials is too high to present a practically viable solution. Secondly, known erosion resistant materials are typically hard and brittle; as a result, they are not able to resist the mechanical stresses induced during turbine operation. Given these facts, the most suitable option, which seems practically viable, is the use of surface modifications. Several surface modifications can be used to alter the surface properties of existing structural materials to resist degradation due to erosion, whereas the underlying structural material can continue to provide the required mechanical strength.

5.1. PLASMA SPRAYING PROCESS

It involves spraying the coating material onto the substrate surface in the form of a powder or wire while using a plasma torch to heat it to a high temperature. Placing the material in a plasma flame melts it and generates a high temperature. Next, this molten substance approaches the substrate. The substance quickly cools when it contacts the substrate, leaving an adhesive coating behind. Plasma gases, including argon, nitrogen, hydrogen, etc., flow via the anodic nozzle around the cathode. The plasma starts as a neutral flame, meaning it has no charge. The electric arc descends the nozzle once the plasma is prepared to spray. Because the cold gas of the anode nozzle is electrically non-conducting and increases the plasma arc's velocity, it constricts the plasma arc. The powder is heated quickly and propelled toward the substrate as it is fed into the plasma flame, creating an extremely strong and durable coating

5.2. DETONATION SPRAYING THERMAL SPRAY COATINGS

The detonation gun, commonly known as the D-gun, is the main component of this process. A long, water-cooled barrel with an open end and a closed end makes up a D-gun. The coating material is delivered into the barrel together with oxygen and fuel gas, and this gas combination is ignited with a spark. The ensuing explosion warms and propels the powder into the barrel at supersonic speed. After every explosion, the barrel is additionally purged with a pulse of nitrogen. This procedure is repeated several times per second. The heated powder particles move toward the substrate's surface at a very fast speed of

roughly 600 m/s. Due to their fast speed, these particles have a high kinetic energy, which, when they collide with a substrate, creates an extremely thick and durable coating.

5.3. WIRE ARC SPRAYING

During this thermal spraying procedure, two continuously fed metal wires are introduced into the spray gun. These wires are electrically charged, and an arc is created between them that melts the incoming wire tips. The air jet originating from the gun then carries the molten metal away. Wire arc spraying is the procedure by which this molten material is applied to the substrate. This type of coating is often used for thick coatings [31, 32].

5.4. FLAME SPRAYING

As part of this process, the fuel gas is burned in order to melt the spray coating material that is applied to the substrate. This fuel gas is often composed of acetylene or propane that has oxygen added to it. Wire or powder versions of the coating material are often used in the manufacturing process. The material is heated before being propelled onto the substrate to generate a surface coating.

5.5. HIGH-VELOCITY OXY-FUEL (HVOF)

This coating process was developed during the 1980s. During this process, a combustion chamber is filled with a mixture of oxygen and fuel, which may be either gaseous or liquid. This mixture is continuously ignited, creating a steady flame. The heated gas that is produced then travels along a straight segment while being subjected to a pressure of one megapascal (MPa) via a nozzle that converges and diverges. The term ‘fuel’ may refer to either a liquid, such as kerosene, or a gas, such as hydrogen, methane, propane, or acetylene. The velocity of the exhaust jet emerging from the nozzle exceeds the local speed of sound, resulting in a supersonic gas stream. When injected into this gas stream, a powder feedstock has the potential to accelerate up to 800 m/s. The surface to be coated is approached and impinged by a stream of powder and hot gas at rapidly increasing temperatures. As the powder travels through the stream, it partially melts and then settles on the substrate. This particular coating has a low porosity and a high bond strength in contrast to other available coatings. There is a possibility that high-velocity oxy-fuel (HVOF) coatings, which are used for ceramic and metal coatings and are often utilized in operations to endure wear and corrosion, may be as thick as 12 mm. Tungsten carbide-cobalt (WC-Co), chromium carbide, alumina, and other powders are often used in the production of these coatings.

5.6. COLD SPRAY (CS)

The cold spray (CS) method uses a De Laval nozzle with a convergent-divergent geometry to accelerate solid powder particles to supersonic velocities. Speeds between 300 m/s and 1200 m/s are used for the ballistic impingement of particles onto a suitable substrate. The parameters of the feedstock powders, in addition to the geometry of the nozzle, affect the ultimate temperature and velocity of the sprayed particles. These factors are directly connected to the microstructure of the coating, as well as its physical and mechanical attributes. The temperature of the gas stream never exceeds the point at which the feedstock material begins to melt. Because it is carried out in the solid state, this coating deposition method has characteristics that are quite different when compared to other standard thermal spray processes. There are now two primary types of cold spray systems: high-pressure cold spray (HPCS), which injects particles from a high-pressure gas source before the spray nozzle throat, and low-pressure cold spray (LPCS), which injects powder into the diverging region of the spray nozzle. Both of these cold spray methods are referred to as cold sprays. The particle speeds that LPCS devices can produce are limited to the range of 300 m/s to 600 m/s, and these devices are often portable and much smaller. They are used to apply light materials, and air or nitrogen that is readily available serves as their standard propellant gas. In systems with high pressure, we use particles with higher density. They typically generate particle velocities between 800 m/s and 1400 m/s, are stationary, and employ higher-pressure gases. Lighter gases such as nitrogen or helium are selected as the HPCS propellant gas [33].

6. PROTECTION AGAINST EROSION

6.1. USE OF COATINGS

The damaging effects of erosion have been discussed in the preceding sections. In this part, we would like to elaborate on various protection schemes used for protection against degradation due to erosion. Erosion degradation can be reduced by either replacing the material with one that has greater erosion-resistant qualities, or by simply altering the surface of the components that are already in place. The drawback of the former approach is the high cost involved in replacing the material with a better one. Furthermore, property requirements may vary between the bulk and the surface. For instance, if the component were to be replaced with a harder material, it would reduce erosion. However, the component might lose certain mechanical properties, such as its toughness, ductility, and fracture resistance. With the use of the latter approach, however, we could certainly reduce cost and retain the desired bulk properties by modifying only the

surface properties to meet requirements. It has been shown that several emerging approaches for surface modification, such as cold spray, microwave-aided coatings, and friction stir processing, have demonstrated promising results and have the potential to replace some of the techniques currently in use.

6.2. WELD OVERLAY TECHNIQUE

It was pointed out by BONACORSO *et al.* [34] that weld overlays are the most preferred technique for repairing eroded components of hydroturbines. However, the presence of complex geometries and unfavorable positions during repair poses difficulties for weld adhesion to the treated surface.

Wear studies on 2.5C2.7Cr and 6.7Cr weld overlays produced by gas metal arc welding (GMAW) having hardness of 550 HV and 650 HV were conducted by DASGUPTA *et al.* [35]. It was observed that hard-faced steel performed better in comparison to the untreated steel specimens. A direct impact of the variation in the microstructure of the hard-faced overlays and steel on the erosion response and degrading mechanism was observed.

The slurry erosion of metal matrix composite (MMC) coatings developed using weld overlays was studied by FLORES *et al.* [36]. Using the plasma transferred arc method, coatings comprising 65 wt.%WC in Fe-Cr-C matrixes were developed. The effect of this addition was evident in the microhardness values, which reached 1140 HV compared to 670 HV for the layer formed without any addition. Clearly, the MMC weld overlay showed more than a twofold improvement in erosion resistance at 20 °C, mainly ascribed to the presence of solid parts. Further extending his studies, FLORES *et al.* [37] estimated the performance of a NiCrBSi matrix at high and low levels of alloying elements. Both matrixes were also reinforced with 65 wt.%WC. The NiCrBSi matrix containing higher alloying elements showed somewhat better erosion resistance than its counterpart. The addition of WC helped in improving the erosion performance by a factor of 4 to 5, however, somewhat better or slightly better performance was observed among the WC-reinforced matrixes.

HATTORI and MIKAMI [38] evaluated the comparative behavior of cavitation erosion of Stellite 6 and 21 weld overlays against SS304 and SS316 using a cavitating jet and ultrasonic rigs. It was observed that Stellite overlays showed lower material losses (7- to 13-fold reduction) compared to SS304 steel. A direct correlation between erosion and hardness was observed.

Cavitation erosion tests of NOREM, D-CAV, CaviTec, and Hydroloy 914 advanced weld overlays deposited by GMAW showed erosion in the range of 1 mg/h to 2.6 mg/h, compared to 3.2 mg/h for SS308 [39]. However, the data presented does not lead to a direct comparison because of the different densities of these materials.

GMAW and SMAW weld overlays of Co-alloyed stainless-steel coatings showed a 12 times lower mass loss rate relative to SS316L when tested under cavitating conditions [40]. The mass loss rate was even lower than that of HVOF-coated Stellite 6 steel by a factor of 5. However, no direct correlation with hardness was noticed, showing a dependence on microstructure.

The problem of cavitation-corrosion erosion of a copper-manganese-aluminum (CAM) alloy during its service in marine conditions was presented by LI *et al.* [41]. It was found that the welded zone showed 4 times higher resistance in comparison to the base material. The HAZ showed a higher erosion response than the weld zone owing to a lack of phase transformation. The benefits of NiTi steel in a cavitating environment were discussed by RICHMAN *et al.* [42]; however, its high cost and brittle behavior when formed into thick sections limit its practical usage. Two different alloys of NiTi (martensitic and austenitic) were explosively welded onto carbon steel at 0.56 mm to 0.68 mm thicknesses. The martensitic NiTi layers performed exceptionally well in comparison to their austenitic counterparts in the as-welded conditions. Heat treatment at 500 °C improved the response of the austenitic NiTi, showing a similar mass loss as to that of martensitic. Low-cycle fatigue showed a good correlation with the cavitation erosion behavior of various steels. A gas tungsten arc welding (GTAW) TiNi alloy subjected to slurry erosion studies showed a ductile response with maximum mass loss at a 30° impingement angle. An AISI 1048 alloy tested under similar conditions showed 50 % more mass loss at the highest level. This better performance was attributed to this alloy system's excellent work hardening and increased hardness, which resulted from the partial conversion of Ti into its oxides during the welding process [43]. KARIMI *et al.* [44] studied the cavitation response of NiCrBSi-based weld overlays produced by the flame spray method. It was found that the presence of a uniformly distributed austenitic hard phase led to the highest erosion resistance of the Ni60B alloy, being even higher than that of austenitic stainless steel. Finally, CrMnB overlays were tested under cavitation-corrosion conditions owing to their better martensitic transformation capabilities, promoting work hardenability [45].

6.3. THERMAL SPRAY COATINGS

According to the literature, surface coatings may help such materials resist slurry erosion. Thermal spraying techniques have recently garnered a lot of interest among many coating deposition techniques. The CE behavior of thermal spray coatings, however, must be assessed [46–51].

Although WC-based thermal spray coatings may provide substantial slurry erosion resistance [52, 53], there is little information on their CE behavior. In [54], the performance of flame-sprayed WC/Co-FeNiCr and Cr₂O₃ coatings was examined and it was discovered that microstructural flaws such porosity and

partially or completely unmelted particles had an impact on this performance. Although these coatings demonstrated greater resistance to slurry erosion, they did not perform well when exposed to CE [55]. Even WC/Co and CrC coatings sprayed using HVOF sustained substantial damage under cavitating erosion conditions [55]. The CE resistance of the bare steel was not increased by other WC coatings. Conversely, FeCrSiBMn coatings applied via HVOF were tested and found to outperform uncoated ZG06Cr13Ni5Mo martensite stainless steel. The inclusion of complex carbides and increased hardness of the coatings were credited for improving resistance [56–59].

The literature also reveals that the slurry and CE behavior of Ni-Al₂O₃-based coatings has not been thoroughly studied. Recent research by HU *et al.* [60] examined the effectiveness of cold-sprayed Ni + 40%Al₂O₃ coatings on Inconel-600 under slurry and CE. They demonstrated that, despite the coating's improved slurry erosion performance, its CE performance was insufficient. The Ni component lower cavitation resistance and lower bonding strength were shown to be the main causes of coating deterioration. Additionally, the coating's microhardness was only 1.5 to 2.5 times greater than that of the substrate. AHMED [61] discovered that contact fatigue loadings are mostly affected by the ratio of coating thickness to maximum shear stress depth. According to that research, this ratio has to be more than 1.5 for a longer operational lifespan [61].

NiTi has been shown to exhibit higher resistance against CE and wear, usually owing to its superelasticity and/or superplasticity properties [62–65]. It was reported that NiTi samples aged at 500 °C possessed 8.6 times higher resistance than those subjected to solution treatment [65]. However, the use of NiTi as a structural material is not a practically viable solution, owing to its high cost. Moreover, there could be a problem of brittleness in thick sections. RICHMAN *et al.* [42] recommended applying the NiTi coating onto steel components using an explosive cladding technique to protect them against CE [42]. This led to a reduction in the erosion rates of low-carbon steel of the order of 30 times in comparison to SS304 steel. Other conventional coating routes, such as fusion welding or thermal spraying, are not well suited to this composition. During fusion reactions, brittle phases are introduced, and the stoichiometry of NiTi is also affected due to uncontrolled reactions [42]. Similarly, thermal-sprayed NiTi coatings often exhibit low adherence to steel and lack the structural integrity to make them resistible against CE [66, 67]. Although around nine times higher CE resistance of NiTi coatings produced via tungsten inert gas (TIG) against that of SS316 steel has been reported [68], this resistance was still significantly lower in comparison to that of solution heat-treated NiTi bulk alloys. This low resistance of NiTi coating in comparison to bulk NiTi alloy was observed despite the high hardness of the former, which could be due to the loss of superelasticity during the coating process. Thermal-sprayed NiTi coatings also did not

show any further improvement due to the presence of pores acting as structural defects. Furthermore, the reaction of the feedstock powders with the surrounding environment (O_2 and N_2) results in the formation of brittle phases that eventually lead to the loss of erosion resistance [69]. Furthermore, variations in erosion rates with a Ni content in the coating was also reported, where coatings containing a Ni content higher than the equiatomic ratio showed higher resistance. After reviewing the relevant literature, it was discovered that a substantial amount of research has not yet been conducted to assess the performance of NiTi alloys and coatings with regard to slurry erosion and/or slurry erosion-corrosion [70].

6.4. NI-ALUMINA-BASED COATINGS

LUO *et al.* [71] evaluated the toughness and strength of the Ni- Al_2O_3 composites prepared using hot isostatic pressing (HIP) with Ni contents varying from 0% to 30%. When the amount of nickel in the composite increased, it was discovered that the strength of the composite decreased, while its structure toughness rose [72]. The maximum hardness reported was 175 HV1.0 for the composite coating containing 45 vol.% of the hard phase, whereas the coating with 40 vol.% Al_2O_3 exhibited the maximum hardness of around 650 HK0.1. This variance in hardness was ascribed to the average size of the reinforcing particles (coatings containing much smaller particles around 1 μm in size), along with a large grain size in the composite on the order of 1 mm. Moreover, the microhardness also increased to 524 HV with a rise in alumina content. Similar results have also been reported in [73, 74]. SZCZYGIEL and KOŁODZIEJ [75] also deposited Ni-6% Al_2O_3 coatings using an electrochemical deposition route and evaluated their corrosion performance. Furthermore, different coatings were investigated by several researchers [76, 79–83] to determine their microstructure and characteristics. Table 2 provides a review of the various coating materials used to combat erosion.

The coating was found to be successful in enhancing the slurry erosion resistance of Inconel 600. LI *et al.* [77] showed the feasibility of depositing Ni- Al_2O_3 coatings containing a large volume fraction of hard-phase constituents using a cold spray technique [76]. The maximum microhardness obtained was only around 173 ± 33 HV0.2, whereas the hardness of the pure Ni coating was found to be around 131 HV0.2. TURUNEN *et al.* [78] deposited nanocrystalline Ni Al_2O_3 . Recently, various researchers have employed distinct thermal spray coatings to combat corrosion, erosion, and other degradation mechanisms in different sectors, such as thermal power generation, hydroelectric power, gas turbines, aerospace engineering and others. Thus, thermal spray coatings have significant potential in the coming future.

TABLE 2. Summary of past research carried out on different coating materials for combating erosion [85–112].

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
1	[85]	Ni-SiC	SS316L	Microwave cladding	500–600	40 ±5	Hydropower plant	– Ni cladding without SiC powder showed a hardness of 360 HV. – The Ni-10 wt.% SiC cladding yielded the highest CE resistance and optimal mechanical results.
2	[86]	Ni and Ni-Inconel 718 powder	Duplex stainless steel	Cold spray	Up to 3000	16–45		– Achieved a 215HV hardness with the Ni-Inconel 718 coatings. – Observed an 80 % reduction in overall mass loss and higher CE resistance due to the addition of Inconel 718.
3	[87]	WOKA-3653 (86WC-10Co-4Cr)	AISI 410	HVOF	227	11–45		– Demonstrated a high slurry erosion resistance. – Tungsten carbide coatings exhibited a 50 % lower erosion rate at an impact angle of 30°.

TABLE 2. [Cont.]

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
4	[38]	WC-10Co-4Cr powder	SS 316	HVOF	302	25–40	Hydropower plant	– The modified PTFE shows high hardness varying from 255 to 290 HV. – Due to its high hardness, it has a better slurry erosion resistance.
5	[39]	Iron-based amorphous powder (SHS 574)	AISI 316	HVOF	180–200	15–53		– Observed lower erosive wear rate at impact angles of 30° and 45°. – Observed a higher erosive wear rate at 90°.
6	[90]	Ni-40TiO ₂ Ni-20TiO ₂ -20Al ₂ O ₃	13Cr-4Ni	HVAF	422 ± 10 354 ± 10	10–40		– Found lower porosity in the range of 0.8 % to 3.6 % for nanostructured composite coatings. – Found higher porosity in the Ni-20TiO₂-20Al₂O₃ coating than the Ni-40TiO₂ coating.
7	[91]	B4C KBF ₄ SiC	13Cr-4Ni	Pack boronizing method	40	–		– Found higher erosive wear resistance of the 13-4 martensitic stainless steel (MSS) compared to bare steel substrate due to coating high hardness (2300 VHN).

TABLE 2. [Cont.]

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
8	[92]	Ni-Cr-40Al ₂ O ₃	SS316L	HVFS	500-600	50-70 60-90 35-60	Hydropower plant	The microhardness of the Ni-Cr-40Al₂O₃ coating was 225 HV; however, after processing coatings it increased 2 times and increased fracture toughness.
9	[93]	Ni60 and WC-17Co	17-4PH stainless steel	APPS and HVOF	238-416	140-325 mesh 350-500 mesh		Volume loss was lowered by 50.6 % compared to the substrate material. WC-17Co is the best coating material for water droplet erosion (WDE) resistance.
10	[94]	Ni+20%Cr7C3	CA6NM	Microwave cladding	920-1010	100-150		Observed high erosion resistance, it was found that the average weight loss reduced by 45 % in clad samples compared to CA6NM.
11	[95]	WC-10Co-4Cr	AISI 410 stainless steel	HVOF	112-155	11-45		Improved the wear properties and high hardness (278 HV) were achieved with low coating porosity of 0.3 % to 0.7 % of the deposited coatings.

TABLE 2. [Cont.]

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
12	[96]	Cr ₃ C ₂ -20(80Ni20Cr) WC-Ni	ANSI 1018	HVOF	221 ±16 118 ±8	10-45 15-45	Turbine steel	– Observed a 20.5 % improvement in hardness, i.e., increasing from 846 HV to 1020 HV. – Observed the lowest porosity of about 1 % for Cr ₃ Cr ₂ -NiCr cermet coatings.
		Cr ₃ C ₂ -25NiCr WC-17Co WC-12Co WC-10Co-4Cr Cr ₃ C ₂ -40NiCr	1.4571 high-alloyed-steel	HVOF	100-200	20-40		
		Conventional WC-12Co Nanostructured WC-12Co CeO ₂ modified WC-12Co	Stainless steel AISI 304	HVOF	–	15-45 5-45 15-45		
		WC-CoCr Cr ₃ C ₂ -NiCr Al ₂ O ₃	S355 steel		163 176 294	15-45 15-45 5-22		
		WC-10Cr-4Co	1Cr18Ni9Ti stainless steel	HVOF HVOF	380-420	16-45 5-30		
13	[97]	Cr ₃ C ₂ -NiCr Ni-Cr-Fe Cr ₃ C ₂ +25NiCr	1018 steel 13Cr4Ni	APS Plasma spraying	200 150 220 ±15	10-90 15-45		– Achieved coating micro-hardness values up to 1050 VHN coatings. – Decreased cumulative mass loss compared to the bare reference steel.

TABLE 2. [Cont.]

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
14	[98]	TiAlN coatings	AA1050 AA5083	HVOF	105–150	2–50 2–70	Hydropower plant	– Observed erosion at 160 g/min, 195 g/min, and 265 g/min at impingement angles of 75°, 60°, and 30°, respectively.
15	[99]	Al ₂ O ₃ -13%TiO ₂ and Cr ₃ C ₂ -25NiCr	CA6NM	D-gun	–	5–30 15–45		– Found lower erosion in Cr ₃ C ₂ -25NiCr coating than the Al ₂ O ₃ -13%TiO ₂ . – Observed cumulative weight loss range from 0.1 mg to 1.5 mg in Cr ₃ C ₂ -25NiCr coating compared to 3.5 mg to 8.5 mg in Al ₂ O ₃ -13%TiO ₂ coating.
16	[100]	WC-CoCr	SS410	HVOF HVOF	380–420	1.2		– Found excellent hardness, fracture toughness, and surface residual stress in WC-CoCr coatings.
17	[101]	86WC-10Co4Cr	SS410	HVOF HVOF	410	16–45 20–45		– Achieved hardness value up to 1473 HV0.3 with lower porosity (0.3% to 0.9%). – Observed lower weight loss due to cavitation.
18	[102]	Al 6061 powder	ASTM C633	HPCS	2000–8000	5–68	Al-6061 hydraulic valve actuator	– Recorded an increase in hardness accompanied by low porosity varying from 0.93% to 1.66%.

TABLE 2. [Cont.]

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
19	[103]	WC-Co-Cr and Ni-Cr-B-Si	13Cr-4Ni	Plasma thermal spray	250–400	5–50	Hydropower plant	– Mass loss increases with increasing particle in velocity. – Recorded cumulative mass loss at 15 mg/cm², 12 mg/cm², 25 mg/cm² in WC-Co-Cr, Ni-Cr-B-Si coatings and for bare steel, respectively.
20	[104]	Stellite-6 and Cr ₃ C ₂ -25Ni4Cr	13Cr-4Ni	D-gun	100–150	10–53 10–45		– Observed overall specific weight loss of 1897.43 g/m² and 345.31 g/m² in 13Cr4Ni and Stellite-6 coatings, respectively, after a 6-hour test run. – Observed a 5.5 times higher specific weight loss in 13Cr4Ni than in the Stellite-6.
21	[105]	Ni-Al ₂ O ₃	CA6NM	HVFS	24–684	20–100 20–60	Turbine steel	– Recorded high erosion resistance for coatings containing 40 wt.% alumina. Uncoated steel showed a 2.2 higher erosion rate.

TABLE 2. [Cont.]

No.	Reference	Coating materials	Substrate	Process	Coating thickness [μm]	Average particles size [μm]	Applications	Remarks/findings
22	[106]	WC-10Co-4Cr	CF8M turbine steel	D-gun	200	15–45	Turbine steel	- Found a reduction in the erosion rate of WC-10Co-4Cr coatings. - Achieved high hardness of 1120HV with low porosity of 1 %.
23	[107]	WC-10Co-4Cr and Al ₂ O ₃ +13TiO ₂	CF8M turbine steel	HVOF	830–1160	15–45 10–45		Found high erosion resistance in WC-10Co-4Cr coatings compared to the Al₂O₃+13TiO₂.
24	[108]	Wallex-50 and Triballoy-700 powders	16Cr-5Ni	Laser surface alloying	600–1000	200	Hydropower plant	Found excellent ductility and superior impact toughness even at low temperatures.
25	[109]	Co-based Stellite 6 powders and Ni-based Colmonoy 88	13Cr-4Ni steel	Laser surface alloying (LSA)	300–400	100		Observed a 1.250 to 2-fold improvement in hardness relative to the base substrate material.
26	[110]	Ti-6Al-4V 17Cr-4Ni PH Steel 12Cr steel	ASTM G73-98	HVOF	250 ±30	5–50	Steam turbine blades	Achieved high surface hardness and a low erosion rate with HVOF coating.
27	[111]	WC-12Co Cr ₃ C ₂ +25NiCr	13Cr-4Ni and T410	D-gun	50–250	5–45	Turbine components	Concluded that the degradation rate of hard coatings depends upon the erodent particle size and total concentration.
28	[112]	Ni-Cr-B-Si composite coating, WC+Ni-Cr-B-Si	Stainless steel 18-8	Plasma spraying process	500 ±5	5–50	Hydropower plant	Reached a peak surface hardness of 1150HV along with a 0.73 mg/10h CE.

7. CONCLUSIONS

Hydroelectric power plants in the Himalayan region and other high-sediment mountainous environments face persistent hydro-abrasive erosion, affecting nearly 40 % of installations worldwide. This review synthesizes three decades of multidisciplinary research into a unified framework for understanding and mitigating silt-induced turbine degradation, highlighting that erosion severity is governed by the combined effects of suspended sediment characteristics, hydraulic operating conditions, and turbine material properties. Field observations indicate that unprotected CA6NM components can suffer erosion rates of 0.5 mm to 3 mm per year, with trailing-edge damage exceeding tip erosion by three to four times. Over time, three complementary mitigation strategies have emerged, including upstream sediment management, advanced protective coatings, and computational fluid dynamics (CFD)-based turbine design optimization, each offering measurable reductions in erosion and associated maintenance costs. Despite these advances, significant research gaps remain, including large site-to-site variability in erosion calibration despite IEC 62364:2019 standardization, limited understanding of coating durability under transient and seasonal operating conditions, insufficient studies on Kaplan turbine erosion in run-of-river schemes, early-stage integration of real-time monitoring with machine-learning-based prediction, and the absence of comprehensive life-cycle assessments comparing mitigation pathways. Available evidence shows that integrated approaches combining moderate sediment reduction, design optimization, and selective coating application can reduce material loss by 40 % to 60 %, lower overall costs by 25 % to 35 %, and extend maintenance intervals by 2 to 3 times. Implementing sediment-aware operational strategies, real-time monitoring, site-specific erosion assessment, cost-optimized coatings, and upstream sediment control can collectively reduce erosion losses by 30 % to 50 %, thereby extending component lifespans from 3–7 years to 12–20 years, and significantly enhancing the long-term sustainability of hydropower systems worldwide.

7.1. FUTURE RESEARCH DIRECTIVES

Future research directives derived from this study are as follows:

- investigate and optimize thermal spraying techniques (air plasma, D-gun, cold spraying, HVOF) to improve the erosion-corrosion resistance of hydroturbine blade steels,
- focus on processing methods to achieve defect-free thermal spray coatings, thereby minimizing porosity, micro-cracks, through-pores, and enhancing coating hardness,

- research rare earth and nanostructured powder materials to optimize coating material combinations for improved performance in hydroturbine applications,
- study the incorporation of solid lubricants in thermal spray coatings to assess their impact on wear reduction and friction, ensuring cost-effectiveness for specific applications,
- investigate the effects of post-spray heat treatment on the microstructure and bond strength of thermal spray coatings to enhance their corrosion and wear resistance,
- research techniques to further harden thermal spray coatings and reduce their porosity to increase erosion resistance in practical applications.

The abovementioned research directives highlight key areas for future exploration and development in improving thermal spray coatings for hydropower applications.

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CONFLICT OF INTEREST

The author declares that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHOR'S CONTRIBUTION

The author conceptualized the study, performed the analysis, wrote the original draft, and approved the final manuscript.

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Technical Note

Exploring the Synergistic Influences of Clay Nanoplatelets and Waste Eggshell Particles on PMMA-Based Hybrid Nanocomposites

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This study investigates the impact of incorporating clay nanoplatelets and waste eggshell particles into poly(methyl methacrylate) (PMMA) to develop hybrid nanocomposites with improved mechanical and structural properties. Waste eggshells, rich in biogenic calcium carbonate, were utilized as a sustainable, low-cost filler, while clay nanoplatelets provided nanoscale reinforcement and improved interfacial bonding. The hybrid nanocomposites were prepared at a filler content of 5 % (hereafter designated as 5 wt.%) using the solution casting method. Scanning electron microscopy (SEM) was employed to examine the dispersion and morphology of the fillers within the structure of the PMMA matrix. At the same time, Fourier-transform infrared spectroscopy (FTIR) was utilized to assess the powder chemical interactions and potential bonding between the fillers (clay nanoplatelets and eggshell particles) and the PMMA matrix. The mechanical behavior of the PMMA and the produced hybrid nanocomposites was also evaluated through tensile testing. The results indicate a synergistic enhancement due to the combination of organic biowaste and inorganic nanoclay, offering a promising route for the development of sustainable, high-performance polymer nanocomposites.

Keywords: clay nanoplatelets, waste eggshell particles, PMMA, hybrid nanocomposites, tensile properties.



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1. INTRODUCTION

The development of polymer-based nanocomposites has garnered considerable attention in recent years due to their improved mechanical and thermal

properties compared to conventional polymer materials. Among polymer matrices, poly(methyl methacrylate) (PMMA) is an acrylic polymer and the most well-known member of the methacrylate family [1, 2]. It is widely used in different applications due to its excellent properties, such as low weight, low density, highly durable polymer providing excellent biocompatibility, high transparency, and great aesthetic properties [3–5]. Such properties make PMMA-based composite materials highly suitable for various biomedical applications such as intraocular lenses, orthopedic devices, removable dentures, and bone tissue engineering. Moreover, PMMA and related linear acrylic polymers have also been utilized in other applications such as lightweight armor, electronics fabrication, transportation components, and glass replacement [4, 6]. Nevertheless, PMMA resins demonstrate weak surface characteristics and poor mechanical specifications, including a lack of toughness, low flexural and impact strengths [4, 5, 7]. Hence, the PMMA polymer matrices require improvement to achieve greater mechanical performance.

Different approaches have been investigated to enhance and expand the range of applications of PMMA polymers, including the development of composites. Consequently, PMMA polymer matrices reinforced by natural nanoparticles, metal oxide nanofillers, carbon-based nanofillers, and synthetic or natural fibers have been developed [8, 9]. Among these fillers, clay nanoplatelets such as montmorillonite (MMT), provide high aspect ratios and surface areas, offering effective stress transfer and enhanced interfacial bonding within polymer matrices [10]. MMT is one of the most available forms of nanoclay, which comprises thin layers with an internal octahedral layer interposed between two tetrasilicate layers [11, 12]. Such a structure is reported to inhibit crack formation [13]. Accordingly, several investigations have confirmed that the utilization of clay nanoplatelets (utilized as fillers) enhances the mechanical performance, thermal stability, and dimensional integrity of various composite materials [14, 15].

On the other hand, biologically derived waste materials have recently received considerable attention as sustainable fillers in various polymer composites due to their low cost, eco-friendliness, and wide availability. Among these fillers, eggshell waste has emerged as a potential reinforcing additive because it consists primarily of inorganic calcium carbonate (95 %) in the form of calcite along with organic type X collagen (5 %) and sulfated polysaccharide components [16, 17].

This high content of calcium carbonate makes eggshell powder functionally similar to traditional mineral fillers used in polymer composites [18]. Therefore, several investigations have reported the effective incorporation of eggshell as a reinforcing material into various polymer matrices, demonstrating enhancements in structural and mechanical properties [19–21]. Although eggshell is a biological material whose composition may vary slightly depending on different

factors, such as the animal's diet and living conditions [22], appropriate processing steps – including cleaning, drying, and grinding into fine powder – can significantly enhance reproducibility and reduce variability. Consequently, the use of waste eggshell-derived fillers not only provides a sustainable approach for waste valorization but also offers a viable alternative to traditional inorganic fillers in polymer composite applications.

The combination of bio-derived fillers, such as waste eggshell particles, and inorganic nanoplatelets, such as clay, presents an innovative approach to fabricating sustainable hybrid nanocomposites [23]. Based on the existing literature, several studies have investigated the impact of clay nanoplatelets on the properties of PMMA-based polymer nanocomposites [13, 24]. Furthermore, other studies have evaluated the impacts of eggshell particles on the properties of PMMA polymer composites [25, 26]. However, none of these studies has addressed the reinforcement of PMMA polymer matrices with waste eggshell particles and clay nanoplatelets as dual fillers.

Therefore, this study aims to explore the synergistic effects of combining clay nanoplatelets and waste eggshell particles in a PMMA matrix, investigating how these fillers influence the morphological, chemical, and mechanical properties of the resulting hybrid nanocomposites. This research seeks to contribute to the advancement of green materials by integrating sustainable waste materials with nanotechnology to provide enhanced functionality for potential industrial applications. For this purpose, PMMA hybrid nanocomposites were fabricated by incorporating waste eggshell particles and clay nanoplatelets as hybrid fillers into PMMA. The morphological properties of the tensile fractured samples, as well as the filler distribution in the PMMA matrix, were examined using scanning electron microscopy (SEM). In addition, the chemical interactions between the waste eggshell particles and clay nanoplatelets were investigated using Fourier transform infrared (FTIR) spectroscopy. Moreover, the tensile properties (tensile strength and Young's modulus) of the pure PMMA and the fabricated nanocomposites were investigated.

2. MATERIALS AND METHODS

2.1. MATERIALS

PMMA was used as the polymer matrix and supplied by Orthocryl-Dentaurum, Germany. MMT-based nanoclay and eggshell particles were used in this study as the reinforcement materials. The MMT nanoplatelets were purchased from Sigma-Aldrich, UK. These nanoplatelets were originally modified with approximately 30% octadecylamine and were used in composites as received. The eggshell particles were prepared according to the procedure described in the following subsection.

2.2. PREPARATION OF EGGSHELL PARTICLES

Brown waste eggshells were collected from household waste. First, the thin inner membrane was removed as much as possible, as it is proteinaceous and may interfere with the polymer. Next, the shells were washed thoroughly with warm, soapy water to remove grease and egg residue. Thereafter, the washed shells were boiled in hot water for 15 min. Then, the cleaned shells were dried in an electric oven at 60 °C for 180 min. After that, the dried shells were crashed by hand into small pieces and then ground into a powder using a high-speed blender. For particle sizes less than 75 μm , a 200-mesh sieve was used.

2.3. FABRICATION OF HYBRID NANOCOMPOSITES

The solution casting method has been performed to fabricate PMMA hybrid nanocomposites. Specimens of pure PMMA were first cast by mixing the resin with a hardener at a weight ratio of 100:2. Thereafter, specimens of PMMA/5 wt.% of clay nanoplatelets were fabricated as follows. First, pure PMMA resin was stirred in a beaker using a mechanical stirrer (WiseStir/HS-50A). While stirring, the clay nanoplatelets were slowly added to the PMMA, and the mixture was stirred continuously for additional 5 min at a speed of 500 r/min. Thereafter, the mixture was sonicated using an ultrasonic probe sonicator (Misonix Sonicator 3000) for 3 min to improve the homogenization of the mixture. After the proper dispersion of the clay nanoplatelets in the PMMA matrix, the hardener was added at a ratio of 100:2. The solution was then poured into a tensile mold (an aluminum mold according to the ASTM D638, supplementary Fig. A1, see Appendix). The same procedure was followed to fabricate the PMMA specimens reinforced with 5 wt.% eggshell particles. However, for the hybrid nanocomposites, a total filler mass fraction of 5 % was achieved by mixing equal mass fractions (2.5 % clay nanoplatelets and 2.5 % eggshell particles), and then adding them to the matrix via the same procedure of mixing and sonification. Hereafter, the produced specimens (shown in supplementary Fig. A2, see Appendix) are referred to as PMMA, PMMA/C, PMMA/E, and PMMA/C/E.

2.4. MORPHOLOGICAL OBSERVATIONS

The surface morphology of the tensile fractured samples (PMMA, PMMA/C, PMMA/E, and PMMA/C/E) was examined using SEM (Tescan VEGA3). In order to eliminate the electrostatic surface charging effect, the specimens were coated with Au. Thereafter, the specimens were mounted onto an aluminum stud using double-sided carbon tape. The images were collected using an XFlash 410 M detector with a beam deceleration of 200 V.

2.5. FTIR SPECTROSCOPY

In order to investigate the chemical interactions between eggshell particles and clay nanoplatelets, FTIR was conducted. A PerkinElmer Frontier spectrometer (Waltham, MA, USA) with a single reflection diamond attenuated total reflectance (ATR) accessory was used. The analysis was conducted over a wavenumber range of 4000 cm^{-1} to 500 cm^{-1} at an angle of incidence of 16° from the perpendicular. For each specimen, a total of 8 scans were averaged at a spectral resolution of 2 cm^{-1} .

2.6. TENSILE TESTING

Tensile properties (strength and modulus) of the pure PMMA polymer and its hybrid nanocomposites were tested using an HST WDW-100E 100 KN computer-controlled electromechanical universal testing machine. The test was performed according to the ASTM D638 standard at a constant crosshead displacement rate of 40 mm/min . The load-displacement values were recorded during the test, and the recorded values were used to calculate the ultimate tensile strength and Young's modulus properties of the produced samples. The tensile strength of each specimen was determined from the maximum stress obtained prior to fracture, while Young's modulus was calculated from the slope of the initial linear elastic region of the stress-strain curve. The tests were performed at room temperature ($22^\circ\text{C} \pm 3^\circ\text{C}$). Three tests were carried out for each type of specimen, and the average values were reported.

3. RESULTS AND DISCUSSION

3.1. FRACTURE SURFACE MORPHOLOGY

The surface morphology of each tensile fractured sample was observed using SEM, and the representative images are presented in Fig. 1. The tensile fracture morphology of pure PMMA exhibits a brittle failure mode with some cracks, sharp edges, and also exhibits a rapid crack structure, as shown in Fig. 1a. This confirms the nature of PMMA, which is intrinsically brittle [27]. However, after the addition of 5 wt.% clay nanoplatelets into the PMMA (Fig. 1b), the micrograph exhibited a comparatively rougher surface with varying morphological features. For instance, relatively uniform and flat surfaces are observed in some zones (Fig. 1b), whereas in others, sharp edges and rough surfaces are observed. Furthermore, the SEM micrograph in Fig. 1b shows some agglomerated clay nanoplatelets (highlighted by arrows) on the fractured surface. The high surface area of nanoplatelets and the strong attractive forces between them results in their agglomeration [28]. This agglomeration can reduce the potential im-

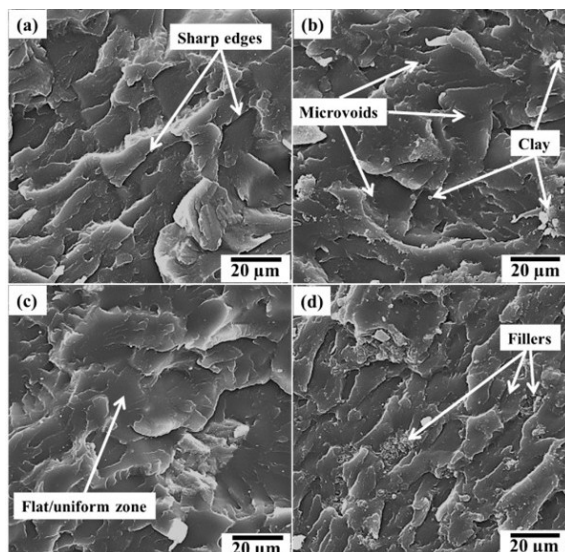


FIG. 1. SEM micrographs of: a) pure PMMA, b) PMMA/C, c) PMMA/E, and d) PMMA/C/E.

provement of mechanical properties (see [Subsec. 3.3](#)) due to a reduction in the interfacial area. Moreover, [Fig. 1b](#) shows some microvoids (indicated by arrows), which may be attributed to the pull-out of the agglomerated clay nanoplatelets. A similar fracture surface morphology was also observed for that of PMMA/E, as shown in [Fig. 1c](#). However, after the addition of both fillers (clay nanoplates and eggshell particles) to PMMA, the SEM micrograph ([Fig. 1d](#)) shows a rougher surface with sharper edges than those of PMMA, PMMA/C, and PMMA/E. Although, agglomerated particles are also observed in [Fig. 1d](#), which may be related to either or both filler types, a more uniform dispersion of fillers is observed ([Fig. 1d](#), indicated by arrows) compared to the single-filler composites (PMMA/C and PMMA/E), when dual fillers were used. This may be attributed to the proper mixing of the clay and eggshell particles before adding them to PMMA. Moreover, eggshell particles could break down any agglomerated clay nanoplatelets during mixing, thereby enhancing the dispersion of nanoclay in the PMMA matrix. This dispersion contributes effectively to the improvement of the tensile properties of the produced hybrid nanocomposites (discussed in [Subsec. 3.3](#)).

3.2. CHEMICAL ANALYSIS BY FTIR

To further understand the actual interaction between the clay nanoplatelets and eggshell particles, FTIR absorption spectra of the powder samples were collected. [Figure 2](#) displays the collected FTIR spectra of the clay nanoplatelets, eggshell particles, and the mixture of both fillers (clay and eggshell). The FTIR

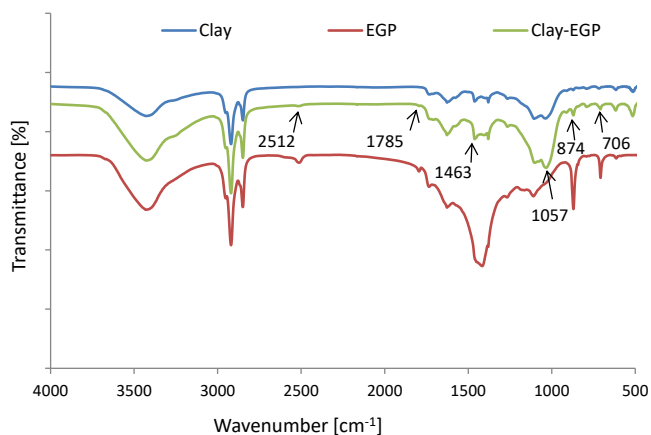


FIG. 2. FTIR spectra of clay nanoplatelets, eggshell particles, and clay/eggshell particles.

spectrum of the clay nanoplatelets in Fig. 2 shows absorption bands that are in good agreement with those presented in previous studies [29]. In Fig. 2, the spectrum of clay nanoplatelets shows characteristic peaks at 3462 cm^{-1} and 1659 cm^{-1} , which corresponds to the O-H stretching vibration of the clay silicate interlayers [29]. The characteristic bands at 1122 cm^{-1} , 1057 cm^{-1} , and 874 cm^{-1} can be ascribed to Si-O stretching vibrations of the clay nanoplatelets [30].

The FTIR spectrum of both fillers (clay nanoplatelets and eggshell particles) is presented in Fig. 2. The spectrum displays an almost identical pattern to that of clay nanoplatelets, except for some new bands indicated by arrows in Fig. 2. For instance, a characteristic band is observed at a wavenumber of 2512 cm^{-1} in the hybrid spectrum. In contrast, there was no absorption peak at this wavenumber range in the clay spectrum. However, this absorption band is clearly visible in the FTIR spectrum of eggshell (Fig. 2), which is attributed to the existence of a hydrogen group in the eggshell structure [20]. Therefore, the observed peak (2512 cm^{-1}) in the spectrum of clay and eggshell could be assigned to the presence of eggshells. Another shoulder peak is observed in the spectrum of both fillers (clay and eggshell) at about 1831 cm^{-1} . This peak could have resulted from the carbonyl group that existed in the eggshell structure [20]. Moreover, the spectrum of clay and eggshell showed two peaks at 1463 cm^{-1} and 1057 cm^{-1} that are not present in the spectrum of eggshell, whereas they exist in the spectrum of clay. The observed peak (1463 cm^{-1}) in the spectrum of clay is attributed to the CH_2 in-plane scissoring vibrations [31], whereas the band at 1057 cm^{-1} is assigned to Si-O stretching vibrations of clay nanoplatelets [30]. Furthermore, the spectrum of clay and eggshell in Fig. 2 also shows two absorption peaks at 874 cm^{-1} and 706 cm^{-1} . Compared to the spectra of clay and eggshell in Fig. 2, these two bands appear as very weak broad bands in the clay spectrum, while they were clearly visible in the eggshell spectrum.

The existence of these two bands (874 cm^{-1} and 706 cm^{-1}) in the spectrum of eggshell is associated with the out-of-plane deformation and in-plane deformation, respectively, which indicates the existence of calcium carbonate (CaCO_3) in the eggshell structure [32]. However, in the spectrum of clay nanoplatelets, the absorption band at 874 cm^{-1} is related to the bending vibration of O-H groups [33], whereas the band at 706 cm^{-1} is attributed to the CH_2 vibration absorption [27]. Therefore, from the FTIR results, it can be concluded that the clay nanoplatelets are successfully grafted onto the surface of eggshell particles. Thus, this confirms the improved mechanical properties of the PMMA/C/E hybrid nanocomposites, which showed higher tensile strength and modulus than those of the single-filler composites (see Subsec. 3.3).

3.3. TENSILE PROPERTIES

The tensile properties (strength and modulus) of pure PMMA and its hybrid nanocomposites (PMMA/C, PMMA/E, and PMMA/C/E) are presented in Fig. 3. Moreover, the detailed results of the tensile tests, including the tensile stress, elongation at break, and Young's modulus for all tested samples, with standard errors, are presented in supplementary Table A1. It can be clearly observed from Fig. 3 that the PMMA/C nanocomposites demonstrate lower tensile strength than the pure PMMA. The average tensile strength of the pure PMMA is $30.722\text{ MPa} \pm 0.54\text{ MPa}$. With the addition of 5 wt.% of clay nanoplatelets to the PMMA-based polymer, the average tensile strength decreased to $20.011\text{ MPa} \pm 1.41\text{ MPa}$ (Fig. 3). This can be attributed to the nature of the PMMA matrix, which is brittle to some extent [6], and the addition of clay nanoplatelets could provide further brittleness characteristics in the amorphous matrix [31]. Furthermore, the observed results (Fig. 3) could also be attributed to the interfacial interaction between the exfoliated clay nanoplatelets and the PMMA matrix. Materials in the size range of less than 100 nm usu-

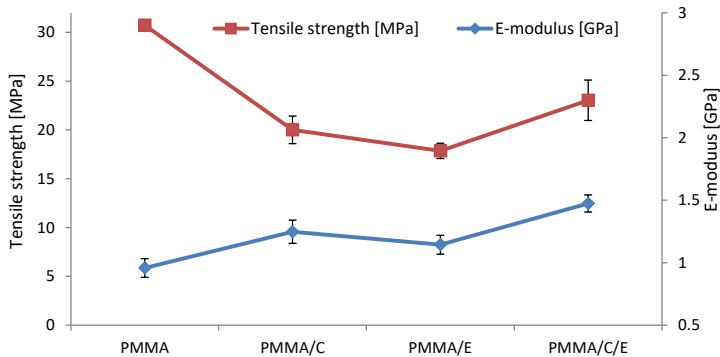


FIG. 3. Comparison of tensile strength and Young's modulus of pure PMMA, PMMA/C, PMMA/E, and PMMA/C/E, obtained from tensile stress-strain curves.

ally tend to cluster or agglomerate, because of the dominant intermolecular van der Waals interactions between them [34]. This results in poor interfacial interaction between the clay and the PMMA matrix, consequently decreasing effective stress transfer within the polymer nanocomposite and weakening the mechanical properties [35]. It is also observed from Fig. 3 that the incorporation of eggshell particles into the PMMA matrix leads to a further decrease in tensile strength ($17.857 \text{ MPa} \pm 0.78 \text{ MPa}$) when compared with those of PMMA and PMMA/C. This is in agreement with other studies that recorded a decrease in tensile strength when eggshell particles were incorporated into epoxy polymer matrices [36]. This drop in tensile strength might be related to the stress concentration effect produced by eggshell particles in the PMMA matrix [37]. The overall deformation behavior and mechanical response of these materials can be further observed in the stress-strain curves presented in supplementary Fig. A3 (see Appendix). However, another work reported an increase in tensile strength when smaller particle sizes ($0.2 \mu\text{m}$) of eggshells were incorporated into a polyvinyl chloride matrix [38]. Such an increase may be attributed to the size of eggshell particles used as filler, as well as the processing technique.

On the other hand, the incorporation of both fillers (clay nanoplatelets and eggshell particles) into the PMMA polymer to produce hybrid nanocomposites shows an increase in tensile strength ($23.043 \text{ MPa} \pm 2.07 \text{ MPa}$) when compared with the single-filler composites (PMMA/C and PMMA/E), as shown in Fig. 3. Generally, the micron-sized particles in hybrid composites carry most of the load, while the nano-sized fillers fill the spaces around the micron-sized particles, improving the interfacial bonding between the polymer and the micron-sized particles and reducing voids or weak regions [39, 40]. Therefore, the achieved tensile strength of PMMA/C/E in Fig. 3 could be attributed to the inclusion of eggshell particles and clay nanoplatelets as hybrid fillers in the structure of PMMA. A similar trend was also reported, where the mechanical properties of the hybrid silica/epoxy composites show an improvement in tensile strength compared to single-filler composites [39].

It is noteworthy to mention that, although the tensile strength is reduced after the inclusion of clay nanoplatelets/ eggshell particles into PMMA, the modulus is constantly increased compared with that of pure PMMA, as presented in Fig. 3. For example, the average tensile modulus of the pure PMMA is $0.958 \text{ GPa} \pm 0.075 \text{ GPa}$. With the addition of clay nanoplatelets (5 wt.%) to the PMMA, the average tensile modulus was increased to $1.248 \text{ GPa} \pm 0.093 \text{ GPa}$, as shown in Fig. 3. This could be attributed to the high modulus properties of clay nanoplatelets when they are dispersed at the nanoscale level in polymer matrices [41]. In contrast, the inclusion of eggshell particles into the PMMA matrix shows a lower tensile modulus ($1.144 \text{ GPa} \pm 0.075 \text{ GPa}$) than that of PMMA/C, as presented in Fig. 3. The size of eggshell particles significantly affects the final

mechanical properties of the produced composites [42]. The decrease in modulus after the incorporation of eggshell particles can be attributed to the large particle size of the eggshells, which acts as a stress concentration. However, it is important to mention that the incorporation of both fillers (clay nanoplatelets and eggshell particles) into the PMMA matrix increases the modulus to a value of $1.474 \text{ GPa} \pm 0.068 \text{ GPa}$. This promising result could be attributed to the uniform distribution of clay nanoplatelets and eggshell particles in the PMMA polymer matrix, as confirmed by the SEM observations in [Subsec. 3.1](#).

4. CONCLUSION

The incorporation of clay nanoplatelets and eggshell particles significantly influenced the structural, chemical, and tensile properties of PMMA hybrid nanocomposites. The structural analysis confirmed a more homogeneous distribution of both fillers within the PMMA polymer matrix, indicating good interfacial bonding and improved composite integrity when compared to those of single-filler composites. FTIR results revealed noticeable chemical interactions between the clay nanoplatelets and eggshell particles, suggesting the formation of a stable hybrid network. Moreover, the introduction of clay nanoplatelets and eggshell particles resulted in a decrease in tensile strength compared with pure PMMA; nevertheless, Young’s modulus increased, indicating a positive contribution of the fillers to the stiffness of the polymer composite matrix. This could be attributed to the synergistic reinforcing effect of the clay nanoplatelets and eggshell particles. Overall, the study demonstrates that combining inorganic clay with bio-derived eggshell fillers offers an effective and sustainable strategy to enhance the mechanical performance and stability of PMMA hybrid nanocomposites.

APPENDIX



FIG. A1. Tensile mold made of aluminum.

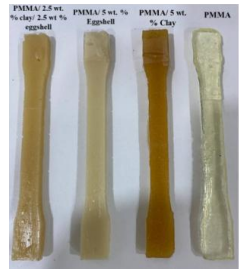
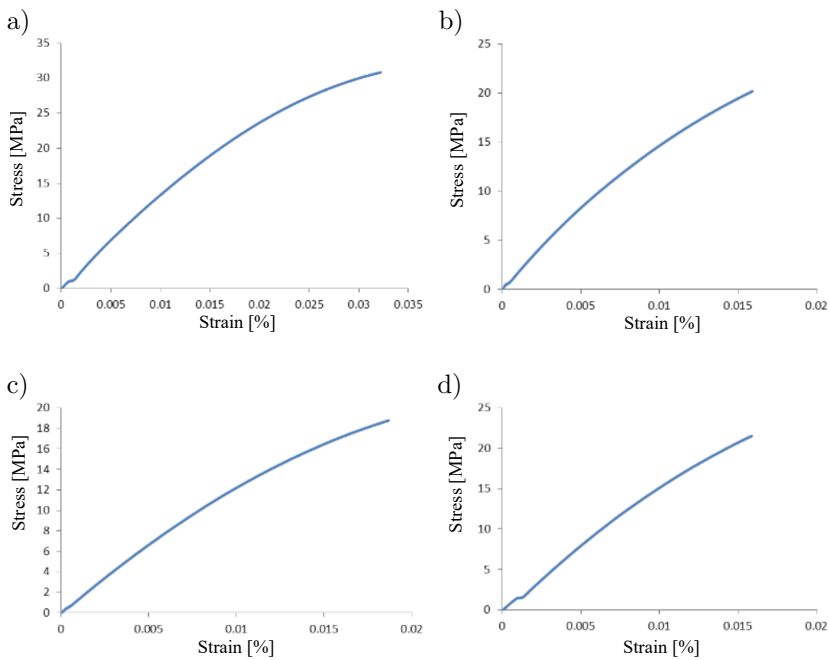


FIG. A2. Fabricated specimens of pure PMMA and its composites.

TABLE A1. Tensile stress, Young's modulus, and elongation at break for all tested samples.

	Tensile strength [MPa]	Young's modulus [GPa]	Elongation at break [%]
PMMA			
Sample 1	31.309	0.993	3.4
Sample 2	30.000	0.853	3.7
Sample 3	30.857	1.028	3.2
PMMA/5 wt.% clay			
Sample 1	20.178	1.327	1.6
Sample 2	18.200	1.300	2.2
Sample 3	21.657	1.116	1.4
PMMA/5 wt.% eggshell			
Sample 1	18.777	1.062	1.8
Sample 2	17.931	1.243	1.9
Sample 3	16.863	1.127	1.4
PMMA/2.5 wt.% clay/2.5 wt.% eggshell			
Sample 1	25.972	1.570	1.5
Sample 2	21.645	1.427	1.3
Sample 3	21.512	1.424	1.4

FIG. A3. Stress-strain curves of:
a) pure PMMA, b) PMMA/C, c) PMMA/E, d) PMMA/C/E.

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CONFLICT OF INTEREST

The authors declare that there are no known competing financial interests or personal relationships that could have influenced the work described in this paper.

AUTHORS' CONTRIBUTIONS

All authors contributed to the study conception, methodology, data analysis, and manuscript drafting, and read and approved the final manuscript.

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