

Ventilation feasibility envelopes for generator rooms with restricted openings and hot air recirculation

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Abstract

A generator room can satisfy an inlet temperature limit while failing an independent airflow requirement. This analytical and computational study couples a prescribed fan curve, opening resistance and steady recirculation heat balance in a hypothetical room. It evaluates 3348 operating cases and an eight parameter uncertainty box with 256 corners. Under assumed limits of 12 cubic metres per second and 50 degrees Celsius, the nominal opening bound is 0.375 and its conservative counterpart is 0.567. At a 45 degree ambient temperature and full opening, maximum recirculation decreases from 0.188 nominally to 0.082 at the worst corner. Of 2231 nominally feasible cases, 1095 fail the conservative assessment. A separate temperature dependent density sensitivity scales fan and resistance pressures together at fixed speed. It rejects 60 cases accepted by the nominal constant density calculation. Assuming fully open flow regardless of opening accepts 650 cases rejected by the density linked joint assessment. Independent nonlinear solutions verify the extension within the stated temperature band. The contribution is a reproducible comparison of hydraulic, thermal, uncertainty and density assumptions. Results do not establish coolant performance, engine derating or the suitability of an actual installation.

1 Introduction

Ventilation deficiencies are not completely described by the physical dimensions of an inlet or an indicated louver position. Pressure resistance, effective free area, outdoor conditions and the route taken by hot discharge air all influence the inlet conditions of a radiator cooled generator. Caterpillar describes a generator enclosure baffle that separates intake and discharge pathways to limit hot air recirculation (Caterpillar n.d.). That description establishes the relevance of the mechanism; it supplies no numerical input to the present room model. Fan testing terminology distinguishes operating points and pressure losses (Air Movement and Control Association International and ASHRAE 2025), supporting the use of a pressure balance rather than an assumption of constant flow through every opening.

Generator room thermal studies can require spatial and time dependent descriptions. Zeng et al. studied an earth air heat exchanger for an underground diesel generator room (Zeng et al. 2019). Their cooling arrangement and thermal objectives differ from the direct radiator discharge configuration examined here. The present analysis deliberately reduces the room to a well mixed steady system. Its purpose is to provide an interpretable screening calculation before a site specific airflow survey or computational fluid dynamics study. It does not reproduce the measurements or simulations of that research.

Two screening errors motivate the study. First, a moderate predicted inlet temperature can conceal inadequate airflow. Second, a nominal calculation can pass although an explicitly chosen combination of pressure, heat and density uncertainties fails. The research question is therefore whether both constraints can be expressed as a joint opening and recirculation envelope, and how much that envelope changes under a stated uncertainty set.

The contribution is a reproducible coupling of standard pressure and energy balances, a closed form recirculation bound, and a conservative parameter audit. The physical balances and the idea of avoiding recirculation are established principles. No claim is made that this is the first ventilation model or that its scenario values represent a particular generator. Its outputs also compare two simplifying baselines with a density linked sensitivity, to quantify when those simplifications change joint acceptance.

2 Materials and methods

2.1 Hypothetical flow and heat paths

The modeled fan passes a volumetric flow v through a room and radiator. A fraction r of the radiator discharge returns to the room inlet mixing region, while the remaining fraction is replaced by outdoor air at temperature T_a . Heat Q_r is added before the radiator; heat Q_d is added across the radiator. The mixed radiator inlet and outlet temperatures are T_i and T_o . The same constant air density ρ and heat capacity c_p apply throughout the path. Combustion air, wall conduction, humidity changes, fan heat, buoyancy and spatial hot spots are excluded. The circuit has one aggregate pressure resistance; a change in recirculation does not itself alter this resistance.

Effective opening fraction α is the available effective area divided by its fully open effective area. It is not blade angle, actuator travel or gross facade area. The opening resistance coefficient scales with the inverse square of α by model assumption. The fan and external restriction pressure relations are:

$$\Delta p_{\text{fan}} = p_0 - a v^2; \quad \Delta p_{\text{path}} = w + \left(b + \frac{\ell}{\alpha^2}\right) v^2 \quad (1)$$

Here p_0 is the fan pressure intercept, a is the fan curve coefficient, b is the fixed external resistance coefficient, ℓ is the fully open opening coefficient, and w is an opposing constant pressure. The coefficients refer to the actual modeled pressure curve. Density is varied independently as a conservative scenario input; no physical fan similarity law is asserted for the entire uncertainty box. The coefficients a and b are nonnegative, ℓ is positive, and their units are $\text{Pa s}^2 \text{m}^{-6}$ when v is in cubic metres per second. With p_0 greater than w and $0 < \alpha \leq 1$, the model has one positive pressure balance root:

$$v = \sqrt{\frac{p_0 - w}{a + b + \frac{\ell}{\alpha^2}}} \quad (2)$$

The numerical heat inputs are in kW and c_p is in kJ per kilogram kelvin, so $\rho c_p v$ has units of kW per kelvin. The inlet mixing and radiator balances are:

$$T_i = (1 - r)T_a + rT_o + \frac{Q_r}{\rho c_p v}; \quad T_o = T_i + \frac{Q_d}{\rho c_p v} \quad (3)$$

Solving these two simultaneous equations gives:

$$T_i - T_a = \frac{Q_r + r Q_d}{(1 - r) \rho c_p v} \quad (4)$$

The factor 1 divided by 1 minus r represents repeated recirculation within the steady idealized path. A model with r approaching one has vanishing outdoor heat removal and an unbounded temperature rise. The evaluated range stops at r equal to 0.35. These temperatures do not incorporate a coolant temperature model or a radiator effectiveness curve.

2.2 Joint feasibility and the conservative corner

Feasibility means v is at least v_{min} and T_i is at most T_L. These are independently imposed scenario requirements. Neither the 12 cubic metres per second flow threshold nor the 50 degree inlet threshold is attributed to a manufacturer or engineering standard. For an ambient temperature below T_L, define H as ρ c_p v times T_L minus T_a. The temperature requirement is equivalent to:

$$H = \rho c_p v (T_L - T_a); \quad r \leq \frac{H - Q_r}{H + Q_d} \quad (5)$$

For nonnegative heat inputs and 0 ≤ r < 1, H smaller than Q_r means that no nonnegative recirculation satisfies the thermal limit. At H equal to Q_r and positive heat input, only r equal to zero qualifies. A positive recirculation bound still does not satisfy the separate flow limit automatically. The opening fraction needed to satisfy the flow threshold is:

$$\alpha \geq \sqrt{\frac{\ell}{\frac{p_0 - w}{v_{\min}^2} - a - b}} \quad (6)$$

With ell positive, a nonpositive denominator means that no finite opening satisfies the flow threshold. A positive denominator gives a finite bound; a required fraction greater than one makes the available fully open geometry insufficient. At p₀ no greater than w, the assumed circuit has no positive flow solution and the temperature formulas must not be evaluated.

Flow increases with p₀ and decreases with w, a, b and ell. Inlet temperature increases with Q_r and Q_d and decreases with ρ and flow; it also increases with r over the evaluated nonnegative range. Consequently the minimum flow and maximum temperature over the chosen rectangular uncertainty set occur at the same corner: minimum p₀ and ρ and maximum w, a, b, ell, Q_r and Q_d. The corner is a conservative bound for the entire stated box, not a statistical confidence limit. Dependencies among real fan and density parameters could make this box overly conservative.

2.3 Scenario design

Table 1 specifies every uncertain input. The scenario constants are cp equal to 1.005 kJ per kilogram kelvin, vmin equal to 12 cubic metres per second, and TL equal to 50 degrees Celsius. No input is fitted to client records, equipment tests or weather observations. Ambient temperatures of 35, 40 and 45 degrees Celsius are selected stress conditions, not an estimate of a city climate distribution.

Table 1
Declared hypothetical inputs and uncertainty limits

Quantity	Unit	Nominal	Lower	Upper
Air density rho	kg per cubic metre	1.15	1.05	1.25
Fan pressure intercept p0	Pa	800	700	900
Opposing pressure w	Pa	0	0	100
Room heat Qr	kW	35	25	45
Radiator heat Qd	kW	260	220	300
Fan coefficient a	Pa s ² m ⁻⁶	1.6	1.4	1.8
Fixed coefficient b	Pa s ² m ⁻⁶	0.4	0.3	0.5
Opening coefficient ell	Pa s ² m ⁻⁶	0.5	0.4	0.6

The grid combines 31 equally spaced opening fractions from 0.25 to 1.00, 36 equally spaced recirculation fractions from zero to 0.35, and the three ambient temperatures. There are 3348 cases. A grid case is accepted conservatively only if both limits hold at the worst corner. The number of accepted grid points depends on this deterministic grid and has no interpretation as a field pass probability.

2.4 Computation and verification

Python with NumPy 2.3.5 and SciPy 1.17.0 was used; SciPy methods are described by Virtanen et al. (2020). For every grid point, an independent scalar root calculation solves the pressure balance, and a two by two linear system solves the coupled temperature balances. At 27 selected combinations of ambient temperature, opening and recirculation, all 256 uncertainty corners are enumerated, giving 6912 corner evaluations. Boundary calculations perturb the analytical recirculation limit on both sides. A separate verifier reconstructs pressure and thermal residuals directly from the saved grid. Online Resource 1 provides scripts, inputs, tables and figures. Additional review checks recover the no-recirculation heat balance, zero thermal margin boundary and fan pressure balance, and reject inputs outside the stated domain.

2.5 Density linked sensitivity and controlled baselines

The original uncertainty box varies actual fan coefficients and density independently. A separate sensitivity instead defines every pressure coefficient at reference density rho0 equal to 1.15 kg per cubic metre. Ambient pressure P is assumed to be 101325 Pa and dry air gas constant Rg is 287.1 J per kilogram kelvin. Density is evaluated at the mixed inlet temperature. At fixed speed, the fan pressure

curve and both quadratic path losses scale by rho divided by rho0; opposing pressure w remains fixed. This small density change approximation is consistent with the incompressible conversion described in Section 8.9 of the fan-testing standard (Air Movement and Control Association International and ASHRAE 2025). It is an aggregate inlet density assumption, not a spatially varying radiator model:

$$\rho(T_i) = \frac{P}{R_g(T_i + 273.15)} \quad (7a)$$

$$v(T_i, \alpha) = \sqrt{\frac{P_0 - \frac{w \rho_0}{\rho(T_i)}}{a + b + \frac{\ell}{\alpha^2}}} \quad (7b)$$

The heat balance then becomes a scalar nonlinear equation:

$$\rho(T_i) c_p v(T_i, \alpha) (T_i - T_a) = \frac{Q_r + r Q_d}{1 - r} \quad (8)$$

Only temperatures between Ta and TL are assessed for acceptance. Across that band the density conversion remains within ten percent of the reference. Let K equal Ti plus 273.15, Ka equal Ta plus 273.15, and beta equal w rho0 Rg divided by P. The heat-removal capacity is proportional to (K minus Ka) times the square root of p0 minus beta K, divided by K. It increases strictly if 2 Ka p0 exceeds beta K times (K plus Ka). The script verifies the sufficient bound at the largest K in all 6696 nominal and adverse coefficient cases. Thus evaluating capacity at TL determines thermal feasibility; thermally feasible cases have one root within the band. Actual flow at that root is then checked separately. No equilibrium outside the admissible band is used for acceptance.

Two controlled baselines use the same reference coefficients and heat inputs. The first fixes density at rho0 while retaining opening resistance. The second additionally fixes flow at its fully open value for every opening. Neither is described as a field practice. Independent simultaneous pressure and heat solutions check the scalar roots; at w equal to zero a separate closed form checks the temperature. This sensitivity does not replace or reduce the original eight parameter uncertainty bound.

2.6 AI assistance in study preparation

OpenAI ChatGPT, accessed on 4 and 5 October 2026, assisted substantially with topic development, model formulation, code, calculation checks, literature discovery, drafting, figure preparation from computed results, and Arabic translation. The results were calculated by the supplied scripts rather than asserted as measurements. AI is not an author and does not replace human accountability. Further assistance on 5 October 2026 covered journal-specific formatting, reference presentation and submission preparation.

3 Results

3.1 Distinct airflow and thermal restrictions

The nominal flow threshold requires alpha of at least 0.375. At the worst corner the corresponding bound is 0.566947, a 51.2 percent increase in effective opening fraction. This is a dimensionless scenario comparison, not a recommended louver opening for a site. Figure 1 plots the independent thermal recirculation envelopes and marks the conservative airflow boundary. Regions left of that flow boundary cannot be accepted even when the plotted thermal limit permits positive recirculation.

Table 2 illustrates the mechanism at 45 degrees ambient temperature. With alpha equal to 0.50 and no recirculation, the conservative inlet temperature is 48.77 degrees, below the imposed 50 degree limit. However, flow is only 11.30 cubic metres per second, so the case fails. At full opening and r equal to 0.10, airflow is adequate but the conservative inlet temperature is 50.49 degrees, so temperature alone causes failure.

Table 2
Selected 45 degree ambient cases

Opening fraction	Recirculation	Nominal flow	Worst flow	Nominal inlet C	Worst inlet C	Conservative result
0.50	0.00	14.14	11.30	47.14	48.77	Flow fails
0.50	0.10	14.14	11.30	49.15	51.99	Both fail
1.00	0.00	17.89	14.38	46.69	47.96	Pass
1.00	0.10	17.89	14.38	48.28	50.49	Temperature fails

Flow columns in Table 2 are in cubic metres per second. At full opening and 45 degree ambient temperature, the nominal thermal limit allows r up to 0.188163, whereas the conservative limit allows only 0.082186. Figure 2 shows the two constraint boundaries in the opening and recirculation plane. It demonstrates that expanding an opening and preventing discharge recirculation affect different parts of the admissible domain.

3.2 Grid audit and numerical checks

Of the 3348 grid cases, 2231 pass nominally and 1136 pass throughout the declared uncertainty box. There are 1095 cases that pass nominally but fail conservatively, equivalent to 49.1 percent of the nominally accepted grid cases. The percentage is a property of the selected grid, not a population estimate. All conservatively accepted cases also pass nominally.

The largest numerical difference between a closed form value and the independent pressure or simultaneous temperature solution is 4.94 times ten to the minus thirteen in the corresponding output unit. Enumerating the corners produces exactly the same worst flow and temperature as the monotonic corner rule at all 27 selected conditions. The largest directly reconstructed pressure or heat balance

residual is 2.27 times ten to the minus thirteen in its respective Pa or kW unit. All 27 boundary checks pass. Numerical agreement establishes implementation consistency within the specified equations and does not validate their physical approximation.

3.3 Acceptance changes under density coupling

Table 3 compares the controlled calculations on the same 3348 point grid for each coefficient set. Density at the imposed 50 degree limit is 1.09214 kg per cubic metre. Joint nominal acceptance falls from 2231 to 2171 cases when density follows temperature. The 60 reclassifications are 2.69 percent of nominal constant density acceptances. The fixed fully open flow baseline accepts 650 points that the density linked assessment rejects; this illustrates the larger error caused by suppressing the opening dependent operating point.

Table 3
Density sensitivity and baseline acceptance

Coefficient set	Constant density accepted	Density linked accepted	Constant density accepted but linked failed	Fixed full flow accepted but linked failed
Nominal	2231	2171	60	650
Adverse reference coefficients	1244	1180	64	1083

The adverse reference calculation uses rho0, so its counts differ from the original conservative corner with rho equal to 1.05. These are different assumptions, not competing estimates of a field failure rate. Figure 3 isolates the thermal recirculation boundary at 45 degrees ambient temperature; the airflow constraint still applies. The 4220 thermally feasible roots agree with independent nonlinear solutions within 9.74 times ten to the minus thirteen in the corresponding flow or temperature unit. The zero opposing pressure closed form differs by at most 2.56 times ten to the minus thirteen degrees.

4 Discussion

The model explains why an inlet temperature check and an opening dimension check should not be substituted for a measured airflow assessment. Its airflow and temperature inequalities identify which information is missing from a site decision: the fan operating curve, aggregate pressure restriction, effective opening, room and radiator heat loads, and recirculation fraction. The fully open area alone is insufficient to determine the effective restriction, and actuator position alone is insufficient to determine alpha.

The screening approach differs from the earth air cooling configuration studied by Zeng et al. (2019). It also addresses a simpler question than laboratory fan characterization (Air Movement and Control Association International and ASHRAE 2025). A real installation can have nonuniform inlet temperature, multiple recirculation routes, different intake and exhaust restrictions, wind dependent mixing, exhaust

gas ingress and equipment protection logic. Those effects require measured or spatially resolved models. The selected 50 degree limit and constant heat rejection must be replaced by the applicable equipment data before a design decision. No coolant overheating temperature, permissible electrical output, alarm threshold or installation approval can be inferred from the present envelope.

The conservative corner combines uncertainties independently, which can exaggerate a physically attainable adverse condition. Conversely, omitted uncertainties can make it optimistic. A measured joint uncertainty set would improve the calculation. The density sensitivity demonstrates a measurable assumption effect within the admissible band, but retains one aggregate inlet density and fixed heat rejection. Pressure scaling cannot reproduce density differences across the radiator or a changing fan speed. The temperature divergence as r approaches one reflects the model's loss of outdoor heat removal and should not be extrapolated into a high temperature regime where constant density, constant heat load and the prescribed fan curve cease to apply.

5 Conclusions

Coupling an assumed fan curve with an opening resistance and recirculation heat balance produces separate airflow and inlet temperature feasibility bounds. In the hypothetical example, the conservative opening requirement rises from 0.375 to 0.567, while full opening recirculation allowance at 45 degrees ambient temperature falls from 0.188 to 0.082. Almost half of the nominally accepted deterministic grid points fail under the stated uncertainty box. Density coupling rejects 60 nominal constant density acceptances, while a fully open flow simplification accepts 650 points rejected by the joint linked assessment. Independent numerical checks support both formulations. The resulting envelope is useful for organizing measurement and sensitivity requirements, with application to an actual room conditional on equipment data and physical validation.

6 Nomenclature

Symbol	Meaning	Unit
v and v _{min}	Volumetric airflow and imposed minimum	cubic metres per second
alpha and r	Effective opening and recirculation fractions	dimensionless
p ₀ and w	Fan pressure intercept and opposing pressure	Pa
a b ell	Fan fixed and opening pressure coefficients	Pa s ² m ⁻⁶
rho and cp	Constant air density and heat capacity	kg per cubic metre and kJ per kilogram kelvin
Q _r and Q _d	Heat added before and across radiator	kW
T _a T _i T _o T _L	Ambient inlet outlet and imposed limit	degrees Celsius
H	Flow heat capacity times inlet temperature headroom	kW
rho ₀ P R _g	Reference density ambient pressure and dry air gas constant	kg per cubic metre Pa and J per kilogram kelvin

Declarations

Competing Interests

The author is employed at High Performance Industry (HPI). This employment is disclosed as relevant professional context. No company or client operational records, proprietary equipment data or financial data were used. The author has declared no other competing interests.

Author Contribution

M.A.A.M. supplied the industrial context and research direction. Substantial OpenAI ChatGPT assistance covered model formulation, software, formal calculations, verification scripts, visualization, original drafting and translation, as disclosed in the manuscript. M.A.A.M. is responsible for scientific review, interpretation and the final manuscript.

Data Availability

All hypothetical inputs, computational scripts, numerical outputs and verification records are supplied in Online Resource 1. No field measurements were collected.

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Figures

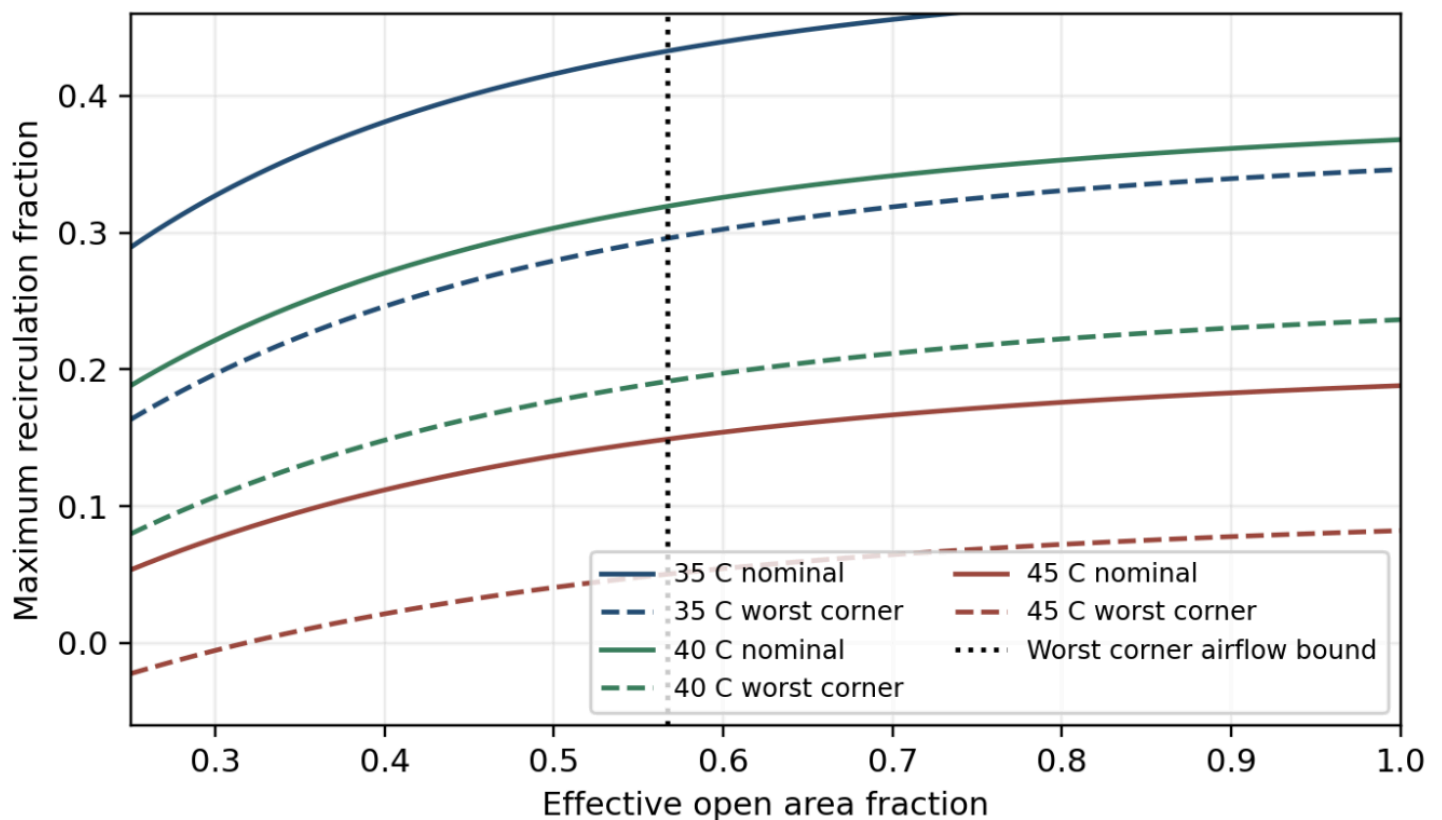


Figure 1

Maximum thermally permitted recirculation versus effective opening fraction for three assumed ambient temperatures. Solid curves are nominal and dashed curves are conservative. The dotted vertical line is the worst corner airflow requirement. Portions below zero admit no nonnegative recirculation

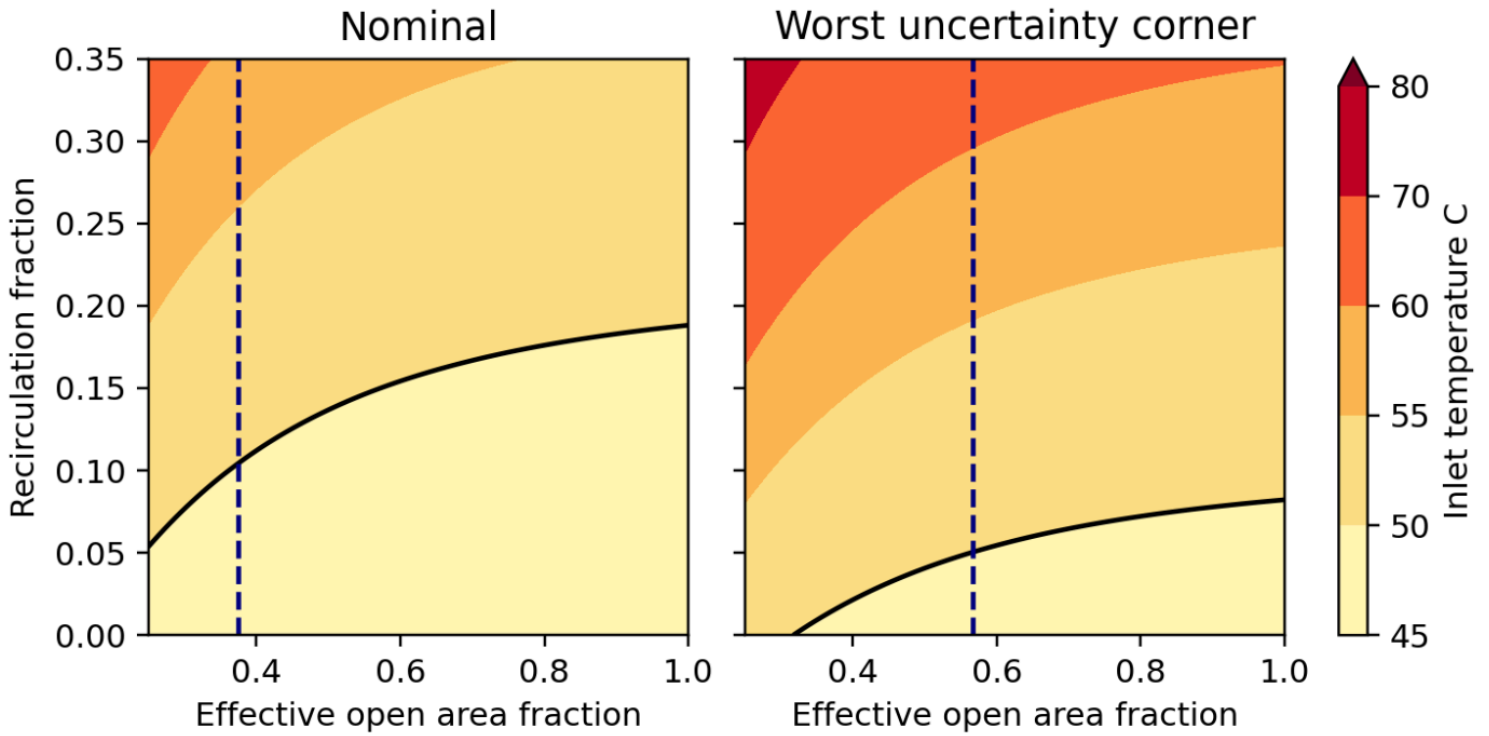


Figure 2

Inlet temperature contours at 45 degrees ambient temperature for nominal and worst corner inputs. The black contour is the assumed 50 degree limit. The dashed blue line is the separate minimum airflow opening. Acceptance requires a point to be right of the blue line and below the black contour

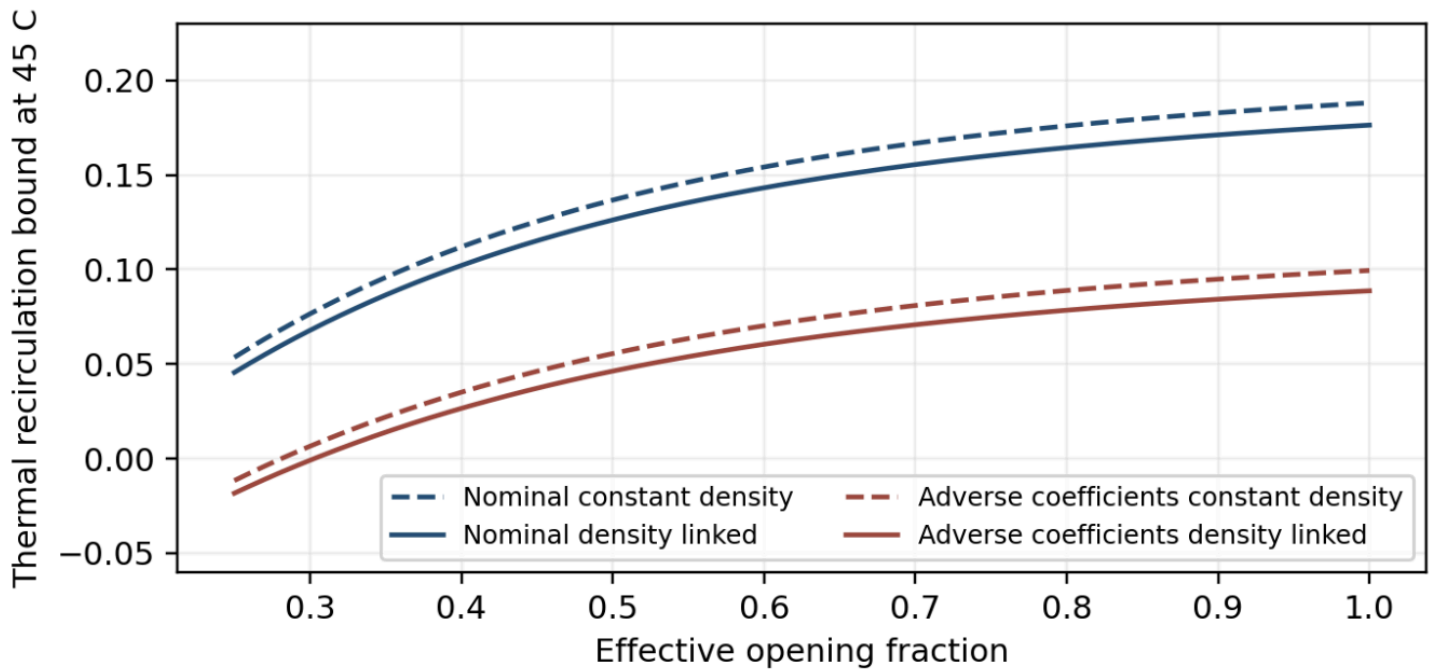


Figure 3

Thermal recirculation boundaries for constant density and temperature linked density at fixed fan speed. Reference density is 1.15 kg per cubic metre. The plot shows thermal boundaries only; acceptable points must also meet the airflow limit

Supplementary Files

This is a list of supplementary files associated with this preprint. Click to download.

- [ESM1.zip](#)