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Preliminary Thermo-Mechanical Assessment of a Pneumatically Actuated Permanent-Mold Gravity-Casting Apparatus

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Abstract.

Low-cost automation of permanent-mold gravity casting requires adequate actuator capacity, guide stiffness, and accommodation of differential thermal growth. This study presents a preliminary analytical assessment of a compact, pneumatically actuated permanent-mold gravity-casting apparatus. The assessment uses documented geometry and conservative screening assumptions for a 100 mm-bore double-acting cylinder at 0.60 MPa, a 25 mm piston rod, two 40 mm S45C guide shafts over a 0.60 m span, locally procured bronze sleeve bushings, and a ferrous mold plate that is 40 mm thick with a 0.40 m axial characteristic length. For horizontal translation, the force model excludes the full assembly weight from the cylinder-axis resistance and includes guide friction ($\mu = 0.15$), 100 N ejector-spring resistance, a 200 N unmeasured process allowance, and a design factor of 2. The resulting factored demand is 0.748 kN. The extension and retraction load ratios are 0.159 and 0.169 relative to theoretical cylinder output, below the 0.50 guideline used for dynamic cylinder selection. The idealized guide-shaft model predicts 0.045 mm deflection and 5.97 MPa bending stress. Nominal thermal growth is 1.20 mm for the 0.40 m mold dimension and 0.44 mm for the 0.80 m rail length under assumed temperature rises, leaving a nominal 0.80 mm margin against a 2.0 mm drawing allowance before tolerance effects. Because the mold grade, bronze alloy, as-built clearance, stroke history, and trial-cycle data were not documented, no durability, measured-alignment, or material-specific PV-compliance claim is made. The contribution is an integrated and explicitly bounded force-stiffness-thermal screening framework for low-cost casting apparatuses.

Keywords: permanent-mold gravity casting; pneumatic actuation; guide-shaft stiffness; thermal expansion; low-cost automation.

INTRODUCTION

Permanent-mold gravity casting remains important for aluminium components because reusable metallic molds can provide stable cavity geometry, rapid heat extraction, and repeatable production when the thermal condition of the mold is controlled [1]. Foundry handbooks describe this process as a robust route for repeated aluminium casting, but they also emphasize that mold material, mold coating, preheating, and thermal cycling strongly affect dimensional repeatability and defect formation [2]. Recent thermal-management reviews further show that permanent-mold operation must balance preheating, heat extraction, and cooling intensity to maintain a stable filling and solidification window [3]. From a solidification viewpoint, this balance is essential because heat removal governs local freezing rate, shrinkage behavior, and the formation of temperature gradients within both casting and mold [4].

For a compact casting apparatus, the thermal problem cannot be separated from the machine-design problem. The mold, mounting plate, guide shafts, bushings, cylinder bracket, and support frame form a single kinematic assembly, but each part experiences a different temperature history. The mold is directly exposed to molten aluminium and preheating, whereas the guide system generally receives indirect conductive and radiant heat. Analytical and numerical studies on permanent-mold heat conduction and interfacial heat-transfer coefficients demonstrate that transient temperature fields and metal-mold contact conditions are central to process performance [5], [6].

Differential thermal expansion can create practical mechanical problems when the moving and fixed mold halves are constrained too rigidly. If the mold expands axially while the guide rails or supports are fixed, parasitic stresses may increase sliding resistance, disturb parting-face alignment, generate bushing misfit, or accelerate wear. The effect is especially relevant when pouring and cooling temperatures vary from cycle to cycle, because such temperature changes influence microstructure and mechanical properties in A356 aluminium permanent-mold casting [7].

The transition from manual mold handling to powered opening, closing, and ejection consequently introduces a coupled verification requirement. The actuator must provide adequate closing and opening force, the guide shafts must limit deflection to preserve alignment, and the support arrangement must allow thermal movement without sacrificing mold positioning. Earlier casting-solidification simulations show the value of evaluating thermal behavior before production trials [8], while broader reviews of casting simulation explain how virtual validation can improve productivity, quality, and defect control [9]. Equipment-level studies also indicate that maintainability and cost-effectiveness should be considered together when designing compact molten-aluminium casting mechanisms [10].

Despite these developments, many low-cost apparatuses intended for small and medium foundry operations are still described mainly in terms of fabrication feasibility or production function. Quantitative evidence for actuator capacity, guide-system stiffness, bushing screening, and thermal-expansion accommodation is often limited. Structural analysis of low-pressure casting-machine components confirms that stiffness and support conditions should be verified before repeated operation [11]. In addition, gravity-die-casting studies show that mold layout, thermal condition, and demolding temperature can influence defect formation and casting distortion, reinforcing the need to treat the apparatus as a coupled thermo-mechanical system [12], [13].

This study addresses that need through a preliminary analytical assessment of a compact, pneumatically actuated permanent-mold gravity-casting apparatus for aluminium motorcycle-accessory components. Its novelty is the integration of a horizontal actuator-resistance model, a dynamic cylinder-load-ratio check, guide-shaft stiffness screening, nominal thermal-clearance accounting, and bushing PV-demand calculation within one explicitly bounded framework. Unlike a comprehensive validation study, the present work separates documented dimensions from unverified material, temperature, clearance, and commissioning assumptions. The study does not evaluate throughput, cost, casting-defect statistics, positional repeatability, or durability.

METHODS

Apparatus Architecture and Assessment Boundary

The investigated machine is a single-station permanent-mold gravity casting apparatus with a horizontal parting arrangement. One mold half is attached to a fixed reference plate, while the opposing mold half is mounted on a movable base plate that translates along two parallel guide shafts. A double-acting pneumatic cylinder drives the movable plate, and a mechanically linked ejector plate releases the casting during the opening stroke.

The assessment boundary covers the mechanical load path from the pneumatic cylinder through the movable base plate and guide shafts, together with the thermally sensitive interfaces between the mold, mounting arrangement, and guide system. The source drawings available for this manuscript did not document the center-to-center spacing of paired bushings, and no publication-quality as-built photograph or CAD export was available; consequently, the schematic is not to scale and is not presented as a fabrication drawing. The boundary excludes coupled filling-solidification simulation, detailed cavity stress analysis, and fatigue-life modeling.

Design Inputs and Acceptance Criteria

The documented dimensions and assumptions used for screening are summarized in Table 1. Values were taken from the final CAD description, manufacturing notes, and the original design worksheet where available. Inputs

not supported by a material certificate, as-built measurement, or instrumented trial record are identified explicitly as assumptions rather than acceptance evidence. Dimensional relationships were interpreted using geometric-tolerancing principles [14].

Table 1. Design inputs and acceptance criteria used for analytical validation.

Subsystem	Input parameter	Design value / criterion
Pneumatic actuator	Bore diameter; rod diameter; regulated gauge pressure	100 mm; 25 mm; 0.60 MPa (6 bar)
Horizontal load model	Normal load; friction coefficient; spring resistance; process allowance; design factor	491 N; 0.15; 100 N; 200 N; 2.0
Guide system	Material; shaft diameter; support span	S45C steel; 40 mm; 0.60 m
Guide-system load	Total conservative load; screened load per shaft	500 N; 250 N
Alignment screening	Internal shaft-deflection threshold	< 0.10 mm; not a measured casting-tolerance limit
Linear bushing	Procurement record; internal diameter; axial length; nominal speed	Bronze alloy/product code not documented; 40 mm; 100 mm; 0.50 m/s (assumed)
Mold thermal case	Material record; plate thickness; axial length; assumed temperature rise	Ferrous plate (“iron”), grade not documented; 40 mm; 0.40 m; 250 °C
Rail thermal case	Material; characteristic length; assumed temperature rise	S45C steel; 0.80 m; 50 °C
Thermal allowance	Nominal drawing allowance; rail support; tolerance evidence	2.0 mm; one floating support; as-built clearance/tolerance stack not measured

The theoretical cylinder output was calculated from the effective piston area at the regulated gauge pressure. Equation (1) uses the full-bore area for extension, whereas Equation (2) uses the annular area after subtracting the piston-rod cross-section. These values represent ideal pressure-area output and do not, by themselves, include seal friction, line pressure loss, acceleration, valve restriction, or regulator variation [15], [16].

$$F_e = P\pi D^2/4 \quad (1)$$

$$F_r = P\pi(D^2 - d^2)/4 \quad (2)$$

In Equations (1) and (2), P is the regulated gauge pressure, D is the cylinder-bore diameter, d is the piston-rod diameter, F_e is the extension force, and F_r is the retraction force.

Because the mold carriage translates horizontally, the full gravitational weight does not act directly against the cylinder axis. The weight is used only to determine the guide normal load N and the corresponding Coulomb friction. The screened axial resistance and factored design demand were therefore calculated using Equations (3) and (4). The original design inputs were $\mu = 0.15$, $N = 491$ N, a spring resistance of 100 N, and an additional 200 N process allowance. The spring resistance and process allowance were not measured during commissioning and are retained as conservative assumptions. A design factor of 2 was applied to cover unmodeled acceleration and operating variation. Adequacy was also expressed as the ratio of factored demand to theoretical cylinder output. Manufacturer selection guidance recommends a ratio no greater than 0.50 for dynamic pneumatic-cylinder operation [17].

$$R = \mu N + F_s + F_a \quad (3)$$

$$F_d = SF \times R \quad (4)$$

In Equations (3) and (4), R is the screened axial resistance, μ is the assumed guide-friction coefficient, N is the normal load generated by the movable assembly, F_s is the nominal ejector-spring resistance, F_a is the unmeasured process-resistance allowance, SF is the design factor, and the factored demand is given by Equation (4).

Guide-shaft stiffness was screened by dividing the 500 N conservative moving load equally between the two shafts and modeling each shaft as a simply supported circular beam over the 0.60 m distance between end supports. The load transferred through the movable plate and paired bushings was represented as a single mid-span point load of 250 N per shaft. This is a reproducible first-order model, but it does not reproduce the undocumented bushing spacing, distributed contact, end-block compliance, plate moment transfer, or assembly misalignment. The model therefore screens global elastic bending only [18], [19].

$$I = \pi d^4 / 64 \quad (5)$$

$$\delta_{\max} = WL^3 / (48EI) \quad (6)$$

$$\sigma_{\max} = M_{\max}c / I, \text{ where } M_{\max} = WL / 4 \quad (7)$$

In Equations (5) – (7), I is the second moment of area, d is the guide-shaft diameter, $\delta_{\max}$ is the predicted mid-span deflection, W is the screened load on one shaft, L is the support span, E is Young's modulus, $\sigma_{\max}$ is the maximum bending stress, $M_{\max}$ is the maximum bending moment, and c is the shaft radius. The 0.10 mm value is an internal preliminary screening threshold; because no measured parting-face mismatch or casting-dimensional tolerance was available, it is not treated as a product-acceptance limit.

Thermal growth was estimated with the one-dimensional linear-expansion relation in Equation (8). The fabrication record describes the mold stock only as a ferrous or “iron” plate and does not contain a material grade or mill certificate. The 40 mm value is the plate thickness, whereas 0.40 m is the in-plane axial characteristic length used for expansion. A generic ferrous-material coefficient of $12 \times 10^{-6} \text{ K}^{-1}$ was used only for screening [20], [21]. The assumed temperature rises of 250 °C for the mold and 50 °C for the rail were design cases rather than thermocouple measurements.

$$\Delta L = \alpha L_0 \Delta T \quad (8)$$

In Equation (8), ΔL is the predicted length change, α is the assumed coefficient of thermal expansion, L_0 is the initial characteristic length, and ΔT is the assumed temperature rise. The mold result was compared with the 2.0 mm nominal axial allowance shown in the design description. Manufacturing tolerance, actual cold assembly clearance, direction-specific constraint, and hot as-built clearance were not measured; therefore, only a nominal remaining margin can be reported.

Bronze-bushing contact was evaluated as an operating-demand calculation rather than a material qualification. The bushing was locally procured, but its bronze alloy, manufacturer, product code, allowable pressure/PV limit, and operating-temperature rating were not documented. The lubrication note specifies periodic grease application, but grease type and interval, shaft hardness and surface roughness, and bushing temperature were not recorded. The 0.50 m/s sliding velocity is an original design assumption and cannot be reconstructed from stroke and travel time because those measurements were not logged. Pressure was calculated from $p = W/(dL)$, where L is bushing length, and PV demand from pV . Product literature emphasizes that bronze-bushing suitability depends on alloy, lubrication, mating-surface condition, velocity, temperature, and the supplier-specific limit [22]; hence no pass/fail PV claim is made.

Commissioning Observations and Evidence Limitations

The available commissioning note states only that the mechanism completed opening and closing motion and that no obvious binding was observed. It does not report the number of cycles, dry versus heated sequence, mold preheating temperature, actual cylinder-port pressure, stroke length, travel time, lubrication condition, positional repeatability, or inspection criterion. These observations are therefore retained only as qualitative context and are excluded from the quantitative assessment and durability conclusions.

RESULT AND DISCUSSION

Horizontal Load Path and Actuator Capacity

The horizontal load path begins at the pneumatic cylinder, passes through the piston rod and clevis into the movable base plate, and is reacted through the paired bushings and guide-shaft supports. Gravitational weight acts normal to the translation direction and contributes to friction, but it is not added directly to the cylinder-axis resistance. Table 2 summarizes the corrected actuator demand, cylinder load ratios, structural screening results, and nominal thermal-clearance calculation.

Table 2. Analytical screening results for the principal motion and thermal-design requirements.

Requirement	Calculated result	Design interpretation
Cylinder theoretical extension output	4.712 kN	Ideal pressure-area output at 0.60 MPa.
Cylinder theoretical retraction output	4.418 kN	Ideal annular-area output at 0.60 MPa.
Horizontal screened resistance	0.374 kN	Friction + spring + process allowance; full weight excluded from axial resistance.
Factored design demand	0.748 kN	Screened resistance multiplied by SF = 2.
Extension load ratio	0.159	Factored demand / theoretical extension output; below 0.50 dynamic-selection guideline.
Retraction load ratio	0.169	Factored demand / theoretical retraction output; below 0.50 dynamic-selection guideline.
Guide-shaft second moment of area	$1.257 \times 10^{-7} \text{ m}^4$	Correct SI exponent notation for $d = 40 \text{ mm}$.
Guide-shaft maximum deflection	0.045 mm	Below internal 0.10 mm threshold; not measured mold alignment.
Guide-shaft maximum bending stress	5.97 MPa	Idealized static beam result; local contacts and fatigue not included.

The ideal pressure-area outputs are 4.712 kN in extension and 4.418 kN in retraction. For horizontal translation, the assumed guide friction is $\mu N = 0.15 \times 491 = 73.7$ N. Adding the 100 N ejector-spring resistance and the 200 N unmeasured process allowance gives a screened resistance of 373.7 N; applying $SF = 2$ gives a factored demand of 747.3 N (0.748 kN). The resulting extension and retraction load ratios are 0.159 and 0.169. Both are below the 0.50 dynamic-selection guideline [29], corresponding to margins of 3.15 and 2.96 relative to 50% of theoretical output. These margins support preliminary sizing adequacy under the stated assumptions, but they do not replace cylinder-port pressure, stroke-speed, or hot-stroke force measurements.

Guide-Shaft Stiffness and Mold Alignment

With a 40 mm shaft diameter, 0.60 m support span, 200 GPa elastic modulus, and 250 N screened point load per shaft, the second moment of area is $1.257 \times 10^{-7} \text{ m}^4$. The idealized simply supported model gives a maximum mid-span deflection of 0.045 mm and a maximum bending stress of 5.97 MPa. The deflection is below the internal 0.10 mm screening threshold, but this comparison does not demonstrate mold-parting alignment because the threshold was not derived from a measured casting tolerance.

The mid-span point-load assumption is conservative relative to a uniformly distributed load of the same total magnitude, but actual end restraint and paired-bushing moment transfer may either reduce global bending or increase local reaction. Shaft straightness, support parallelism, bushing concentricity, contact spacing, local wear, and contamination are not captured. The calculated value should therefore be treated as one component of an alignment budget. Structural machine-design references and casting-machine component analysis support the use of stiffness screening before fabrication [11], [18], [19], whereas final acceptance requires ambient and hot alignment surveys.

The projected bushing pressure is 0.0625 MPa, and the calculated PV demand at the assumed 0.50 m/s speed is 0.0313 MPa·m/s. These values are mathematically consistent with the selected load and geometry, but they are not compared with an acceptance limit because the installed bronze alloy and supplier rating are unknown. The result should be interpreted only as the operating point that must be checked against a verified bushing data sheet. Intermittent start-stop motion, edge loading from misalignment, grease condition, shaft finish, and thermal exposure can produce local behavior not represented by the nominal projected-area calculation [23].

Thermal-Expansion Screening and Nominal Clearance Margin

The thermal calculation distinguishes the 40 mm mold-plate thickness from the 0.40 m in-plane axial length. Using the generic screening coefficient $\alpha = 12 \times 10^{-6} \text{ K}^{-1}$ and the assumed 250 °C mold temperature rise gives a predicted mold growth of 1.20 mm. For the 0.80 m S45C rail length, $\alpha = 11 \times 10^{-6} \text{ K}^{-1}$ and

the assumed 50 °C rise give a predicted rail growth of 0.44 mm. Neither temperature rise was measured, and the mold material grade remains unverified; the results are therefore design-case estimates rather than calibrated thermal predictions.

Against the 2.0 mm nominal drawing allowance, the calculated nominal remainder is $2.0 - 1.20 = 0.80$ mm before manufacturing and assembly tolerances. Because the actual cold clearance, tolerance stack, and direction of constraint were not measured, the 0.80 mm value cannot be interpreted as a certified as-built margin. The floating rail support remains a rational concept for preventing axial thermal preload, but its travel and retention should be verified directly at operating temperature.

Permanent-mold heat transfer changes throughout each cycle and depends on mold temperature, interface condition, coating, and process history [3], [5], [6]. Consequently, a single assumed temperature rise cannot resolve the local gradients that govern fit and alignment. The present calculation is useful for identifying the scale of required accommodation, while thermocouple mapping and hot-clearance measurement are needed to establish the actual boundary conditions.

Qualitative Commissioning Observations

No reproducible functional-trial dataset was available for analysis. The original commissioning note reports smooth full-stroke motion and no obvious rail seizure, but the cycle count, preheating condition, stroke length, stroke speed, lubrication state, and inspection threshold were not recorded. Accordingly, the observation is not used to verify positional repeatability, clamping force, hot-stroke resistance, or durability.

The intended operating sequence comprises pressure verification, mold preheating, controlled closure, manual pouring, solidification, pneumatic opening with mechanical ejection, cavity cleaning, and periodic rail lubrication. Flow controls remain important because a favorable static force ratio does not limit impact energy at mold closure. Future commissioning records should distinguish dry cycling, heated no-pour cycling, and actual casting operation and should report the variables listed in Table 3.

Design-Risk Matrix and Practical Controls

A compact apparatus should be assessed as a system of interacting failure modes rather than as isolated components. Table 3 translates the analytical findings into practical controls and identifies the evidence required before scale-up or a stronger reliability claim.

Table 3. Design risks, controls, and recommended verification evidence.

Potential design risk	Mechanism	Current design control	Evidence required before scale-up
Mold/rail binding	Differential thermal expansion creates axial restraint or bushing misfit.	Nominal 2.0 mm drawing allowance; floating rail support; actual hot clearance not measured.	Material grade, thermocouple mapping, cold/hot clearance measurement, and hot-stroke force trace.
Parting-face misalignment	Guide deflection, mounting non-parallelism, or bushing wear changes relative mold position.	Two 40 mm guide shafts; 0.045 mm idealized screening deflection.	Bushing spacing, ambient/hot alignment survey, and dimensional audit of castings.
Impact at mold closure	High cylinder speed generates local contact shock.	Port flow control and operator-controlled sequence; no impact measurement.	Stroke-speed measurement and contact-impact assessment.
Incomplete ejection	Insufficient opening force, ejector friction, or local casting adhesion.	Low calculated retraction load ratio and mechanically linked ejector; no ejection-force trace.	Ejection-force measurement and defect inspection over repeated cycles.
Pressure loss or force drift	Regulator error, line restriction, or valve leakage reduces available cylinder force.	Regulator, FRL unit, and nominal 6 bar setting; cylinder-port pressure not logged.	Pressure logging at cylinder ports and preventive-maintenance records.
Thermal fatigue and mold degradation	Repeated heating/cooling produces cracks, distortion, or altered surface condition.	Nominal clearance concept and visual checks; mold grade and thermal history undocumented.	Long-term thermal-cycle programmed, crack inspection, and mold-temperature history.
Bushing wear or seizure	Unverified alloy, poor lubrication, shaft-finish error, edge loading, or high temperature.	Pressure and PV demand calculated; grease use noted; alloy and limit unverified.	Supplier data sheet, lubricant, shaft finish/hardness, temperature, and wear record.

Implications For Permanent-Mold Apparatus Design

The corrected force balance shows that low-cost pneumatic actuation is feasible under the stated horizontal-load assumptions, but force capacity alone is not the governing proof of machine performance. The dynamic load ratios are low, whereas the larger unresolved risks are hot alignment, unknown as-built clearance, pressure variation, impact at closure, and bushing material identification. The apparatus should therefore be treated as a preliminary design whose next acceptance gate is instrumented operation.

S45C guide shafts are a documented design input, whereas the mold grade and bronze-bushing alloy are not. This difference is important: shaft calculations can be linked to a specified material, but thermal-expansion and PV calculations remain screening estimates until material certificates and supplier data are obtained. Future hardware revisions should record shaft hardness and roughness, bushing alloy and lubricant specification, mold grade, nominal and measured clearances, and support-fit tolerances.

Thermal fatigue deserves particular attention in the next development stage. Studies of die-casting tools and surface-engineered hot-work tool steels demonstrate that repeated heating and cooling can initiate cracks, change surface condition, and shorten tool life, with temperature gradient and cooling history affecting the severity of the damage [24], [27], [28]. The thermal loading in the present gravity-casting apparatus is less severe than high-pressure die casting, but the same physical principle applies: repeated gradients can change mold geometry and contact conditions over time.

Limitations and Future Validation Program

This assessment has five principal limitations. First, spring resistance, the 200 N process allowance, cylinder-port pressure, acceleration, and line losses were not measured. Second, the thermal model is one-dimensional, the temperature rises are assumed, and the mold grade is undocumented. Third, the guide-shaft model idealizes support and load transfer and omits actual bushing spacing, assembly tolerances, and wear. Fourth, the bushing alloy, supplier limit, shaft finish, lubricant specification, and operating temperature are unavailable, preventing a material-specific PV qualification. Fifth, the commissioning record lacks the cycle count and operating variables needed for reproducible functional evidence.

The next-stage program should combine instrumented trials with a thermal-structural model. K-type thermocouples should be installed in the mold body, base plate, bushings, and guide supports; pressure sensors should be placed near both cylinder ports; and a displacement sensor should record stroke time, end position, and repeatability. Cold and hot axial clearances, mold-parting mismatch, bushing temperature, ejection force, and closure impact should be measured over documented dry, heated, and casting cycles. Material certificates

and the bushing supplier data sheet should be obtained before the PV calculation is used as an acceptance criterion.

CONCLUSION

This study presents a preliminary analytical assessment of a compact, pneumatically actuated permanent-mold gravity-casting apparatus. Correcting the horizontal force balance gives a screened resistance of 0.374 kN and a factored design demand of 0.748 kN. At 0.60 MPa, the 100 mm-bore cylinder provides theoretical extension and retraction outputs of 4.712 and 4.418 kN, producing load ratios of 0.159 and 0.169, respectively, below the 0.50 dynamic-selection guideline. The idealized two-shaft model predicts a second moment of area of $1.257 \times 10^{-7} \text{ m}^4$, a maximum deflection of 0.045 mm, and a bending stress of 5.97 MPa. The thermal design cases predict 1.20 mm mold growth along the 0.40 m axial dimension and 0.44 mm rail growth. Relative to the 2.0 mm nominal drawing allowance, the mold result leaves a nominal 0.80 mm margin before tolerance effects. The calculated bushing pressure and PV demand are 0.0625 MPa and 0.0313 MPa·m/s, but no compliance conclusion is possible without the installed alloy and supplier limit. Because material grades, as-built clearances, and reproducible commissioning data are incomplete, the findings support preliminary design screening only. Instrumented pressure, temperature, displacement, hot alignment, and durability testing is required before operational validation or scale-up.

REFERENCES

- [1] J. Campbell, *Complete Casting Handbook: Metal Casting Processes, Metallurgy, Techniques and Design*, 2nd ed. Oxford, UK: Butterworth-Heinemann, 2015.
- [2] ASM International, *{ASM Handbook}, Volume 15: Casting*. Materials Park, OH: ASM International, 2008.
- [3] C. Ahn, C. Söderhjelm, and D. Apelian, “Enabling Technologies for Thermal Management During Permanent Mold Casting: A Critical Review,” *Int. J. Met.*, 2025, doi: 10.1007/s40962-025-01784-4.
- [4] J. A. Dantzig and M. Rappaz, *Solidification*, 2nd ed. Lausanne, Switzerland: EPFL Press, 2016.
- [5] M. Ahmadein, A. H. Elsheikh, and N. A. Alsaleh, “Modeling of Cooling and Heat Conduction in Permanent Mold Casting Process,” *Alexandria Eng. J.*, vol. 61, no. 2, pp. 1757–1768, 2022, doi: 10.1016/j.aej.2021.06.048.
- [6] G. de M. Stieven, D. dos R. Soares, E. P. Oliveira, and E. F. Lins, “Interfacial Heat Transfer Coefficient in Unidirectional Permanent Mold Casting: Modeling and Inverse Estimation,” *Int. J. Heat Mass Transf.*, vol. 166, p. 120765, 2021, doi: 10.1016/j.ijheatmasstransfer.2020.120765.
- [7] X. P. Hu, Y. Zhao, Q. Wang, X. Z. Zhang, R. X. Li, and B. R. Zhang, “Effect of Pouring and Cooling Temperatures on Microstructures and Mechanical Properties of As-Cast and

- T6 Treated A356 Alloy,” *China Foundry*, vol. 16, no. 6, pp. 380–385, 2019, doi: 10.1007/s41230-019-9068-8.
- [8] T. R. Vijayaram, S. Sulaiman, A. M. S. Hamouda, and M. H. M. Ahmad, “Numerical Simulation of Casting Solidification in Permanent Metallic Molds,” *J. Mater. Process. Technol.*, vol. 178, no. 1–3, pp. 29–33, 2006, doi: 10.1016/j.jmatprotec.2005.09.025.
- [9] M. A. A. Khan and A. K. Sheikh, “Simulation Tools in Enhancing Metal Casting Productivity and Quality: A Review,” *Proc. Inst. Mech. Eng. Part B J. Eng. Manuf.*, vol. 230, no. 10, pp. 1799–1817, 2016, doi: 10.1177/0954405416640183.
- [10] B. Bakhtiar, N.-H. Wang, C.-M. Hsu, K.-K. Fan, C.-F. Yang, and D.-C. Cai, “Design and Simulation of a Maintenance-Friendly and Cost-Effective Molten Aluminum Alloy Casting Mechanism,” *Sensors Mater.*, vol. 36, no. 7–2, pp. 2955–2964, 2024, doi: 10.18494/SAM5158.
- [11] Y. N. Aydin, O. Özyaydin, and A. A. Akilcs, “Static Analysis of the Slotted Upper Plate of Low Pressure Casting Machine,” *Celal Bayar Univ. J. Sci.*, vol. 17, no. 1, pp. 35–41, 2021, doi: 10.18466/cbayarfbe.779687.
- [12] M. R. M. Kamal, M. S. Sulaiman, S. M. Sapuan, and M. M. H. M. Ahmad, “Design and Thermal Analysis Simulation of Gravity Die Casting Mold,” *J. Adv. Manuf. Technol.*, vol. 12, no. 2, pp. 197–209, 2018.
- [13] S. Gor, A. K. Jha, and S. K. Gupta, “Numerical Analysis of the Influence of Demolding Temperature on Distortion in Gravity Die Casting of Aluminum Alloy A356,” *Int. J. Met.*, 2025, doi: 10.1007/s40962-024-01398-2.
- [14] American Society of Mechanical Engineers, “ASME Y14.5-2018, Dimensioning and Tolerancing.” New York, NY, USA, 2018.
- [15] S. R. Majumdar, *Pneumatic Systems: Principles and Maintenance*. New Delhi, India: Tata McGraw-Hill, 1995.
- [16] A. Parr, *Hydraulics and Pneumatics: A Technician’s and Engineer’s Guide*, 3rd ed. Oxford, UK: Butterworth-Heinemann, 2011.
- [17] “Air Cylinders: Model Selection.” 2023.
- [18] W. C. Young and R. G. Budynas, *Roark’s Formulas for Stress and Strain*, 7th ed. New York, NY, USA: McGraw-Hill, 2002.
- [19] R. G. Budynas and J. K. Nisbett, *Shigley’s Mechanical Engineering Design*, 11th ed. New York, NY: McGraw-Hill Education, 2020.
- [20] J. P. Holman, *Heat Transfer*, 10th ed. Dallas: Higher Education, 2010.
- [21] T. L. Bergman, A. S. Lavine, F. P. Incropera, and D. P. DeWitt, *Fundamentals of Heat and Mass Transfer*, 8th ed. Hoboken, NJ, USA: Wiley, 2017.
- [22] SKF Group, “Bushing Data: Solid Bronze Bushings.” Gothenburg, Sweden.
- [23] B. J. Hamrock, S. R. Schmid, and B. O. Jacobson, *Fundamentals of Fluid Film Lubrication*, 2nd ed. Boca Raton, FL, USA: CRC Press, 2004.
- [24] M. Liu, B. Sang, C. Hao, G. Chen, and J. Yan, “Thermal Fatigue Life Prediction Method for Die Casting Mold Steel Based on the Cooling Cycle,” *J. Mater. Process. Technol.*, vol. 321, p. 118131, 2023, doi: 10.1016/j.jmatprotec.2023.118131.
- [25] D. Klobčar, L. Kosec, B. Kosec, and J. Tušek, “Thermo Fatigue Cracking of Die Casting Dies,” *Eng. Fail. Anal.*, vol. 20, pp. 43–53, 2012, doi: 10.1016/j.engfailanal.2011.10.005.

- [26] A. Persson, S. Hogmark, and J. Bergström, “Thermal Fatigue Cracking of Surface Engineered Hot Work Tool Steels,” *Surf. Coatings Technol.*, vol. 191, pp. 216–227, 2005, doi: 10.1016/j.surfcoat.2004.03.044.
- [27] G. W. Stachowiak and A. W. Batchelor, *Engineering Tribology*, 4th ed. Oxford, UK: Butterworth-Heinemann, 2013.
- [28] ASM International, *ASM Handbook, Volume 18: Friction, Lubrication, and Wear Technology*. Materials Park, OH, USA: ASM International, 1992.