

Topology optimization and thermal modeling of wire arc additive manufacturing for formula SAE front upright

Nathan Eskinder Alemayehu¹, Mesay Alemu Tolcha¹

¹Faculty of Mechanical Engineering, Jimma Institute of Technology, Jimma University, Jimma, Ethiopia

E-mail: nathan.eskinder@ju.edu.et (N.E. Alemayehu), mesay.alemu@ju.edu.et (M.A. Tolcha)

Abstract. Topology optimization enables the development of lightweight and high-performance components, although the resulting complex geometries are often difficult to manufacture using conventional methods. Wire Arc Additive Manufacturing (WAAM) provides an effective solution due to its high deposition rate, material efficiency, and capability to fabricate large metallic structures. However, the thermal history during WAAM significantly affects temperature distribution, residual stresses, distortion, and component quality. This study presents an integrated computational framework combining Solid Isotropic Material with Penalization (SIMP)-based topology optimization, transient thermal finite element analysis, and process parameter optimization for WAAM. Topology optimization was performed in Tosca Structure to minimize structural volume while satisfying stress constraints, and the optimal design was selected using multi-criteria decision analysis. The selected geometry was analyzed in Abaqus using Goldak's double-ellipsoidal heat source model to simulate thermal evolution during WAAM. Layer height and heat source parameters were optimized using the Evolutionary Optimization Algorithm in Isight, and the optimized model was evaluated at deposition travel speeds of 480, 600, and 750 mm/min. Increasing travel speed reduced peak temperatures and reheating cycles, improving thermal stability. The proposed framework provides a systematic design-to-manufacturing approach for producing lightweight WAAM-fabricated metallic components with enhanced manufacturability and process reliability.

1 Introduction

Modern automotive design increasingly seeks to balance efficient material usage with the need to maintain adequate strength and stiffness in lightweight components [1]. This objective can be addressed through topology optimization which enables lightweight design without altering the material itself, as performance requirements can be explicitly defined through optimization objectives and constraints. However, the optimized geometries that result from such approaches are often highly complex and difficult to manufacture using conventional techniques [2]. This complexity necessitates advanced manufacturing methods capable of producing intricate shapes with minimal defects and material waste. Among these, wire arc additive manufacturing (WAAM) has emerged as a particularly suitable solution [3].

The concept in integrating topology optimization and WAAM can be implemented in several automotive components. One such component is the FSAE upright (steering knuckle). The upright is the main structural component of the wheel assembly which supports the axle bearings, the brake calliper and transmits all the forces from the wheel to the suspension members [4]. The steering and cornering responsiveness of the vehicle can be increased by reducing the weight of the wheel assembly components and hence the upright.

Several studies followed different approaches to the topology optimization problem. The material distribution that minimizes the compliance (or maximizes the stiffness) of the upright is one of the objective functions with mass constraint. Ghungarde et al.[5] related a bionic shape with that of steering knuckle using topology optimization as a tool for getting optimum material layout with an objective function of maximum stiffness with minimum mass constraints of 30%. Gunev and Iliev [6] applied a SIMP based topology optimization to maximize the stiffness of the steering knuckle with single and multiple load cases. Objective function of topology optimization can also constitute a volume or mass minimization with stress constraints. Various literature have followed this approach. Kamble et al. [7] utilized the SIMP method with an objective of volume minimization and the constraint being von Mises stress. The result of the optimization exhibited a reduction of weight by 9.859%. Dhinakaran et al. [8] carried out the topology optimization with an objective to minimize weight setting the constraints on maximizing the stiffness and the results show that a 8.35% mass reduction.

A macro-scale thermal model of WAAM, coupled with finite element analysis (FEA), is often employed to efficiently predict the temperature field during and after material deposition [9]. Some literature focused on the integration of topology optimization with different types of metal additive manufacturing. Dalpadulo et al. [10] studied sequential thermo-mechanical modeling on Powder Bed Fabrication of topologically optimized upright using AlSi10Mg alloy as part material. The study in general focused on the integration of computer aided design (CAD) with optimization and additive manufacturing simulation. V. Pavlos [11] integrated topology optimization and selective laser melting (SLM) thermo-mechanical analysis for a FSAE upright. Different printing orientations and support types were comparatively studied to observe the effects on the upright.

In this study Solid Isotropic Material with Penalization (SIMP)-based topology optimization, transient thermal finite element analysis, and process parameter optimization for WAAM is performed on FSAE upright. The remainder of this paper is structured as follows. Section 2 presents the material and methods used in the study. Section 3 provides the results of the topology optimization and thermal FEA of WAAM. Section 4 discuss the results and validates them against benchmark studies. Section 5 summarizes with concluding remarks.

2 Topology optimization of upright

2.1 Material selection

The material selection criteria differ from those of a simple axial members as the upright experiences multiaxial and cyclic loading. The criteria are: *specific stiffness* (E/ρ) which controls deflection under cornering/braking loads, *specific strength* (σ_y/ρ) - determines resistance to short-duration overload events such as curb strikes and impacts, *fatigue resistance* in which the upright is subjected to millions of loading cycles, and *WAAM compatibility* - central to enabling the topology-optimized geometry. Figure 1 shows candidate materials against the criteria.

The specific-modulus values for all conventional wrought metals cluster near 25 GPa · cm³/g, a well-known result on the Ashby chart [12] — meaning stiffness-per-mass alone rarely differentiates Ti, steel, and Al alloys. Inconel718 trails the other metals slightly on specific stiffness due to its higher density.

A weighted scoring matrix (Table 1) ranks candidates against criteria weighted by relevance to a safety-critical, multiaxially loaded upright. For each material, the weighted score is:

$$S = \sum_i w_i \times r_i$$

where w_i is the criterion's weight (as a decimal fraction, e.g. 25% = 0.25) and r_i is the material's rating on that criterion (1–5 scale).

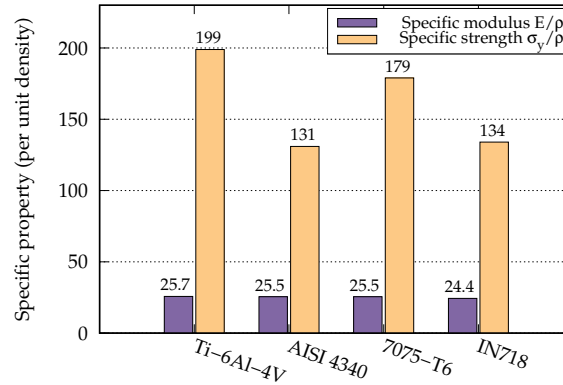


Figure 1: Comparative properties of candidate Upright materials

Table 1: Weighted decision matrix for upright material selection

Criterion (weight)	Ti-6Al-4V	AISI 4340	7075-T6	IN718
Specific strength (25%)	4	3	4	3
Fatigue life (25%)	4	4	2	4
AM/WAAM compatibility (20%)	4	3	2	5
Toughness/ductility (15%)	4	4	2	4
High-temp stability (15%)	3	2	1	5
Weighted score	3.85	3.25	2.35	4.1

The material selected for this study is Inconel-718, a precipitation-hardenable nickel-based superalloy widely recognized for its high strength, excellent corrosion resistance, and superior mechanical stability at elevated temperatures [13].

2.2 Structural finite element analysis

The initial geometry (Figure 2) is analyzed under the governing load cases to establish stress distribution, displacement, and factor of safety providing the reference stiffness and stress state used later to evaluate improvement.

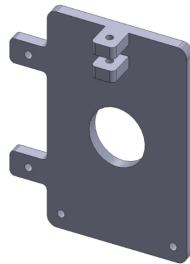


Figure 2: Initial Geometry

The bearing loads applied on the front upright are at the maximum value in the case of braking than in cornering [4]. Hence the equations for the maximum braking will be used. Taking moments about the

center of the inner bearing:

$$\begin{aligned} V_{outer} &= \frac{W_{vert} \times (l_1 + l_2)}{l_1} \\ H_{outer} &= \frac{W_{long} \times (l_1 + l_2)}{l_1} \end{aligned} \quad (1)$$

where, W_{vert} is vertical load and W_{long} is Longitudinal load (kN), l_1 is the hub length, and l_2 is the distance between the hub and center line of the wheel. Summing vertical and horizontal forces, we find:

$$\begin{aligned} V_{inner} &= V_{outer} - W_{vert} \\ H_{inner} &= H_{outer} - W_{long} \end{aligned} \quad (2)$$

All of the torque is resisted by the brake calliper with an assumption that the wheel is on the point of locking under braking. The force will act at the center of area of the brake pad:

$$F_{brake} = \frac{W_{long} \times R_r}{(l_3 + l_4)} \quad V_{brake} = \frac{F_{brake}}{2} \quad H_{brake} = \frac{F_{brake} \times l_4}{l_5} \quad (3)$$

where, R_r is rolling radius, l_3 and is the distance between the center of the hub and center of points where the brake calliper is mounted, l_4 is the distance between the center of the brake mounting points and the center of the calliper itself, and l_5 is the distance between the two mounting points.

Based on the 2022-2023 Electric Vehicle Senior Design Class's designs (published in [14]), the loads applied and the measurements of lengths are given as follows in Table 2.

Table 2: Parameters for maximum braking [14]

Parameter	Value	Unit	Parameter	Value	Unit
W_{vert}	2634.54	N	R_r	266.7	mm
W_{long}	3161.44	N	l_1	83.64	mm
			l_2	8.09	mm
			l_3	76.78	mm
			l_4	28.88	mm
			l_5	95.40	mm

By substituting the values of the parameters in Table 2 into equations 1-3, the following results in Table 3 have been obtained.

Table 3: Parameters for maximum braking [14]

Location	Boundary Condition	Load (N)
RP-1	-	$V_{brake} + (-H_{brake})$
RP-2	-	$V_{brake} + H_{brake}$
RP-3	Fixed	-
RP-4	Fixed	-
RP-5	Fixed	-
RP-6	-	$V_{outer} + (-V_{inner}) + H_{outer} + (-H_{inner})$

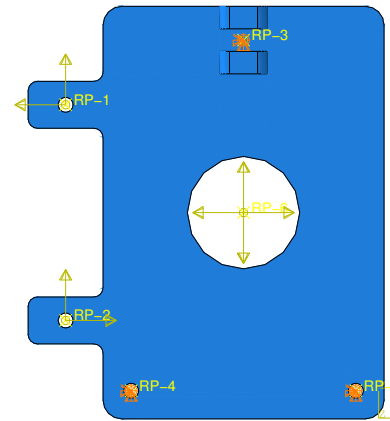


Figure 3: Boundary conditions and loads

A "static, linear perturbation" analysis is performed in Abaqus/CAE 2024 to compute the linearized structural response to small disturbances about a previously established equilibrium configuration [15]. The mesh is done with tetrahedral element type (C3D10). The result of the analysis is depicted as follows in Figure 4.

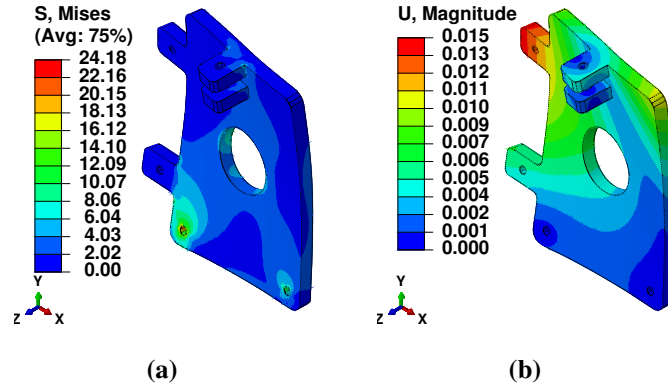


Figure 4: Structural FEA results: (a) Von Mises stress (b) deformation

From the FEA results, it can be shown that the stress is maximum the fixed point (RP4) with a value of 28.42 MPa, while the maximum displacement is at loading point (RP-1) with a value of 0.0146 mm. The design can be validated against the Yield Stress of the material (Inconel-718) which is between 1000 - 1200 MPa [16]. the value of the maximum stress calculated with the FEA is well below the Yield stress. Furthermore, the factor of safety can be calculated as $\sigma_{yield}/\sigma_{max} = 1200/28.42 = 43.6$, which indicates the predicted stress is far below yielding under the applied load case.

2.3 Topology optimization

The topology optimization in this research is implemented with Tosca Structure , which is a modular system for non-parametric structural optimization [15]. Structural optimization using Tosca Structure is an iterative process in which the structural response of the component is evaluated in every design cycle by an external finite element (FE) solver. For this study, Abaqus is used as the FE solver.

The topology optimization problem is formulated as:

$$\begin{aligned}
 \min_{\rho} \quad & V(\rho) = \sum_{e=1}^N \rho_e v_e \\
 \text{s.t.} \quad & \mathbf{K}(\rho)\mathbf{u} = \mathbf{F} \\
 & \max_{e \in \Omega_{nf}} \sigma_{vm}^{(e)} \leq 284 \text{ MPa} \\
 & \rho_e = 1, \quad \forall e \in \Omega_f \\
 & g_{\text{overhang}}(\rho, d, \theta_{max}) \leq 35 \\
 & 0 \leq \rho_e \leq 1, \quad e = 1, 2, \dots, N
 \end{aligned} \tag{4}$$

The objective function minimizes total structural volume $V(\rho)$. The optimized structure must satisfy static equilibrium $\mathbf{K}(\rho)\mathbf{u} = \mathbf{F}$. The von Mises stress in all active design elements must remain below the allowable stress $\sigma_{vm}^{(e)} \leq 284 \text{ MPa}$ that prevents yielding. Table 4 show the cases for the constraints used in the study. [Constraint values were determined iteratively, starting from the yield stress of Inconel 718 \(\$\approx 1000 \text{ MPa}\$ \) and reduced in 100 MPa increments until a viable geometry was obtained at 300 MPa.](#)

Higher values left nearly the entire design domain void ($\rho = 0$). Furthermore, values below 250 MPa, produced no significant volume reduction. An approximate midpoint (≈ 278 MPa) of this range (250 - 300 MPa) was therefore adopted, along with two additional constraint values, giving three cases in total (Table 4). This iterative selection is necessitated by the dependence of the optimal constraint value on material properties and applied loads, which would yield different values under different conditions.

Table 4: Optimization constraints

Case	Stress Constraints (MPa)
Case 1	284
Case 2	280
Case 3	278

To ensure manufacturability during the optimization process, a *overhang* constraint is imposed on the design domain. This constraint enforces that all structural features are self-supporting with respect to a specified pull direction, defined here as $\mathbf{d} = (0, 0, 1)$. The admissible geometry must satisfy an angular limitation such that the local overhang angle does not exceed 35° , i.e., $g_{\text{overhang}}(\rho) \leq 35^\circ$. This prevents the formation of unsupported features and ensures feasible part removal during manufacturing.

The Solid Isotropic Material with Penalization (**SIMP**) method interpolates material stiffness using a power-law function:

$$E_e(\rho_e) = E_{\min} + \rho_e^p(E_0 - E_{\min}) \quad (5)$$

The corresponding element and global stiffness matrix becomes:

$$\begin{aligned} \mathbf{K}_e(\rho_e) &= [E_{\min} + \rho_e^p(E_0 - E_{\min})] \mathbf{K}_e^0 \\ \mathbf{K}(\rho) &= \sum_{e=1}^N \mathbf{K}_e(\rho_e) \end{aligned} \quad (6)$$

The parameter p penalizes intermediate densities. Typical values are: $p = 3$ to 5 . Higher values force the design toward black-and-white solutions (solid or void). The value of p used in this study is 3.

The density filtering method used is the general sensitivity filter, not a Helmholtz-type filter. The filtered density $\tilde{\rho}(\mathbf{x})$ at each point is computed as a weighted average of the densities in a neighborhood defined by a filter radius $r_{\min}$:

$$\tilde{\rho}_i = \frac{\sum_{e=1}^n H_{ie} \rho_e}{\sum_{e=1}^n H_{ie}}, \quad H_{ie} = \max(0, r_{\min} - \|\mathbf{x}_e - \mathbf{x}_i\|) \quad (7)$$

where $r_{\min}$ is the filter radius and $\mathbf{x}$ denotes element centroids.

The relative material volumes of elements of the structure are updated using the method of moving asymptotes (MMA).

2.4 Results - topology optimization

The SIMP based topology optimization of the upright is solved using Tosca Structure over 30 iterations. The objective function (minimize volume) in reduced percentage form and the factor of safety based on distortion energy theory is shown in Table 5.

Figure 5 show the relative material distribution and smoothed surface for each constraint cases. A static linear perturbation FEA for the post smoothed geometries is done with Abaqus. The mesh used for all the geometries are of quadratic tetrahedral elements of type C3D10. Figure 6 show the results.

Table 5: Volume reduction results

Case	Minimized Volume	Factor of safety
Case 1	42.61 %	7.3
Case 2	42.62 %	10.6
Case 3	41.58 %	8.7

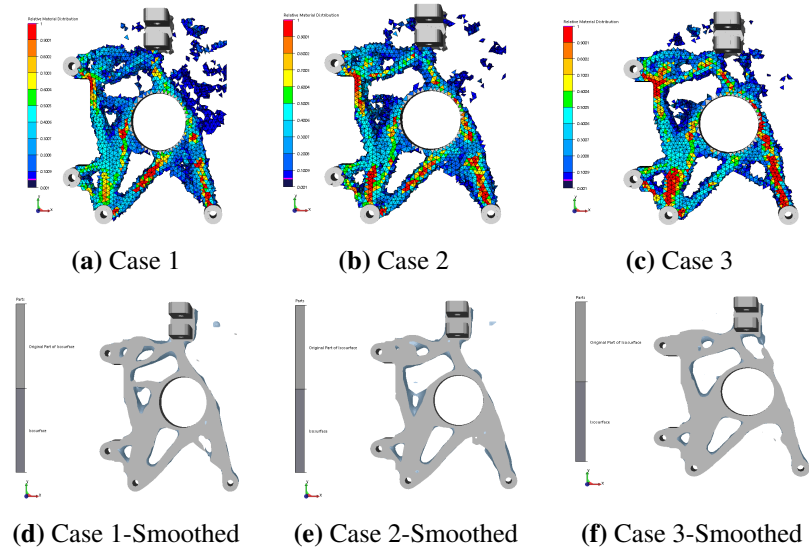


Figure 5: Relative material distribution and smoothed surfaces

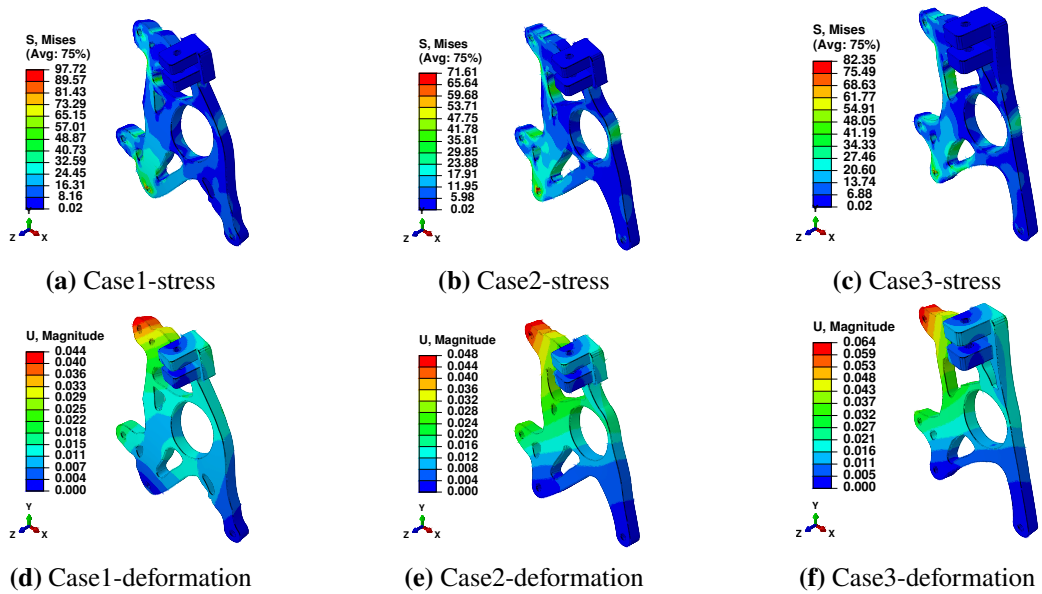


Figure 6: Structural FEA on optimized uprights

To systematically evaluate and rank the three upright optimized designs, a weighted scoring Multi-Criteria Decision Analysis (MCDA) [17] was implemented. The optimized upright are assigned as A, B and C across six criteria: maximum von Mises stress (0.30), maximum displacement (0.20), volume reduction (0.15), safety factor (0.15), aesthetics (0.10), and manufacturability (0.10). Figure 7 shows the results of the MCDA in which the candidate optimized upright C (Case 3) is selected.

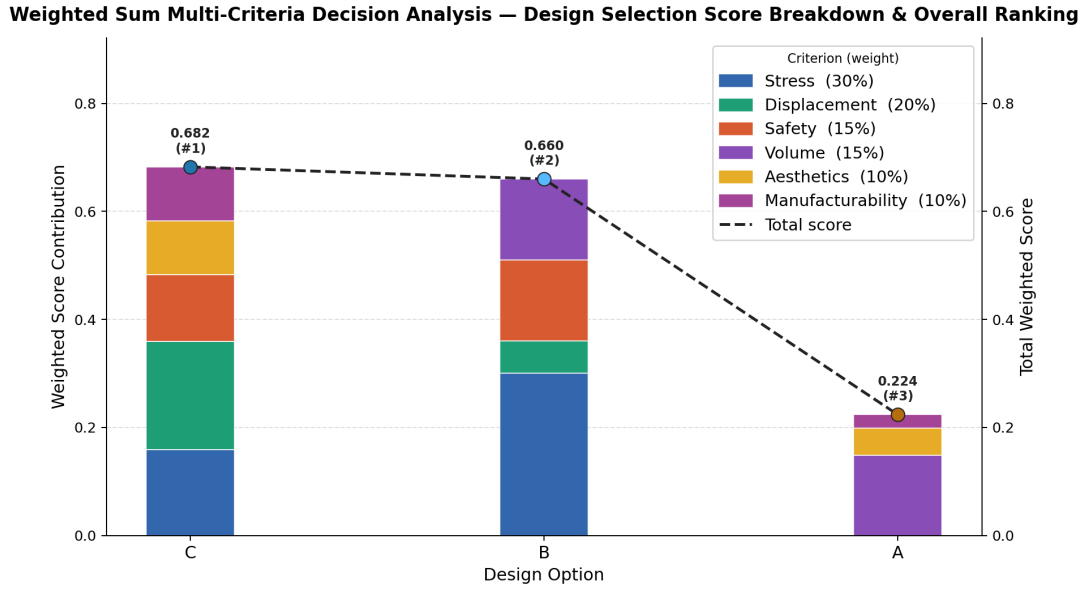


Figure 7: Multi-Criteria Decision Analysis (MCDA) design selection

The comparison of stress and deformation for the initial and selected optimized uprights shows that the maximum von Mises stress for the initial and optimized Uprights are 28.42 MPa and 94.88 MPa, respectively. The maximum displacement result are 0.015 mm and 0.084 mm, respectively for the initial and optimized Uprights.

Weight/volume reduction alone is insufficient if achieved at the expense of stiffness in which excessive compliance in the upright under cornering and braking loads introduces unintended toe and camber change (compliance steer), degrading handling predictability. The **stiffness-to-weight ratio** is therefore used as the governing comparison parameter, since it captures whether mass removed through topology optimization corresponds to structurally inefficient material or comes at a disproportionate cost to stiffness [18]. It is calculated by dividing the weight to the stiffness of the structure as follows:

$$\frac{k}{W} = \frac{k}{m \cdot g} \quad (8)$$

where k is the stiffness (N/mm), m is the part mass (Kg) and $g = 9.81 \text{ m/s}^2$.

The mass of the initial and optimized uprights are 3.27 kg and 1.5 kg, respectively. The stiffness is calculated as the fraction of maximum values of applied load and displacement (F/u) from the FEA results with which the stiffness of the initial and optimized uprights respectively are 1.16 and 0.52 ($\times 10^6 \text{ N/mm}$). Hence, with the values of mass, gravity and stiffness, the stiffness-to-weight ratio is calculated and the results are 3.6 and 3.57 ($\times 10^4 \text{ mm}^{-1}$), respectively for the initial and optimized uprights.

Comparing the values of the ratio, it can be shown that the optimized Upright has an approximately 1% lower value than the initial one, which indicates that the mass/weight has been effectively reduced while maintaining a stiffness-to-weight ratio relatively close to that of the initial Upright.

3 Parametric optimization of WAAM

3.1 Parametric process optimization

The parametric optimization for the WAAM parameters is done with non-gradient based optimization using Isight. The parameters to be optimized are layer height and Goldak's heat source parameters. The optimization problem is defined as:

$$\begin{aligned} \min_{\mathbf{x}} \quad & T_{\max}(\mathbf{x}) \\ \text{s.t.} \quad & 1 \leq \text{Layer Height} \leq 4 \\ & 2 \leq b \leq 6, \quad 1 \leq c \leq 4 \\ & 3 \leq a_f \leq 7, \quad 7 \leq a_r \leq 11 \end{aligned} \quad (9)$$

where, b is the lateral semi-axis, c is depth semi-axis, a_f is front semi-axis in travel direction and a_r rear semi-axis in travel direction (all in mm).

The optimization problem is solved with the *Evol Optimization Algorithm*. The Evol algorithm is an evolutionary optimization method derived from the foundational work of Rechenberg and Schwefel [19]. It improves design solutions by introducing mutations to the design variables using normally distributed random perturbations. The optimized parameters are then used in another thermal modeling of WAAM with three travel speeds to investigate temperature history over the material deposition.

3.2 Results - parametric process optimization

Initial thermal modeling of WAAM is carried out before the parametric optimization. The analysis is run with 600 mm/min and layer height of 2 mm. Using the results of this analysis, the optimization is run and 24 different results of the objective and constraints were found. The values of the best optimized results of the parameters are: Layer height = 4 mm; Lateral semi-axis, $b = 2$ mm; Depth semi-axis, $c = 1$ mm; Front semi-axis, $a_f = 5$ mm and Rear semi-axis, $a_r = 11$ mm. With these values the transient thermal FEA is run by varying three travel speeds: 480 mm/min (TS1), 600 mm/min (TS2) and 750 mm/min (TS3).

4 Thermal modeling of WAAM

4.1 Modeling approach

Transient thermal analysis of WAAM is implemented with heat transfer analysis using Abaqus AM Modeler plugin and scripting method. Progressive material deposition is modeled by utilizing progressive element activation. Before the analysis begin, all elements of the part to be fabricated are deactivated. During the beginning of each deposition a new element is activated progressively by a material deposition sequence defined in an *event series* (ES) data that defines fields that vary with time and space (x , y and z axes). However, if the geometry of the part to be fabricated is complex, the event series becomes complicated. Hence, an alternative way is needed to acquire the data and one of which is from gcode. However, the syntax of a gcode and event series are quite different. A program that converts the gcode to ES data is needed. An open-source python code called Additive Manufacturing Process Event Series Generator (AMPES) by Failla Jr. and Nguyen [20] is used for the conversion purpose. In this way, the upright is sliced with 10 layers and each layer is deposited with 0 dwell time.

The heat transfer analysis solves a transient heat conduction equation where the temperature field $T(\mathbf{x}, t)$ in the solid domain Ω satisfies:

$$\rho c_p \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T) + \dot{Q}_{\text{vol}} \quad \text{in } \Omega \times (0, t_{\text{end}}] \quad (10)$$

where, ρ is the mass density [kg/mm^3], c_p is the specific heat capacity [$\text{mW/mm} \cdot \text{K}$], k is the thermal conductivity [$\text{mW/mm} \cdot \text{K}$], and $\dot{Q}_{\text{vol}}$ is the volumetric heat source [mW/mm^3].

The progressive heating of the deposited material is related with a moving source which is modeled with the well known Goldak's Double-Ellipsoid model [21]. The double-ellipsoid domain is split into a *front* half-ellipsoid (ahead of the arc) and a *rear* half-ellipsoid (behind the arc). In a coordinate system (ξ, η, ζ) moving with the heat source (where ξ is along the travel direction), the *volumetric power density* is:

Front half-ellipsoid ($\xi \geq 0$)

$$\dot{q}_f(\xi, \eta, \zeta) = \frac{6\sqrt{3} f_f Q}{a_f b c \pi \sqrt{\pi}} \exp\left(-\frac{3\xi^2}{a_f^2} - \frac{3\eta^2}{b^2} - \frac{3\zeta^2}{c^2}\right) \quad (11)$$

Rear half-ellipsoid ($\xi < 0$)

$$\dot{q}_r(\xi, \eta, \zeta) = \frac{6\sqrt{3} f_r Q}{a_r b c \pi \sqrt{\pi}} \exp\left(-\frac{3\xi^2}{a_r^2} - \frac{3\eta^2}{b^2} - \frac{3\zeta^2}{c^2}\right) \quad (12)$$

where, $Q = \eta_{\text{abs}} P_{\text{arc}} = \eta_{\text{abs}} (\text{VI})$ is the total absorbed power [mW], a_f is the front semi-axis in the travel direction [mm], a_r is the rear semi-axis in the travel direction [mm], b is the lateral semi-axis [mm], c is the depth semi-axis [mm], f_f and f_r are the fractions of heat deposited in the front and rear ellipsoids, respectively.

Thermal boundary conditions include heat input from the deposition source, convection to the surrounding environment, radiation losses, and heat conduction through the substrate. Following boundary condition definition, the model is discretized through meshing with tetrahedral element type of DC3D4. The mesh is set finer in regions of high thermal gradients and deposition paths where higher accuracy is required, while coarser elements were used in less critical regions to improve computational efficiency (the far edges of the substrate). Figure 8 shows the results.

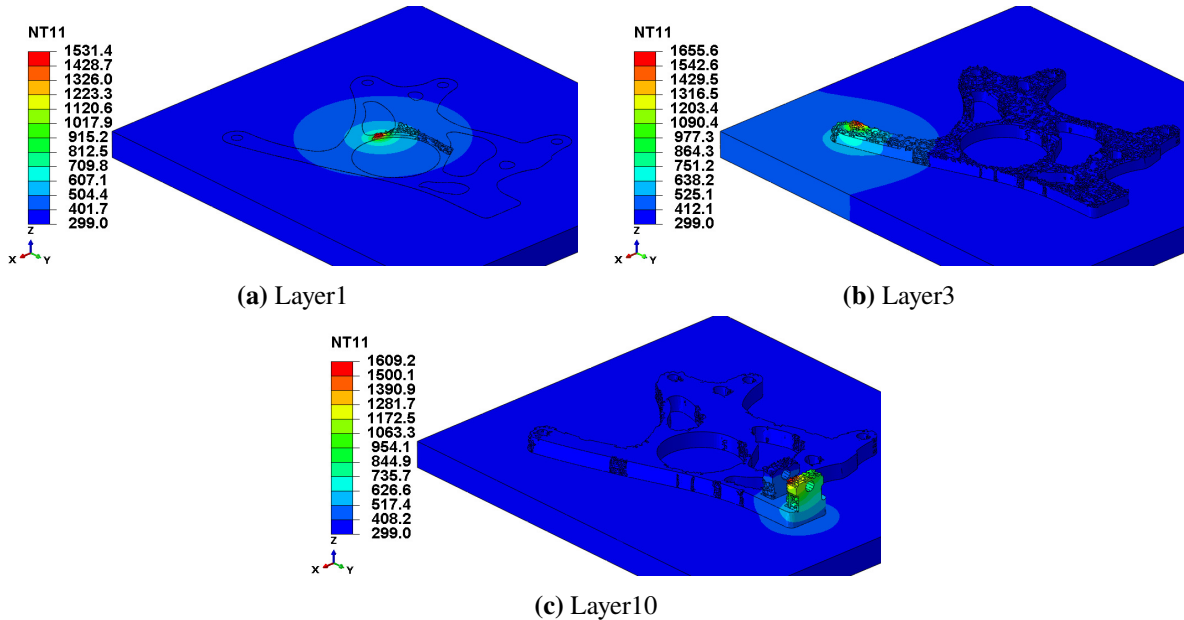


Figure 8: Temperature histories

4.2 Results - temperature histories

A comparative study of the thermal histories of three models by varying three travel speeds are studied by observing the histories over the first layer, third layer and the last (10th) layer. These layers are important

as they represent the parts of the upright, the first and third layers consist of the central hub and mounting brackets, while the last layer is in the external steering arm mounting. Across all the three models with the varying travel speeds, three monitoring nodes in the first deposition of each layer are selected. Figure 9 shows the temperature histories.

Layer 1 is activated earliest and experiences the greatest number of thermal cycles than the other layers. It is located in the lower, earlier-deposited region of the build and is repeatedly reheated by subsequent layer depositions. The maximum temperature reached are 1664.31, 1623.48 and 1671.91 K, respectively for TS1, TS2 and TS3

Layer 3 activates during the mid-build stage and exhibits fewer (than layer 1) but higher-amplitude thermal cycles. The maximum temperature reached are 1626.40, 1619.12 and 1667.22 K, respectively for TS1, TS2 and TS3.

Layer 10 activates near the end of the build and shows only one or two significant thermal peaks, representing the near-top region with minimal subsequent reheating. As a result of the accumulated thermal history of the preceding nine layers, this layer has the highest peak deposition temperature reaching as much as 1726.33 K, 1716.16 K and 1682.84 K for TS1, TS2 and TS3 respectively.

A clear trend is observed as deposition speed increases from 480 $\rightarrow$ 750 mm/min. The total process duration shortens, peak reheating temperatures reduces, and the number of thermal reheat spikes slightly decreases. These behaviours are consistent with lower nonlinear heat input per unit length at higher deposition speeds.

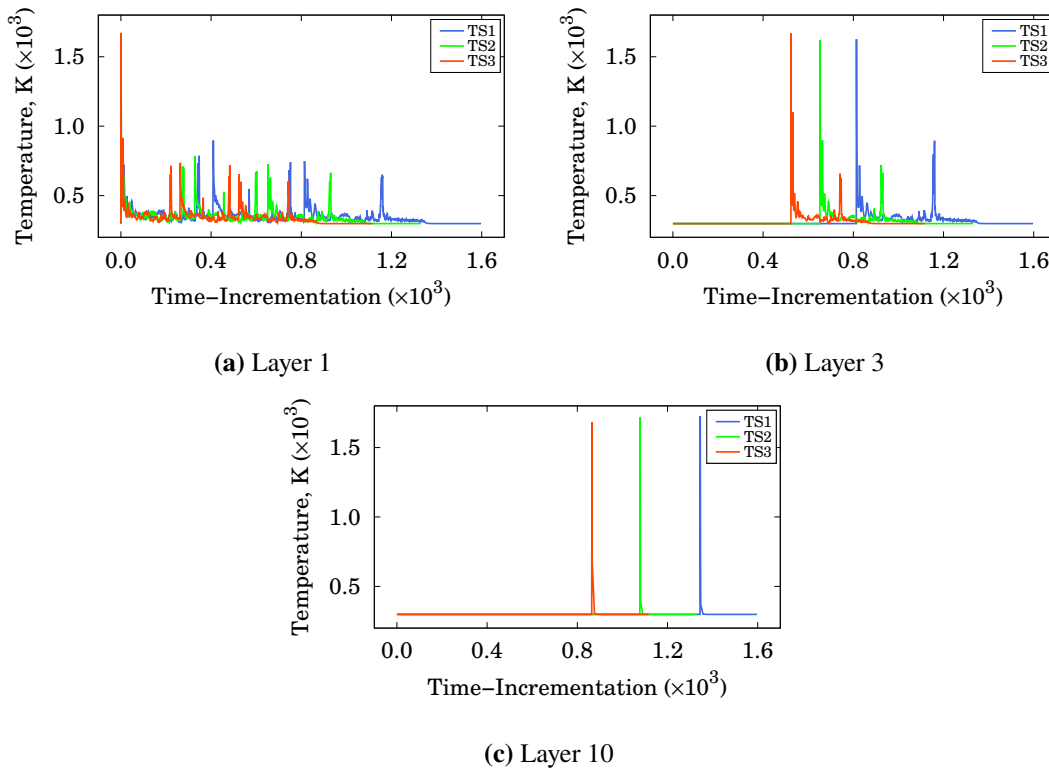


Figure 9: Temperature history of deposited upright at studied layers

The present thermal studies are validated against the experimental data reported by Zhang et. al [22] with cooling rate as the comparison metric. Figures 10 present the cooling rate as a function of time increment for the three travel speeds—TS1, TS2 and TS3. The cooling rate is expressed in units of K/s. Heating rate is related with it as the inverse of cooling rate.

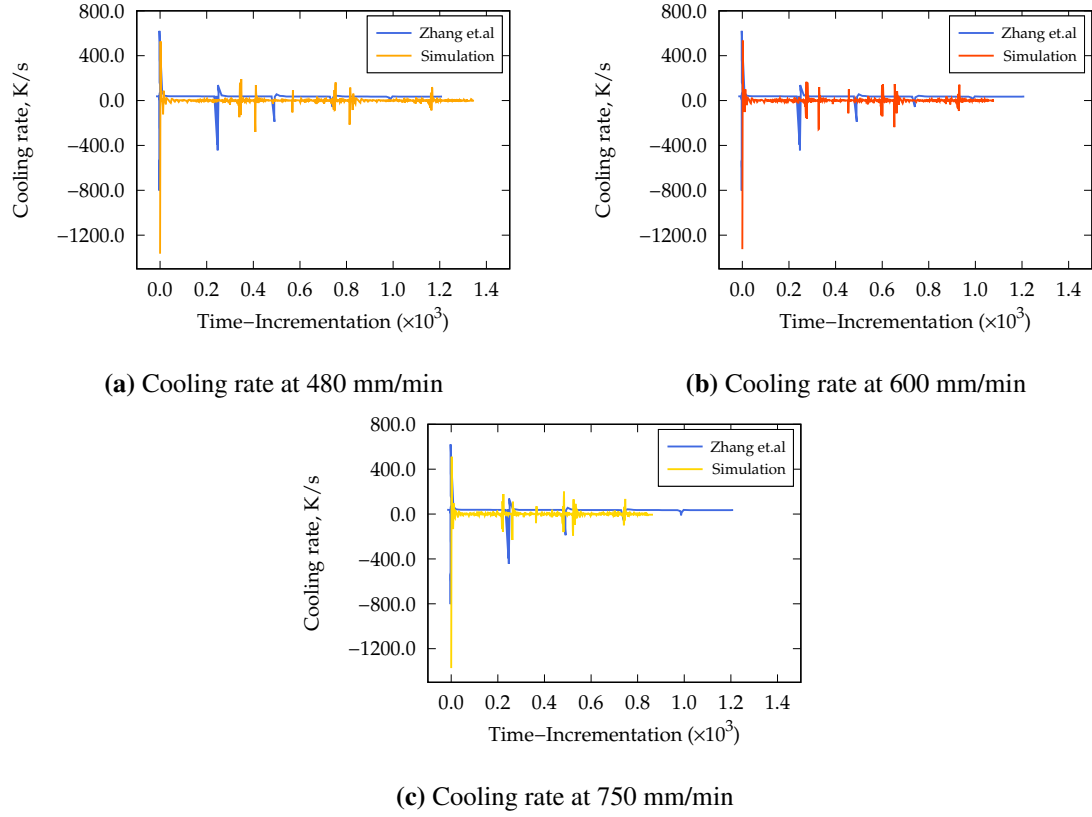


Figure 10: Cooling rate comparison

Across all three cases, the largest thermal cycle coincides with the initial material deposition. As the travel speed increases from TS1 to TS3, the measured peak cooling rate increases from approximately 527.75 K/s (TS1) to 540.97 K/s (TS3). This trend is physically consistent: a faster-moving arc travels any given point in a shorter time interval, intensifying the instantaneous thermal gradient while reducing the total heat input per unit length.

Using the peak cooling rate values of both the benchmark result of Zhang et al. [22] (at 621.3 K/s) and of the three thermal models of this study (527.75, 534.44 and 540.97 K/s, respectively for TS1, TS2 and TS3), it can be deduced whether it is possible to validate the study's results by calculating the percentage error as follows:

$$\text{Percentage error} = \frac{\text{Benchmark result} - \text{Simulated FEA}}{\text{Benchmark result}} \times 100 \quad (13)$$

The percentage errors were 15.06%, 13.98%, and 12.92% for TS1, TS2, and TS3, respectively, indicating that TS3 achieves the closest overall agreement with the benchmark data. However, TS2 provides the most accurate prediction of total time duration, at the cost of only a 1.06% increase in percentage error relative to TS3. Given this marginal trade-off, TS2 was selected as the best-validated model, since accurate simulation of the thermal cycle duration is more critical to capturing the precipitation-sensitive cooling behavior of Inconel 718 than a marginal reduction in aggregate error.

5 Conclusion

The synergy between topology optimization and WAAM provides a systematic method in the design and development of lightweight and complex structures of automotive components. In this study, FSAE

vehicle's front upright is redesigned to attain weight reduction. Topology optimization gives the complex structure while WAAM builds it with better accuracy.

SIMP based topology optimization with an objective function of volume minimization and constraints of von Mises stress was carried out. Three alternative geometries were found from which the best optimized geometry (with 41.58 % volume reduction) was selected based on multi-decision criteria analysis and was used for the thermal modeling of WAAM. A parametric process optimization of WAAM was then carried out with which the optimized parameters were used in a transient thermal modeling with three travel speeds (480, 600, 750 mm/min) assuming 0 dwell time. The results of the thermal modeling is summarized below:

- Increasing travel speed reduces peak reheating temperatures and overall process duration
- Lower layers build experience the most thermal histories across all deposition speeds, accumulating the highest number of thermal cycles

Future work may include implementing other types of material interpolations for the topology optimization. The effect of temperature gradients on residual stress and distortion can also be studied.

Overall, the proposed framework establishes a systematic design-to-manufacturing methodology for developing lightweight metallic components fabricated by WAAM, with improved manufacturability and enhanced process reliability.

References

- [1] Czerwinski F 2021 *Materials* **14** 6631
- [2] Junk S, Klerch B and Hochberg U 2019 *Procedia CIRP* **84** 277–282
- [3] Hamza A, Bousnina K, Dridi I and Ben Yahia N 2025 *Vehicles* **7** 24
- [4] Seward D 2022 *Race Car Design* (Bloomsbury Publishing Plc)
- [5] Ghungarde V, Awachar S, Vaidya N K and Jagadeesha T 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **624** 012023
- [6] Gunev D and Iliev S 2023 “Shell Eco-marathon” prototype car's steering knuckle, topology optimization and design thinking *AIP Conf. Proc.* vol 2868 p 020017
- [7] Kamble A V, Patil S S and Harugade M L 2017 *Int. J. Sci. Res. Dev.* **5** 77–81
- [8] Dhinakaran V, Kumar A R, Ramgopal R, Kannan S, Stalin B and Jagadeesha T 2021 Topology Optimization of Steering Knuckle *Adv. Ind. Autom. Smart Manuf.* vol 23 pp 207–224
- [9] Béraud N, Ledoux Y, Mechekour E H, Vignat F and Pourroy F 2024 *CIRP J. Manuf. Sci. Technol.* **55** 234–246
- [10] Dalpadulo E, Pini F and Leali F 2020 *Int. J. Interact. Des. Manuf.* **14** 899–909
- [11] Pavlos V 2024 *Design with topology optimization for Additive Manufacturing of a racing car component* Diploma thesis UNIVERSITY OF ATHENS
- [12] Mazínová I and Florian P 2014 Materials Selection in Mechanical Design *Lect. Notes Mech. Eng.* vol 16 (Springer International Publishing) pp 145–153
- [13] Pratheesh Kumar S, Elangovan S, Mohanraj R and Ramakrishna J 2021 *Mater. Today: Proc.* **46** 7892–7906
- [14] Webber J A, Liverett G K and Webber J A 2023 *Finite Element Analysis of a FSAE Electric Vehicle Suspension Upright* Honors diploma thesis University of Alabama in Huntsville
- [15] Dassault Systemes 2024 SIMULIA Assistance User

- [16] Atabay S E, Wanjara P, Bernier F, Sarafan S, Gholipour J, Soost J, Amos R, Patnaik P and Brochu M 2022 *Materials* **16** 1 ISSN 1996-1944
- [17] Wieckowski J, Sałabun W, Kizielewicz B, Bączkiewicz A, Shekhovtsov A, Paradowski B and Wątróbski J 2023 *Int. J. Knowledge-Based Intell. Eng. Syst.* **27** 367–393
- [18] Rosado-Tamariz E, Davila-De-La-Peña R, Campos-Amezcu A, García-Beltrán C D, Campos-Amezcu A and Colín-Castillo J Y 2026 *Appl. Sci.* **16** 4257
- [19] Zhou M, Cui M, Xu D, Zhu S, Zhao Z and Abusorrah A 2024 *IEEE/CAA J. Autom. Sin.* **11** 1092–1105
- [20] Failla Jr D P and Nguyen C J 2025 AMPES: Additive Manufacturing Process Event Series Generator
- [21] Al Hamahmy M I and Deiab I 2020 *Int. J. Adv. Manuf. Technol.* **106** 1223–1238
- [22] Zhang T, Li H, Gong H, Wu Y, Chen X and Zhang X 2022 *J. Mater. Res. Technol.* **18** 3056–3072