

Parametric study of wall-mounted annular ring fins for heat transfer enhancement in a flat-bottom cookstove using CFD

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Abstract

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Flat-bottom biomass cookstoves, widely used in developing countries, typically achieve thermal efficiencies below 20% because most combustion energy is lost with the exhaust gas before reaching the pot. Enhancing convective heat transfer at the pot wall therefore represents the most direct pathway to improved efficiency. Although annular fins are well-established passive heat transfer enhancers in industrial heat exchangers, their geometric design space remains unexplored within cookstove combustion chambers. This study addresses that gap through a parametric computational fluid dynamics (CFD) investigation of wall-mounted ring fins in a flat-bottom biomass cookstove. A three-dimensional steady-state CFD model, developed in ANSYS Fluent using the SST k- ω turbulence model with enhanced wall treatment, was validated against measured temperatures at three vertical levels and the chimney outlet. The model showed close agreement at the outlet (2.21% error), while larger discrepancies at the interior levels (16–28%) were attributed to the simplified uniform-inlet boundary condition. Using a one-factor-at-a-time design, thirteen configurations were evaluated across four geometric parameters: fin height, blockage ratio, tier count, and installation height. Fin height showed a non-monotonic effect, with thermal performance peaking at a moderate height before excessive blockage limited further gains. Tier count showed a monotonic trend, reaching a thermal performance factor (TPF) of 1.118 with three tiers. Installation height emerged as the most influential parameter: a single fin positioned near the hot gas inlet achieved the highest performance, with a TPF of 1.138, corresponding to a 14.1% increase in heat transfer to the pot at a pressure-drop penalty of only 0.68%. These findings identify axial position as the primary parameter governing thermal performance. A single fin near the hot inlet offers a low-cost, retrofit-ready route to higher cookstove efficiency, aligned with Sustainable Development Goal 7.

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1. Introduction

Biomass cookstoves remain the primary cooking device in many developing countries, accounting for roughly 90% of household energy use [1]. In Thailand, traditional cookstoves with a cylindrical combustion chamber heated from the base and a flat-bottom pot resting on the upper opening are widely used in commercial settings such as noodle shops, street stalls and community kitchens. Hot gas rises along the pot, but much of it escapes through the chimney or leaks through the uninsulated walls resulting in reported thermal efficiencies below 20% [1 – 3]. The smooth pot wall offers insufficient surface area for efficient

convection, so most combustion energy never reaches the pot and thereby increasing both fuel use and cooking time.

Most attempts to raise cookstove efficiency target the combustion chamber. MacCarty & Bryden [4] report 8–15% gains from reshaping the air inlet. Pande *et al.* [5] reach 13.6% by adjusting the inlet-to-pot area ratio (0.18–0.28), and Chibuzor *et al.* [6] gain 6–9% with wall insulation. However, none of these approaches disturbs the flow inside the annular gap where most pot-side convective transfer occurs. Ring-type turbulators and annular fins are well-studied as passive enhancers. Boonloi [7] identifies an optimum of approximately BR = 0.20–0.25 in square channels (Re = 5,000 – 25,000). Eiamsa-ard & Promvong

[8] measure $TPF = 1.05\text{--}1.25$ in round tubes with serrated twisted tapes ($Re = 4,000\text{--}20,000$). Promvonge & Eiamsa-ard [9] and Promvonge [10] report similar TPF or Nusselt enhancements of 197–333% with conical-ring inserts. These conditions are far from the low- Re , impingement-dominated flow inside a cookstove annular gap.

The Thermal Performance Factor (TPF) compares enhanced and baseline configurations at equal pumping power. Defined as $TPF = \frac{Nu/Nu_0}{(f/f_0)^{1/3}}$, it is the standard figure-of-merit when both heat transfer and pressure drop change simultaneously [8, 11]. $TPF > 1$ means the enhancement is worth its pumping penalty. Despite the maturity of fin and turbulator research, the cookstove literature has barely explored surface modifications inside the combustion chamber. Pande *et al.* [5] study inlet area but add no flow disturbers, while Kraus *et al.* [12] provide general extended-surface theory without cookstove specifics. Husain *et al.* [13] and Ghiwe *et al.* [14] couple combustion and conjugate heat transfer but focus on emissions and hydrodynamics. To the authors' knowledge, no parametric, OFAT-based mapping of annular ring fins (varying h_f , n , and z together) inside a flat-bottom cookstove has been reported. The present study fills this gap.

Combustion chemistry is not solved explicitly. The hot gas enters as a velocity–temperature boundary at 830°C and 2.5 m s^{-1} , taken directly from experimental measurements at steady state. This focuses computation on the convective field inside the annular gap (which lies downstream of the flame), keeps cost an order of magnitude lower than full LPG chemistry without changing the relative ranking of fin configurations [13, 14], and uses a physically realistic post-combustion inlet rather than an idealised one.

The present study uses three-dimensional steady-state CFD (ANSYS Fluent, SST $k\text{--}\omega$) to investigate wall-mounted annular ring fins inside a flat-bottom biomass cookstove. Thirteen configurations (C0–C12) follow a one-factor-at-a-time design [15], with fin height $h_f = 20\text{--}50\text{ mm}$ ($BR = 0.20\text{--}0.50$), tier count $n = 1\text{--}3$, and installation height $z = 150\text{--}450\text{ mm}$. The ranges span the literature optimum

for ring turbulators [7], the manufacturable retrofit limit, and the three axial regions (lower, middle, upper) of the 600 mm chamber. The CFD model is validated against measured wall temperatures at three vertical levels and the chimney outlet. To the authors' knowledge this is the first systematic CFD parametric map of wall-mounted ring fins inside a flat-bottom cookstove. The study also identifies axial installation height, rather than fin height or tier count, as the controlling design parameter and shows that a single fin near the inlet (C9, $z = 150\text{ mm}$) can outperform multi-tier configurations. This offers a low-cost retrofit route consistent with Sustainable Development Goal 7 (Affordable and Clean Energy).

2. Materials and Methods

Experimental Apparatus and Instrumentation

The cookstove system consists of a flat-bottom pot placed inside a cylindrical combustion chamber, as shown in Fig. 1(a). The pot has an outer diameter of 400 mm, a height of 600 mm, and a volumetric capacity of 60 litres. The combustion chamber has an inner diameter and height of 600 mm each, leaving a 100 mm annular gap between the pot outer surface and the chamber inner wall. Both components are fabricated from 3 mm thick mild steel (ASTM A36).

Hot air enters vertically through a 100 mm diameter inlet at the base centre, impinges on the pot bottom, spreads radially into the annular gap, flows upward along the pot surface, and exits through a 60 mm diameter chimney at the top.

Temperature measurements are acquired using sixteen K-type thermocouples (accuracy: $\pm 1.5^\circ\text{C}$) connected to a data logger interfaced with a personal computer. The thermocouples are arranged at three vertical levels within the combustion chamber — Level 1 (T1–T4), Level 2 (T5–T8), and Level 3 (T9–T12) — with four additional positions at the inlet, outlet, pot bottom, and water, as illustrated in Fig. 1(b)

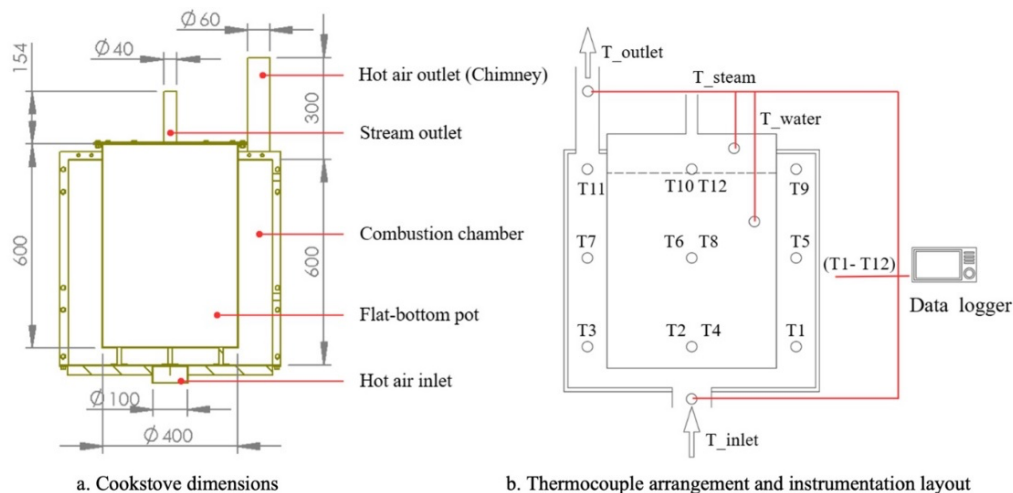


Fig. 1 Experimental cookstove.

Geometric Model for CFD Simulation

The three-dimensional computational domain was built in CAD to match the experimental apparatus described in Section 2.1, as shown in Fig. 2(a). Because this study

focuses on the hot-air flow behaviour, the computational domain covers only the fluid region in the annular space between the chamber wall and the pot wall, as shown in Fig. 2(b).

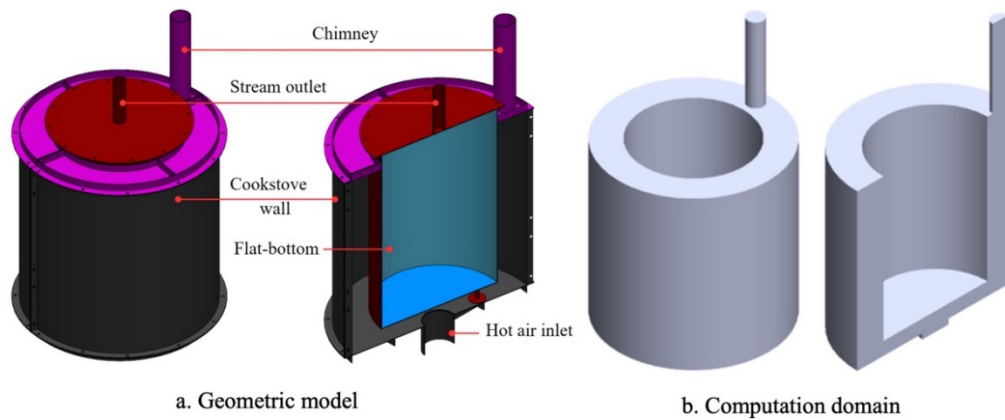


Fig. 2 CFD geometric model.

Annular Ring Fin Configurations

Annular ring fins welded to the inner combustion chamber wall serve as passive heat-transfer enhancement, as shown in Fig. 3. Each fin is a continuous 360° band protruding radially inward toward the pot, fabricated from the same material as the stove body. The combustion chamber and fins are fabricated from 3 mm thick mild steel (ASTM A36) with thermal conductivity of $k = 50.0 \text{ W m}^{-1} \text{ K}^{-1}$ [16]. In commercial practice, the cookpot is typically stainless steel or aluminium. Mild steel is used in the present CFD model as a wall-material proxy because the focus of this study is the hot-gas flow inside the annular gap, not the solid-side conduction. Because the wall is thin and the Biot number is small ($Bi \approx 0.001 \ll 1$), the relative TPF ranking is insensitive to the specific wall material within the range typical of metallic cookware ($k = 15\text{--}250 \text{ W m}^{-1} \text{ K}^{-1}$). Three geometric parameters are varied:

(i) Fin height (h_f): 20, 30, 40, and 50 mm, giving blockage ratios $BR = h_f/\delta = 0.20\text{--}0.50$ ($\delta = 100 \text{ mm}$). The lower bound follows Boonloi [7], who found optimum enhancement near $BR = 0.20\text{--}0.25$, while $BR = 0.50$ is the practical upper limit before excessive blockage [7, 8].

(ii) Number of tiers (n): 1, 2, and 3. A single tier is placed at $z = 300 \text{ mm}$; two tiers at $z = 200$ and 400 mm ; three tiers at $z = 150, 300$, and 450 mm . The spacing maintains a pitch-to-height ratio near $P/h_f \approx 6.0$, which provides a favourable thermal-hydraulic trade-off for ring-type turbulators [8–10].

(iii) Installation height: for the single-tier case, the ring is placed at $z = 150 \text{ mm}$ (lower), 300 mm (mid), or 450 mm (upper) to examine the effect of axial position relative to the base inlet.

The 13 configurations follow a one-factor-at-a-time (OFAT) approach [15] supplemented with selected factorial combinations. C1–C4 vary fin height at a single tier ($z = 300 \text{ mm}$). C2, C5, and C6 vary the number of tiers at $h_f = 30 \text{ mm}$. C2, C9, and C10 vary installation height at $h_f = 30 \text{ mm}$ with a single tier. The remaining cases (C7, C8, C11, C12) combine different heights and tier counts to capture interaction effects. The configurations are summarised in Table 1.

The design diverges from prior work in three ways. Where earlier cookstove studies adjusted only the inlet area [5] or added wall insulation [6], the present work introduces wall-mounted ring fins as in-flow turbulators that disturb the pot-wall boundary layer directly. The geometry impingement-driven and low-Re also departs from the straight ducts under fully turbulent flow used in prior turbulator studies [7, 8]. The parameter ranges follow the literature optimum ($BR = 0.20\text{--}0.50$), the manufacturable retrofit range ($n = 1\text{--}3$), and three axial zones along the 600 mm chamber ($z = 150/300/450 \text{ mm}$).

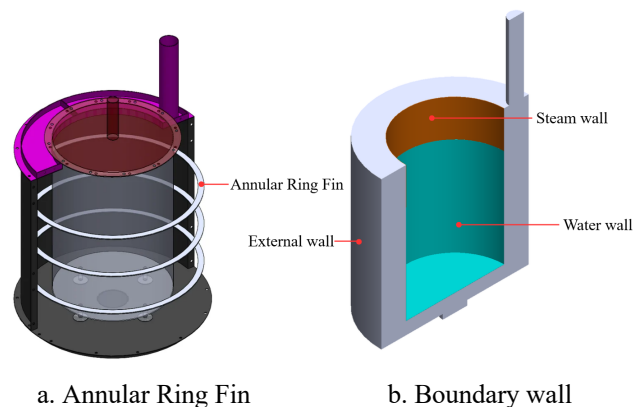


Fig. 3 Annular Ring Fin and Boundary wall

Table 1 Summary of annular ring fin configurations.

Config.	h_f (mm)	BR	Tiers	z (mm)	Description
C0	—	—	—	—	Baseline (no ring fin)
C1	20	0.20	1	300	Single tier, small protrusion
C2	30	0.30	1	300	Single tier, medium protrusion
C3	40	0.40	1	300	Single tier, large protrusion
C4	50	0.50	1	300	Single tier, maximum protrusion
C5	30	0.30	2	200, 400	Two tiers, medium protrusion
C6	30	0.30	3	150, 300, 450	Three tiers, medium protrusion
C7	40	0.40	2	200, 400	Two tiers, large protrusion
C8	40	0.40	3	150, 300, 450	Three tiers, large protrusion
C9	30	0.30	1	150	Single tier, lower position
C10	30	0.30	1	450	Single tier, upper position
C11	20	0.20	3	150, 300, 450	Three tiers, small protrusion
C12	50	0.50	2	200, 400	Two tiers, max protrusion

Governing Equations

The steady-state Reynolds-Averaged Navier–Stokes (RANS) equations were solved including continuity of mass (Eq. 1), continuity of momentum (Eqs. 2–4), and continuity of energy (Eq. 5), together with the SST k – ω turbulence

model transport equations for turbulent kinetic energy k (Eq. 6) and specific dissipation rate ω (Eq. 7) [13, 14]. Since the analysis was performed under steady-state conditions, all time-derivative terms were set to zero.

$$\frac{\partial \rho}{\partial t} = \text{div}(\rho \mathbf{u}) = 0 \quad (1)$$

$$\frac{\partial(\rho u)}{\partial t} + \text{div}(\rho u \mathbf{u}) = -\frac{\partial \rho}{\partial x} + \text{div}(\mu \text{grad } u) + S_{Mx} \quad (2)$$

$$\frac{\partial(\rho v)}{\partial t} + \text{div}(\rho v \mathbf{u}) = -\frac{\partial \rho}{\partial y} + \text{div}(\mu \text{grad } v) + S_{My} \quad (3)$$

$$\frac{\partial(\rho w)}{\partial t} + \text{div}(\rho w \mathbf{u}) = -\frac{\partial \rho}{\partial z} + \text{div}(\mu \text{grad } w) + S_{Mz} \quad (4)$$

$$\begin{aligned} \rho \frac{DE}{Dt} = & -\text{div}(p \mathbf{u}) + \left[\frac{\partial(u\tau_{xx})}{\partial x} + \frac{\partial(u\tau_{yx})}{\partial y} + \frac{\partial(u\tau_{zx})}{\partial z} + \frac{\partial(v\tau_{xy})}{\partial x} + \frac{\partial(v\tau_{yy})}{\partial y} + \frac{\partial(v\tau_{zy})}{\partial z} + \frac{\partial(w\tau_{xz})}{\partial x} \right. \\ & \left. + \frac{\partial(w\tau_{yz})}{\partial y} + \frac{\partial(w\tau_{zz})}{\partial z} \right] + \text{div}(k \text{grad } T) + S_E \end{aligned} \quad (5)$$

The SST k – ω model was selected for its superior near-wall resolution in flows involving impingement, separation, and recirculation [7, 17, 18]. Transport equations:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho u_j k)}{\partial x_j} = P - \beta^* \rho \omega k + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_k \mu_t) \frac{\partial k}{\partial x_j} \right] \quad (6)$$

$$\frac{\partial(\rho \omega)}{\partial t} + \frac{\partial(\rho u_j \omega)}{\partial x_j} = \frac{\gamma}{\nu_t} P - \beta \rho \omega^2 + \frac{\partial}{\partial x_j} \left[(\mu + \sigma_\omega \mu_t) \frac{\partial \omega}{\partial x_j} \right] + 2(1 - F_1) \frac{\rho \sigma_{\omega 2}}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \quad (7)$$

where P_k represents turbulent production. Model constants [17]: $\beta^* = 0.09$, $\alpha = 0.44$, $\beta = 0.0828$, $\sigma_k = 1.0$, $\sigma_\omega = 0.856$, $Pr_t = 0.85$.

Boundary Conditions

The boundary conditions are summarised in Table 2. Hot air enters the combustion chamber through the base inlet as a velocity inlet with uniform velocity and temperature derived from experimental measurements at steady-state conditions. The chimney outlet is defined as a pressure outlet with zero gauge pressure. The pot inner surface (wall1) and upper side wall (wall2) were assigned convective boundary conditions representing heat transfer to water and steam, respectively. The chamber outer wall was subjected to natural convection to the ambient environment at 34 °C. The working fluid was modelled as air with temperature-dependent thermophysical properties from the ANSYS Fluent material database [18]. Combustion was not explicitly resolved; the heat source was represented as a velocity-inlet boundary condition at 830 °C and 2.5 m s^{-1} , consistent with experimentally measured values at T_{inlet} .

The Reynolds number was calculated using the hydraulic diameter of the annular gap as the characteristic length ($D_h = 2\delta = 200 \text{ mm}$). The bulk Re in the annular gap was $\sim 1,560$, while $Re_{\text{inlet}} \approx 2,100$ at the impingement zone and $Re_{\text{fin}} \approx 2,200$ beneath a fin with $BR = 0.30$ due to a 30% cross-section reduction. The inlet turbulence intensity was prescribed as $Tu = 5\%$ with turbulent viscosity ratio = 10. The SST $k-\omega$ model with enhanced wall treatment was retained because the impinging jet and fin wakes generate locally turbulent conditions, and the model automatically switches to a low- Re formulation where $y^+ < 5$ [18].

The Boussinesq approximation held within the annular gap. The bulk temperature variation (250–340 °C) gave $\beta \cdot \Delta T \approx 0.08$, below the 0.1 validity threshold [19]. The 830 °C inlet was confined to a small impingement zone ($< 5\%$ of the domain) and did not dominate the buoyancy balance. Thermal radiation was likewise omitted. The post-combustion CFD domain excludes soot and flames which are the principal participating-media sources [13, 14]. Convection dominates radiation in analogous pot-enclosed cookstoves [20], and the residual offset is systematic and cancels in the relative TPF, Q_{pot} , and ΔP comparisons [21].

Table 2 Boundary conditions used in the CFD simulation.

Type	Details
Fluid	Air
Solid	ASTM A36 mild steel
Inlet (temperature)	830°C
Inlet (velocity)	2.5 m s^{-1}
Outlet (gauge pressure)	0 Pa.
Chamber wall temperature	36°C
Pot wall (steam space)	120°C
Pot wall (water space)	100°C
Thermal conductivity k	$50.0 \text{ W m}^{-1} \text{ K}^{-1}$

Mesh Generation and Independence Study

Five mesh densities (252,139–2,694,983 elements) were tested for grid independence, using chimney-outlet temperature T_{out} as the convergence indicator (Fig. 4). Because T_{out} is a globally integrated quantity, any unresolved upstream feature shows up as a deviation here, making it a suitable single-parameter test [21]. T_{out} changed by less than 1% between Mesh-3 (607,587 elements) and the finest Mesh-5 (2,694,983 elements). Therefore Mesh-3 was adopted as the production grid (Fig. 5(a)), balancing accuracy against computational cost. Inflation layers on all solid surfaces (including the ring fins) were sized for $y^+ \approx 1$ (Fig. 5(b)), and the steam, outer, and water wall regions were assigned separate boundary conditions.

All simulations were conducted under steady-state assumptions. Experimental measurements confirmed that the water reached boiling point at 100 °C at the 18th minute of the experiment. Temperature values recorded at this instant were adopted as thermal boundary conditions for the CFD model.

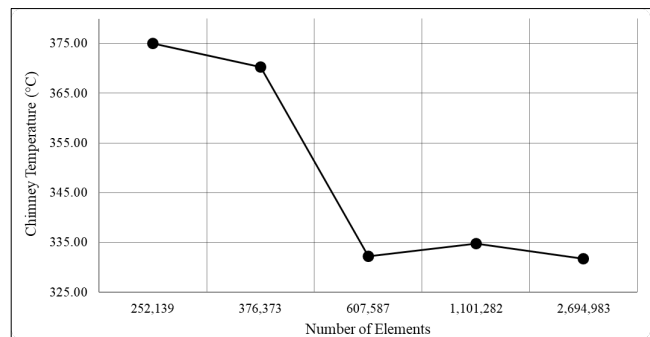


Fig. 4 Mesh Independence.

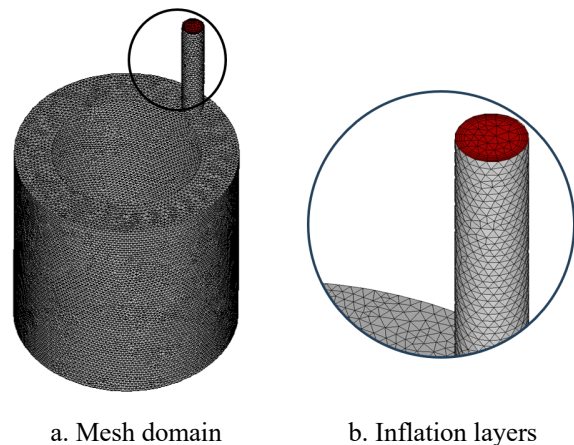


Fig. 5 Computational mesh.

Model Validation

The numerical model was validated by comparing CFD predictions of the baseline configuration (C0) with experimental measurements at the 18th minute, the time at which the water reached boiling point at 100 °C. The comparison includes level-averaged wall temperatures at three vertical positions along the combustion chamber and the chimney outlet temperature.

As shown in Table 3, the CFD model consistently overpredicted temperatures, with relative errors of 28.21%, 24.88%, and 16.13% at levels L1, L2, and L3, respectively. The overprediction is caused by the simplified boundary conditions, which assume uniform hot air entry at 830 °C, whereas actual combustion produces non-uniform temperature and velocity distributions. Additionally, radiation heat loss from the outer wall was not modelled.

Despite these deviations, the decreasing error trend from L1 to the outlet indicates improved prediction as the flow develops. The chimney outlet temperature was predicted with a relative error of only 2.21%, confirming that the model captures the overall thermal behaviour of the system.

Direct comparison of thermal efficiency between steady-state CFD and the cumulative experimental method was not attempted, as the two approaches represent fundamentally different energy accounting methods. The experimental thermal efficiency of the baseline stove was 40.74%, based on 1.36 kg of LPG input and the total energy absorbed by 50 L of water. This value serves as the reference for evaluating relative performance improvements of each fin configuration.

Table 3 Comparison of temperature between experimental and CFD results for baseline configuration (C0).

Location	T _{exp} (°C)	T _{CFD} (°C)	Relative Error (%)
L1 (T1–T4)	341.25	437.50	28.21
L2 (T5–T8)	262.25	327.50	24.88
L3 (T9–T12)	294.50	342.00	16.13
T _{outlet}	325.00	332.19	2.21

Performance Evaluation Parameters

The thermal performance of each fin configuration is evaluated relative to the baseline (C0). The average Nusselt number is defined as:

$$Nu_{avg} = \frac{h_{avg} \cdot D_h}{k_f} \quad (8)$$

where h_{avg} is the average heat transfer coefficient, D_h is the hydraulic diameter, and k_f is the fluid thermal conductivity [22].

The thermal performance factor, which accounts for both heat transfer enhancement and pressure drop penalty, is expressed as:

$$TPF = \frac{Nu_{fin}/Nu_0}{(f_{fin}/f_0)^{1/3}} \quad (9)$$

where Nu_{fin} and Nu_0 are the average Nusselt numbers of the finned and baseline configurations, f_{fin} and f_0 are the corresponding friction factors [8].

A TPF greater than unity indicates a net thermal-hydraulic benefit over the baseline configuration.

The thermal-efficiency ratio η_{th} in Table 4 is defined as $\eta_{th} = Q_{pot,i} / Q_{pot,0}$, where $Q_{pot,i}$ is the pot-side heat-transfer rate of configuration i and $Q_{pot,0} = 4,236$ W is that of the baseline. It is dimensionless and represents the heat-transfer enhancement of each finned configuration relative to the baseline, independent of pressure-drop penalty.

3. Results and Discussion

Overall Simulation Results

Table 4 summarises the thermal-hydraulic performance of all thirteen configurations. The baseline (C0) yields $Q_{pot} = 4,236$ W with $\Delta P = 54.94$ Pa. Among all configurations, C9 (single fin, $h = 30$ mm, $z = 150$ mm) achieves the highest heat transfer rate of 4,832 W and the highest TPF of 1.138, while C3 (single fin, $h = 40$ mm, $z = 300$ mm) shows negligible improvement with $TPF \approx 1.000$.

TPF was selected as the principal performance metric because it integrates both heat-transfer enhancement and pressure-drop penalty into a single dimensionless ranking. Standard tube correlations (Dittus–Boelter, Gnielinski) are not used as a benchmark because they were derived for fully developed turbulent flow in straight tubes.

The relatively small pressure-drop variations across all 13 configurations ($\Delta P = 55.04$ – 57.18 Pa, corresponding to only 0.2–4% above the baseline value of 54.94 Pa) are physically reasonable. The annular gap is wide (100 mm × 600 mm) compared to the fin obstruction, so even at BR = 0.50 the fin reduces only a thin band of the total flow cross-section. Moreover, the baseline ΔP is dominated by inlet/outlet contraction–expansion and chimney losses rather than by friction along the annular gap; the fins therefore add only a small incremental contribution to the overall pressure loss.

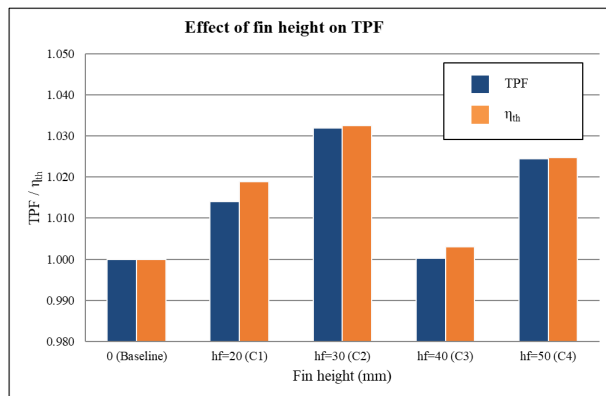
Table 4 Summary of CFD results for all configurations.

Config.	Description	Q_{pot} (W)	ΔP (Pa)	η_{th}	TPF
C0	Baseline	4236	54.94	1.000	1.000
C1	$h = 20$, 1-tier, $z = 300$	4316	55.73	1.019	1.014
C2	$h = 30$, 1-tier, $z = 300$	4374	55.04	1.032	1.032
C3	$h = 40$, 1-tier, $z = 300$	4249	55.41	1.003	1.000
C4	$h = 50$, 1-tier, $z = 300$	4341	54.99	1.025	1.024
C5	$h = 30$, 2-tier	4425	55.28	1.045	1.043
C6	$h = 30$, 3-tier	4750	55.43	1.121	1.118
C7	$h = 40$, 2-tier	4475	55.22	1.056	1.055
C8	$h = 40$, 3-tier	4815	55.84	1.137	1.130
C9	$h = 30$, 1-tier, $z = 150$	4832	55.31	1.141	1.138
C10	$h = 30$, 1-tier, $z = 450$	4250	54.47	1.003	1.006
C11	$h = 20$, 3-tier	4677	55.37	1.104	1.101
C12	$h = 50$, 2-tier	4555	55.21	1.075	1.073

Note: $\eta_{\text{th}} = Q_{\text{pot},i} / Q_{\text{pot},0}$ (dimensionless thermal-efficiency ratio relative to baseline C0).

Effect of Fin Height

Fig. 6 shows the effect of fin height on heat transfer for a single-tier fin at $z = 300$ mm. Increasing the fin height from 0 to 30 mm (BR = 0.30) improves Q_{pot} from 4,236 W to 4,374 W (TPF = 1.032). However, at $h = 40$ mm (BR = 0.40), the heat transfer drops sharply to 4,249 W (TPF = 1.000), indicating that the fin obstructs the flow rather than enhancing convection. At $h = 50$ mm (BR = 0.50), the performance recovers slightly to TPF = 1.024. This non-monotonic trend suggests that an optimal blockage ratio exists near BR = 0.30, beyond which flow restriction offsets the benefit of increased fin surface area.

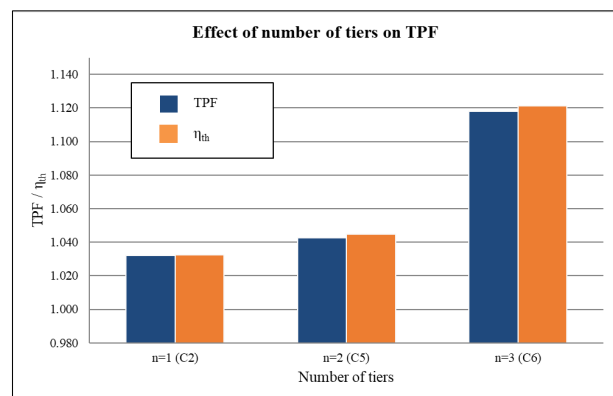
**Fig. 6** Effect of fin height on TPF ($h_f = 0$ –50 mm).

This non-monotonic trend reflects a balance between boundary-layer disruption and flow obstruction. At BR = 0.30 the separation bubble reattaches quickly and enhances convection whereas at BR = 0.40 the bubble elongates into a low-velocity dead zone that weakens convection while pressure drop rises. The slight recovery at $h_f = 50$ mm occurs because the further cross-section contraction accelerates the

flow and re-establishes impingement on the pot wall, while the 67% larger fin surface adds extended-surface conduction. These partially offset the higher form drag but remain below the BR = 0.30 optimum.

Effect of Number of Fin Tiers

Fig. 7 shows the effect of fin tiers on TPF for $h = 30$ mm fins. Increasing from 1-tier (C2) to 2-tier (C5) raises Q_{pot} from 4,374 W to 4,425 W (TPF from 1.032 to 1.043). A significant improvement is observed with 3-tier (C6), which achieves $Q_{\text{pot}} = 4,750$ W and TPF = 1.118. The pressure drop increase is minimal across all cases (55.04–55.43 Pa), indicating that multiple tiers enhance heat transfer primarily by increasing the total fin surface area and creating repeated flow disturbances without significantly restricting the flow passage.

**Fig. 7** Effect of number of tiers on TPF (1–3 tiers).

Effect of Fin Installation Height

Fig. 8 shows the effect of installation height on TPF for a single 30 mm fin at three positions. Placing the fin at $z =$

150 mm (C9) produces the highest Q_{pot} of 4,832 W with TPF = 1.138, whereas $z = 300$ mm (C2) gives TPF = 1.032 and $z = 450$ mm (C10) gives only TPF = 1.006. The superior performance at the lower position is due to the higher gas temperature and velocity near the inlet, which maximises convective heat transfer at the fin surface. At $z = 450$ mm, the gas has already cooled considerably, and the fin contributes little additional heat transfer.

The 0.68% pressure-drop increase in C9 reflects the local velocity at $z = 150$ mm, where the impinging jet has just deflected radially and the axial flow is still slow and weakly developed. At $z = 300$ and 450 mm the flow is fully developed and accelerated by chimney contraction, producing higher form drag.

Each fin generates four characteristic flow features: an upstream stagnation region, a horseshoe vortex at the fin-wall junction, a separation bubble extending $3\text{--}4 h_f$ downstream, and a reattachment line where the thermal boundary layer is thinned by approximately 40%. The local heat-transfer peak therefore occurs downstream of the fin at the reattachment line, not at the upstream stagnation point [9].

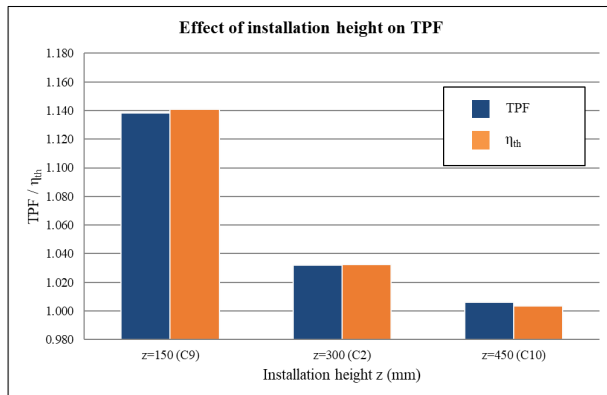


Fig. 8 Effect of installation height on TPF ($z = 150\text{--}450$ mm).

Optimal Configuration

Fig. 9 compares the temperature contours of the baseline (C0) and the best case (C9). Configuration C9 (single fin, $h = 30$ mm, $z = 150$ mm) achieves the best overall performance with a TPF of 1.138, corresponding to a 14.1% increase in heat transfer over the baseline with only a 0.68% increase in pressure drop. This result shows that placing a fin near the high-temperature inlet zone is more effective than increasing the number of tiers or fin height. However, C6 (3-tier, $h = 30$ mm) and C8 (3-tier, $h = 40$ mm) also show strong performance (TPF = 1.118 and 1.130), suggesting that multi-tier configurations may offer comparable benefits when the lower position is not feasible.

In C6 the upper two fins operate in a flow already disturbed by the first fin, so their incremental contribution is small while the cumulative pressure drop rises faster than the heat-transfer gain — therefore fin placement matters more than fin count, and C9 (single, $z = 150$ mm) outperforms C6 (3-tier).

Although C8 (3-tier, $h_f = 40$ mm, TPF = 1.130) and C9 (single, $h_f = 30$ mm, $z = 150$ mm, TPF = 1.138) deliver nearly identical heat-transfer rates, they reach this result by very different mechanisms. C8 distributes the enhancement across three fins, accumulating a pressure-drop penalty of $\sim 4\%$ ($\Delta P = 57.18$ Pa), whereas C9 concentrates the enhancement at the high-velocity inlet zone using a single fin and pays only 0.68% ($\Delta P = 55.31$ Pa). The 0.71% TPF gap therefore reflects the pressure-drop advantage of *placement* over *proliferation*. From a retrofit standpoint C9 is preferred because it is simpler to fabricate and imposes minimal back-pressure on the combustion process. Fig. 10 compares the temperature contours of C2 ($h = 30$ mm) and C3 ($h = 40$ mm), visualising the non-monotonic fin-height effect.

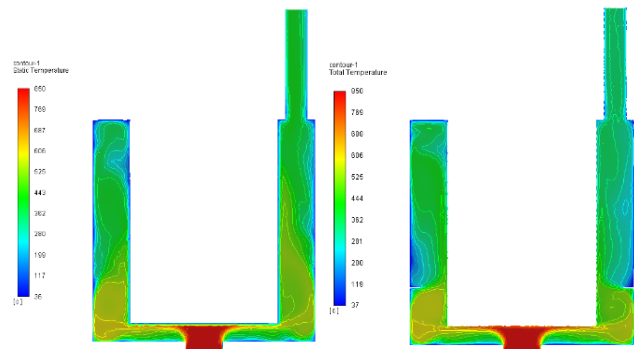


Fig. 9 C0 (baseline) vs C9 (best case).

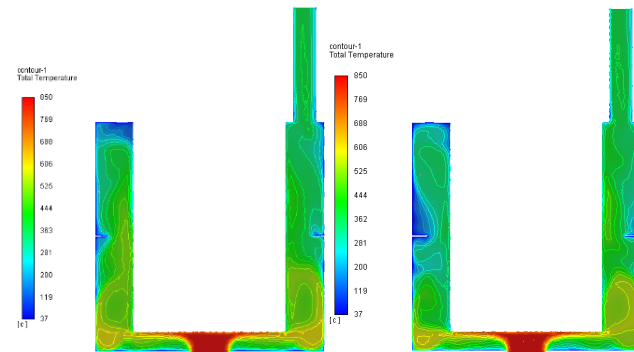


Fig. 10 C2 vs C3 ($h=30$ vs $h=40$ mm).

4. Conclusion

This study examined the thermal-hydraulic performance of wall-mounted annular ring fins on a flat-bottom cookstove using three-dimensional steady-state CFD simulations with the SST $k\text{--}\omega$ turbulence model. Thirteen configurations (C0–C12) were analysed using the OFAT approach, varying fin height (20–50 mm), number of tiers (1–3), and installation height ($z = 150\text{--}450$ mm). The key findings are as follows:

1. Fin height exhibits a non-monotonic effect on heat transfer. The optimal blockage ratio is $BR = 0.30$ ($h = 30$ mm), yielding TPF = 1.032. Beyond this, flow obstruction reduces the thermal benefit.

2. Increasing the number of fin tiers consistently improves heat transfer with minimal pressure drop penalty. A 3-tier configuration with $h = 30$ mm (C6) achieves $TPF = 1.118$.

3. Installation height has the most significant influence on performance. A single fin at $z = 150$ mm (C9) outperforms all multi-tier configurations at $z = 300$ mm, achieving $TPF = 1.138$ with a 14.1% increase in Q_{pot} and only 0.68% increase in ΔP . Placing a fin in the high-temperature zone near the inlet is more effective than adding multiple fins at higher positions.

4. These results provide design guidelines for enhancing cookstove thermal performance through simple passive fin modifications without significant flow resistance.

Several caveats should be acknowledged. Wall-validation errors reached 16–28% under the uniform-inlet boundary condition. In addition, the simulations were performed under steady-state conditions, and did not include explicit combustion or radiation modelling. Furthermore, the OFAT design cannot resolve parameter interactions. Planned follow-up work will tackle these through experimental validation of C9, a DOE/Taguchi interaction study, coupled combustion–radiation modelling, alternative fin shapes (helical, perforated, slotted), and scale-up to 100 litre commercial stoves. Nevertheless, the proposed retrofit aligns with SDG 7 (Affordable and Clean Energy) and SDG 13 (Climate Action), cutting fuel use and emissions at negligible manufacturing cost.

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6. Declaration of Generative AI in Scientific Writing

The use of a generative AI tool in this study was limited to facilitating literature searches and enhancing grammatical accuracy and linguistic clarity.

7. CRediT Author Statement

Komsan Wongkalasin: Conceptualization, Methodology, Data curation, Investigation, Writing-Original draft preparation, Visualization, Writing -Reviewing and Editing. **Somchat Sonasang:** Conceptualization, Validation. **Chuthong Summatta:** Methodology, Data curation, Supervision.

8. Research Involving Human and Animals Rights

Not applicable

9. Ethics Approval and Consent to Participate

Not applicable

10. Declaration of Competing Interest

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

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