

Forced-convection channel sizing: a thermal–hydraulic screening decision

Aero engineering pipeline

Analytical verification study; not a peer-reviewed publication

Abstract

Three circular-passage layouts are compared using an energy balance, laminar friction and a fully developed uniform-heat-flux relation. The design question is which tested layout meets an inner-wall temperature limit and a straight-channel pressure budget with explicit adverse scenarios. Layout B (2 x 2 mm) has the lowest straight-passage pumping power among candidates passing the stated screening gates. The result is a conditional preliminary screen, not a thermal CFD result or hardware certification.

1 Requirements and problem definition

Quantity	Value
Total heat into coolant	100 W
Total volume flow	1 mL/s
Coolant inlet temperature	25 °C
Wetted inner-wall limit	72 °C
Straight-passage pressure budget	2000 Pa
Heated length	500 mm
Available channel-array width	12 mm

The interface asks for missing heated length or available width before the user can authorize a report. A calculation is invalidated when an input changes. The report preserves the exact inputs, formulas and result identity.

The fixed candidates are A: four 1 mm passages; B: two 2 mm passages; C: one 4 mm passage. They have equal total wetted perimeter, but different velocity, heat transfer coefficient and resistance. A 1 mm ligament separates adjacent passages; the required spans are 7, 5 and 4 mm, excluding outer casing. These are geometric options, not a tolerance-checked or manufactured part.

Property assumptions. Constant water properties are specified as $\rho = 997 \text{ kg/m}^3$, $c_p = 4180 \text{ J/(kg K)}$, $\mu = 0.00089 \text{ Pa s}$, $k = 0.607 \text{ W/(m K)}$, representative near 25 °C. They are fixed benchmark inputs, not temperature-updated property data. Use a property service such as NIST [3] before application to other conditions.

2 Model and governing relations

For N equally fed circular passages of diameter D and heated length L :

$$\dot{m} = \rho \dot{V}, \quad T_{b,o} = T_{b,i} + \frac{Q}{\dot{m}c_p}, \quad (1)$$

$$U = \frac{4\dot{V}}{N\pi D^2}, \quad Re_D = \frac{\rho U D}{\mu}, \quad (2)$$

$$Nu_D = \frac{hD}{k} = \frac{48}{11} \simeq 4.364, \quad A_w = N\pi DL, \quad (3)$$

$$T_{w,o} = T_{b,o} + \frac{Q}{hA_w}, \quad \Delta p = \frac{128\mu L \dot{V}}{N\pi D^4}, \quad (4)$$

$$P_{hyd} = \Delta p \dot{V}, \quad f_D = \frac{64}{Re_D}. \quad (5)$$

The heat-transfer relation assumes constant axial and circumferential heat flux, fully developed laminar convection [1]. Friction is independently cross-checked with $f_D(L/D)\rho U^2/2$ [2]. P_{hyd} is fluid power, not electrical pump power. Equal heat and flow sharing are imposed; a real manifold must establish them.

The inner-wall estimate increases monotonically along the heated length. Fully developed Nu is used as an asymptotic film-resistance screen, not as a resolved entrance-region temperature field. We require the outlet to exceed the estimated thermal entry length $L_t \simeq 0.05 Re_D Pr D$, with $Pr = c_p \mu / k$. The corresponding hydrodynamic estimate is $0.05 Re_D D$. Even when the outlet passes this check, entry pressure losses are omitted.

2.1 Adverse scenarios and selection rule

The thermal scenario uses 10% more heat, 10% less total flow and 20% lower h :

$$T_{w,g} = T_{b,i} + \frac{1.1Q}{0.9\dot{m}c_p} + \frac{1.1Q}{0.8hA_w}.$$

The separate pressure scenario uses 20% higher viscosity, 10% higher flow and 2% smaller diameter: $\Delta p_g = \Delta p(1.2)(1.1)/(0.98)^4$. These are adverse scenarios, not simultaneous observations or statistical bounds. They do not cover omitted physics or all water-property variation.

Each layout must fit the width, remain below an adverse $Re = 1800$, have $1.1L_t/0.8 \leq L$, remain below the 90 °C single-phase screening ceiling, and meet both guard budgets. The ceiling is not a boiling model; pressure, orientation and mixed convection require review. The selected layout minimizes nominal fluid pumping power among the passing candidates only. No global optimum or external solid-wall temperature is established.

3 Thermal–hydraulic decision

Layout B (2 x 2 mm) has the lowest straight-passage pumping power among candidates passing the stated screening gates.

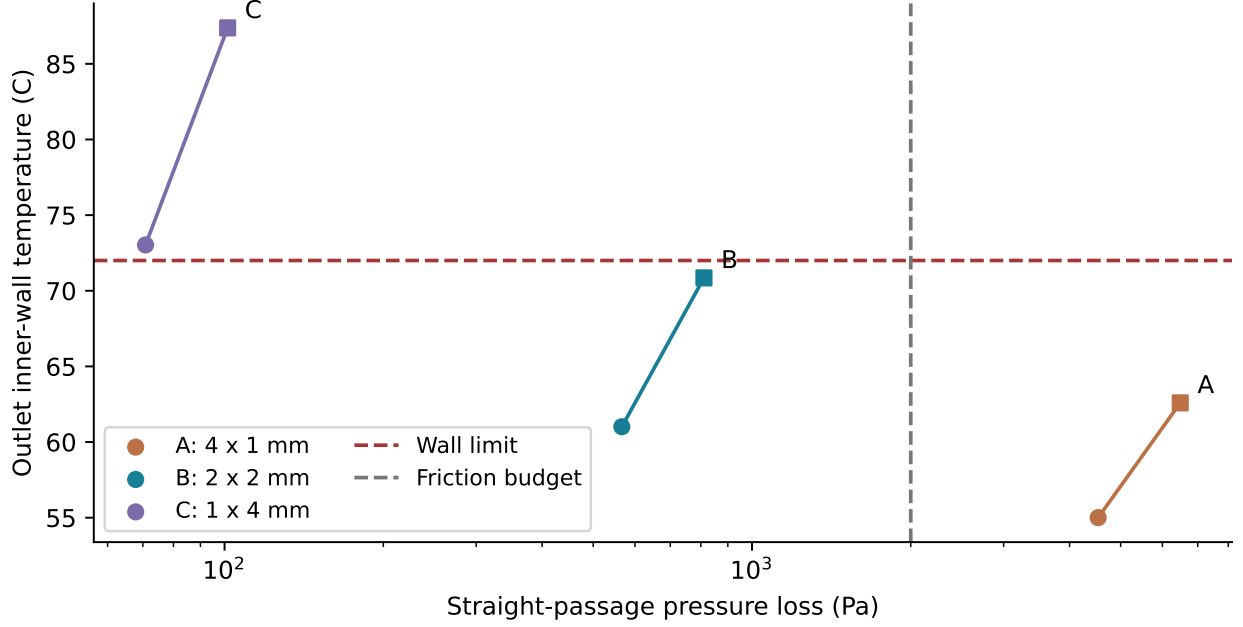


Figure 1. Nominal circles and adverse-scenario squares. The region below both budget lines is feasible only if the other gates also pass. Pressure and thermal guard points combine separate scenarios.

Layout	Wall C	Guard C	Friction Pa	Guard Pa	Gates
A	55.00	62.59	4532.7	6486.8	FAIL
B	61.01	70.85	566.6	810.8	PASS
C	73.03	87.38	70.8	101.4	FAIL

The nominal bulk-fluid rise is 23.995 K for every layout because total heat, total mass flow and heat capacity are unchanged. Channel count changes film resistance, not this energy balance. A lower pressure loss alone is therefore not enough to choose a cooling layout.

4 Temperature development and geometry

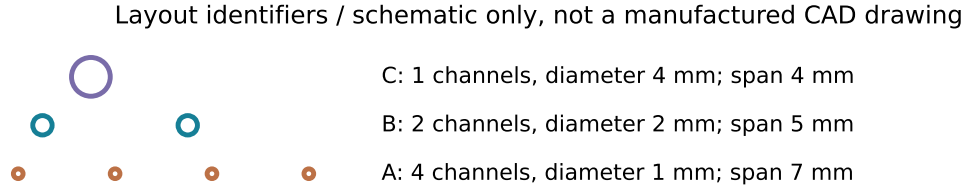
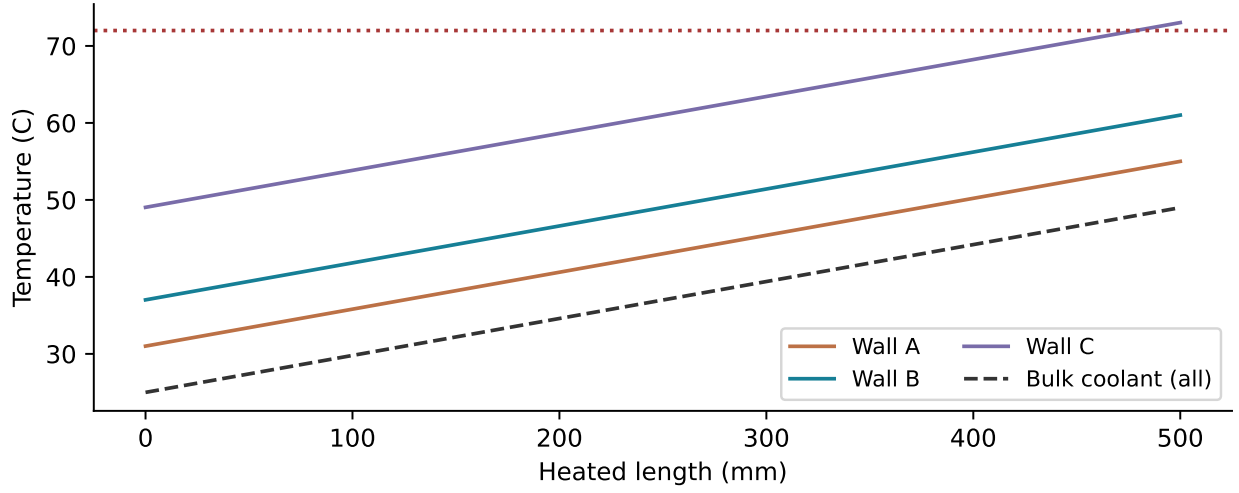


Figure 2. Analytical bulk and asymptotic inner-wall profiles and layout identifiers. These are computed one-dimensional estimates, not CFD contours, mesh images or experimental measurements.

Layout	Re	h W/m ² K	L_t mm	Guard K margin	Guard Pa margin
A	356.6	2648.7	109.3	9.41	-4486.8
B	356.6	1324.4	218.5	1.15	1189.2
C	356.6	662.2	437.1	-15.38	1898.6

Positive budget margins alone do not override regime or packaging gates. Failed gates remain visible in the evidence JSON, including the no-feasible-layout case.

5 Verification, not physical validation

An independent radial finite-volume energy integration checks the Nusselt coefficient. With $s = r/R$ and $t = k(T - T_w)/(q''R)$:

$$\frac{1}{s} \frac{d}{ds} \left(s \frac{dt}{ds} \right) = 4(1 - s^2), \quad t'(0) = 0, \quad t'(1) = 1, \quad t(1) = 0.$$

Face-integrated fluxes and midpoint velocity-weighted bulk temperatures yield $Nu = -2/t_b$. The analytic solution is $t = s^2 - s^4/4 - 3/4$ and $t_b = -11/24$. This verification does not call the production Nu constant to compute its answer.

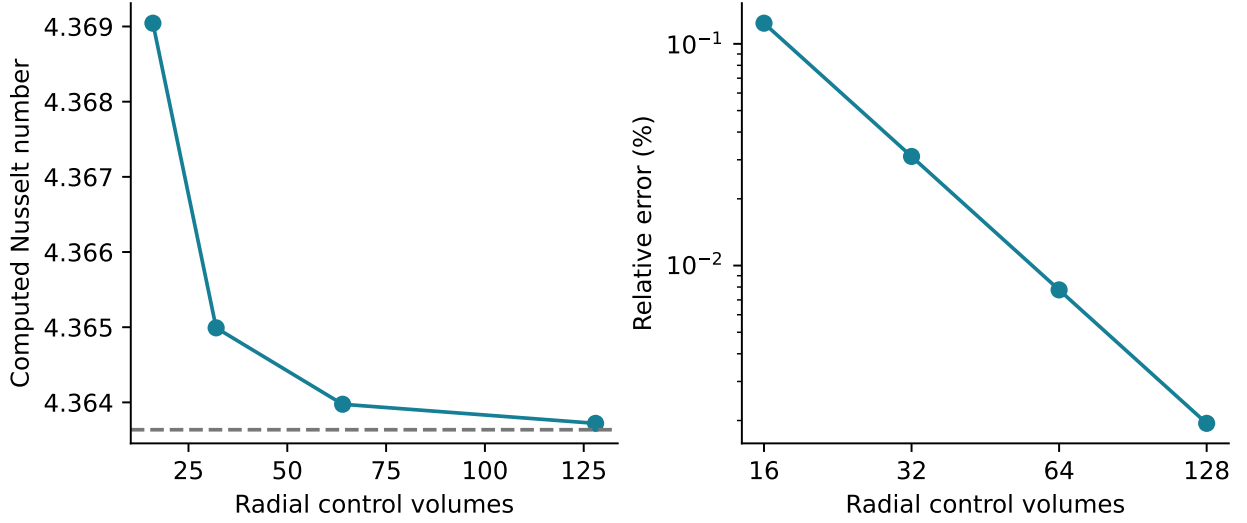


Figure 3. Radial energy discretization approaching 48/11. This is not an OpenFOAM grid-convergence study.

Radial cells	Computed Nu	Error %
16	4.3690439	0.12392
32	4.3649913	0.03105
64	4.3639753	0.00777
128	4.3637211	0.00194

The energy-balance relative closure error is 0. The two algebraically equivalent pressure formulations agree to floating-point precision. These checks detect implementation mistakes; they do not provide independent experimental validation of the assumptions or empirical entrance-length criterion.

6 Sensitivity and next fidelity level

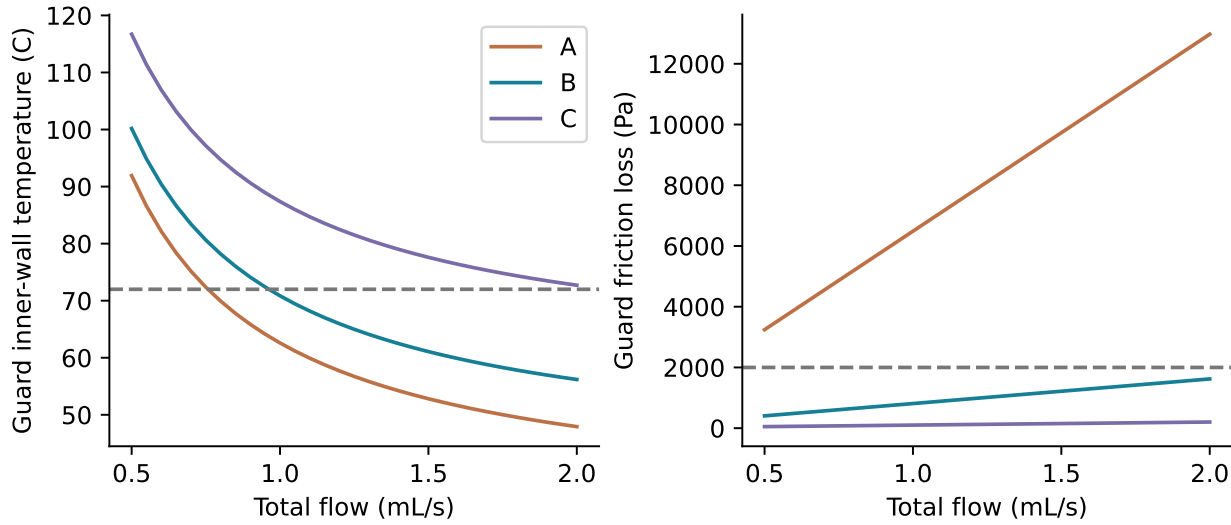


Figure 4. Separate adverse thermal and pressure scenarios versus total flow. Curves outside regime, width or entry-length gates must not be interpreted as valid designs.

Increasing flow lowers the bulk-fluid rise but raises friction loss. A real follow-on should import manufactured geometry, include manifolds, review flow sharing and property variation, and use conjugate heat-transfer CFD for solid conduction, local hot spots, contact resistance and the heated footprint. It needs a thermal mesh study, energy and mass conservation, and a matched thermal reference or measurement before design readiness is claimed. No thermal CFD was performed in this study.

Run identity: 1212c96216c4f6eda0468997a14fc7d146ed39cf1a0aeb53

Calculator source SHA-256:

f51913b75b2837c2e324a5d1823d5f22ef1d915e8bdd0292caa17499436ba7b5

The hashed evidence bundle retains inputs, all gate results, verification records, calculator source, this PDF, LaTeX source and figures. Live runs are retained for 48 hours; download the bundle to keep it.

References

- [1] NPTEL, Internal forced convection, uniform-flux circular tubes and entrance regions.
<https://archive.nptel.ac.in/content/storage2/courses/112108149/pdf/M6/M6TeacherSlides.pdf>
- [2] R. Fitzpatrick, Poiseuille Flow.
<https://farside.ph.utexas.edu/teaching/336L/Fluidhtml/node136.html>
- [3] NIST, Chemistry WebBook, Thermophysical Properties of Fluid Systems.
<https://webbook.nist.gov/chemistry/fluid/>

Browser evidence 1

Actual browser capture from the same run. Interface evidence is distinct from numerical verification.

AERO

Cooling-channel study

Capability papers

Flow study

Load worked example

01 Requirements

100% supplied

02 Math

Ready to inspect

03 Layouts

Ready to inspect

04 Compare

Ready to inspect

05 Report

Ready to inspect

ENGINEERING PARTNER

Which layout keeps the wall cool?

gemma3:12b · live conversation

What specific missing geometry do you need before this cooling-channel comparison, and why? Do not assume dimensions.

AERO - LIVE MODEL

To proceed with the analytical screening, I need the heated length of the passages. This value directly impacts the surface area available for heat transfer and the frictional losses due to flow. Without it, I cannot estimate the heat transfer coefficient or pressure drop. The available width is also needed to assess the overall packaging constraints. You can edit these values directly in the user controls; I cannot modify them myself.

Once you provide the heated length, I can begin to estimate the heat transfer and pressure drop, keeping in mind the fully developed laminar flow and uniform heat flux assumptions.

REQUIREMENTS CHECK · NOT A MODEL REPLY

Worked example loaded: 100 W into water at 1 mL/s, 25 °C inlet, 72 °C inner-wall limit and 2,000 Pa straight-channel budget. Length 500 mm; available width 12 mm. Review these assumed requirements before calculating.

Dimensions supplied. Review the assumptions before starting.

Discuss the requirements

What geometry do you need before we can compare layouts?

Ask Aero ↕

Live chat explains. Only your reviewed numeric inputs start a calculation. Chat cannot change them or run a solver.

ANALYTICAL FIRST CUT · NOT THERMAL CFD

Define heat, flow and limits.

Inputs changed. The previous result no longer applies; review and run again.

Heat into coolant (W)

100

Total water flow (mL/s)

1

Inlet temperature (°C)

25

Inner-wall limit (°C)

72

Friction budget (Pa)

2000

Heated length (mm)

500

Available array width (mm)

12

► Model assumptions and what "wall" means

☒ I reviewed these inputs and the model assumptions.

Prepare comparison →

← Back

Review each stage before execution

Math →

Reviewed requirements and missing-input guidance.

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Browser evidence 2

Actual browser capture from the same run. Interface evidence is distinct from numerical verification.

AERO Cooling-channel study

Capability papersFlow studyLoad worked example

01 Requirements100% supplied

02 MathReady to inspect

03 LayoutsReady to inspect

04 CompareReady to inspect

05 ReportReady to inspect

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REQUIREMENTS CHECK · NOT A MODEL REPLY

Worked example loaded: 100 W into water at 1 mL/s, 25 °C inlet, 72 °C inner-wall limit and 2,000 Pa straight-channel budget. Length 500 mm; available width 12 mm. Review these assumed requirements before calculating.

Dimensions supplied. Review the assumptions before starting.

Discuss the requirements

What geometry do you need before we can compare layouts?

Ask Aero ↗

Live chat explains. Only your reviewed numeric inputs start a calculation. Chat cannot change them or run a solver.

ANALYTICAL FIRST CUT · NOT THERMAL CFD

Inspect the three candidate layouts.

Inputs changed. The previous result no longer applies; review and run again.

Inspect layout

B · 2 × 2 mm

2 passages · Ø 2 mm · 500 mm long



Parametric fluid passages. Drag to rotate; scroll to zoom. Diameter and spacing exaggerated ×15 for inspection. No imported CAD or CFD mesh is claimed.

← Back

Review each stage before execution

Compare →

Interactive parametric circular-passage layouts; no imported or verified manufacturing CAD.

Browser evidence 3

Actual browser capture from the same run. Interface evidence is distinct from numerical verification.

01 Requirements

Complete

02 Math

Complete

03 Layouts

Complete

04 Compare

Complete

05 Report

Complete

AERO

Cooling-channel study

Capability papers

Flow study

Load worked example

ENGINEERING PARTNER

Which layout keeps the wall cool?

gemma3:12b · live conversation

replace them. Ask the live model for help below.

YOU

What specific missing geometry do you need before this cooling-channel comparison, and why? Do not assume dimensions.

AERO · LIVE MODEL

To proceed with the analytical screening, I need the heated length of the passages. This value directly impacts the surface area available for heat transfer and the frictional losses due to flow. Without it, I cannot estimate the heat transfer coefficient or pressure drop. The available width is also needed to assess the overall packaging constraints. You can edit these values directly in the user controls; I cannot modify them myself.

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REQUIREMENTS CHECK · NOT A MODEL REPLY

Worked example loaded: 100 W into water at 1 mL/s, 25 °C inlet, 72 °C inner-wall limit and 2,000 Pa straight-channel budget. Length 500 mm; available width 12 mm. Review these assumed requirements before calculating.

Dimensions supplied. Review the assumptions before starting.

Discuss the requirements

What geometry do you need before we can compare layouts?

Ask Aero I

ANALYTICAL FIRST CUT · NOT THERMAL CFD

Make the decision with evidence.

Start with your requirements or load the worked example.

complete

Layout B (2 x 2 mm) has the lowest straight-passage pumping power among candidates passing the stated screening gates.

Layout B (2 x 2 mm) has the lowest straight-passage pumping power among candidates passing the stated screening gates.

Layout	Nominal wall °C	Guard wall °C	Guard Δp Pa	Disposition
A	55	62.59	6486.79	friction budget
B	61.01	70.85	810.85	Pass
C	73.03	87.38	101.36	outlet developed, wall budget

Layout A

62.59 °C

Layout B

70.85 °C

Layout C

87.38 °C

Guard wall estimates above. Energy and radial-equation verification: passed. Thermal CFD: not performed.

PDF paper ^

LaTeX + evidence ↓

► What remains before thermal CFD or a hardware decision?

This is a reproducible analytical screen. It does not establish a safe hardware operating point or validated thermal CFD.

Completed analytical comparison and paper download.

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