

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	100.0 W
Total volume flow	1.0 mL/s
Coolant inlet temperature	25.0 °C
Wetted inner-wall limit	72.0 °C
Straight-passage pressure budget	2000.0 Pa
Heated length	500.0 mm
Available channel-array width	12.0 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.

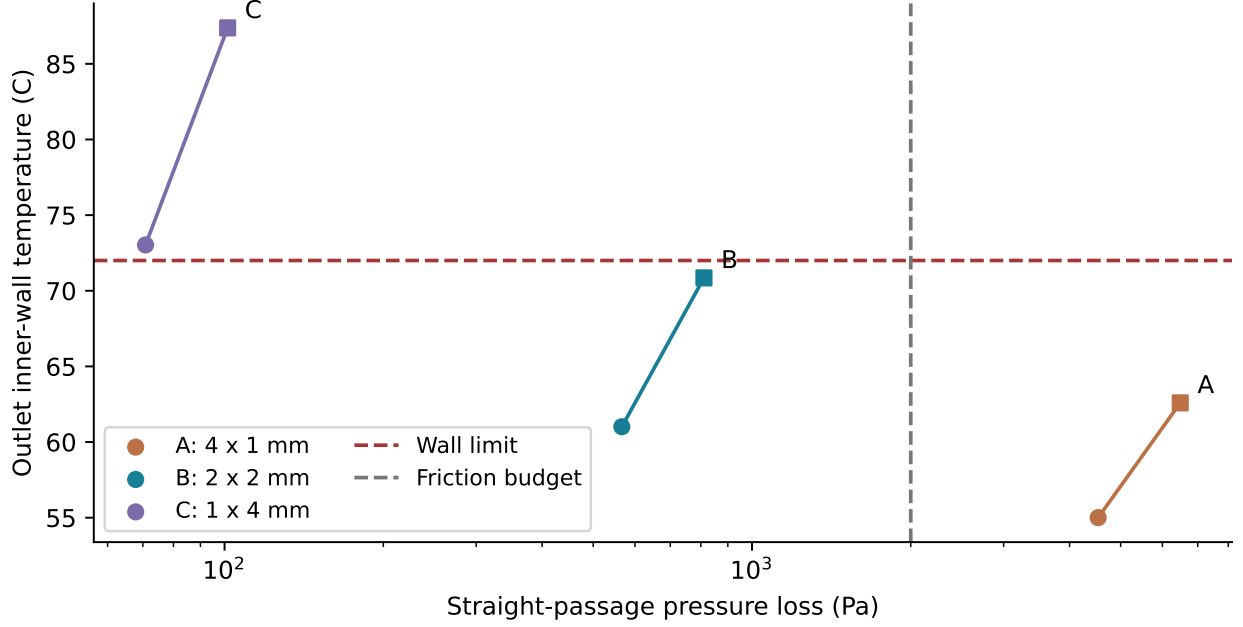


Figure 1. Nominal circles and adverse-scenario squares. The region below both budget lines is feasible only if the other gates also pass. Pressure and thermal guard points combine separate scenarios.

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

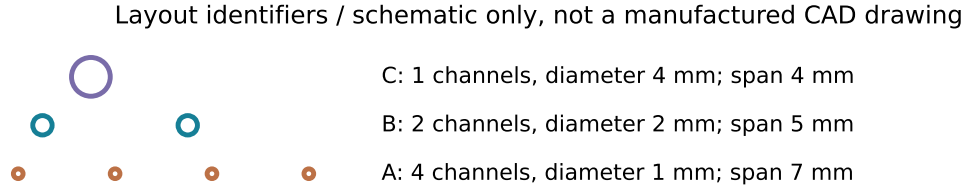
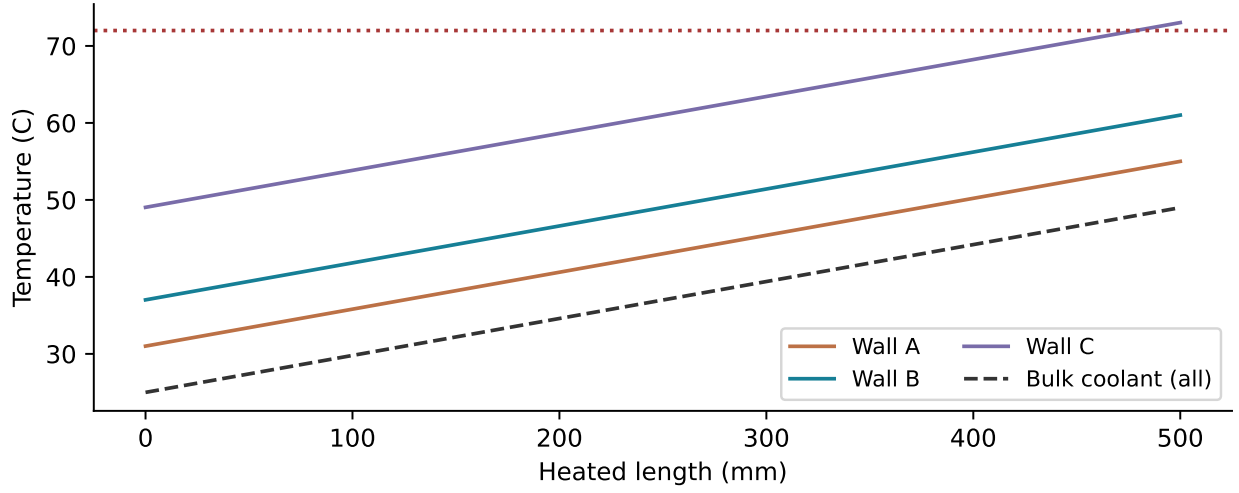


Figure 2. Analytical bulk and asymptotic inner-wall profiles and layout identifiers. These are computed one-dimensional estimates, not CFD contours, mesh images or experimental measurements.

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.

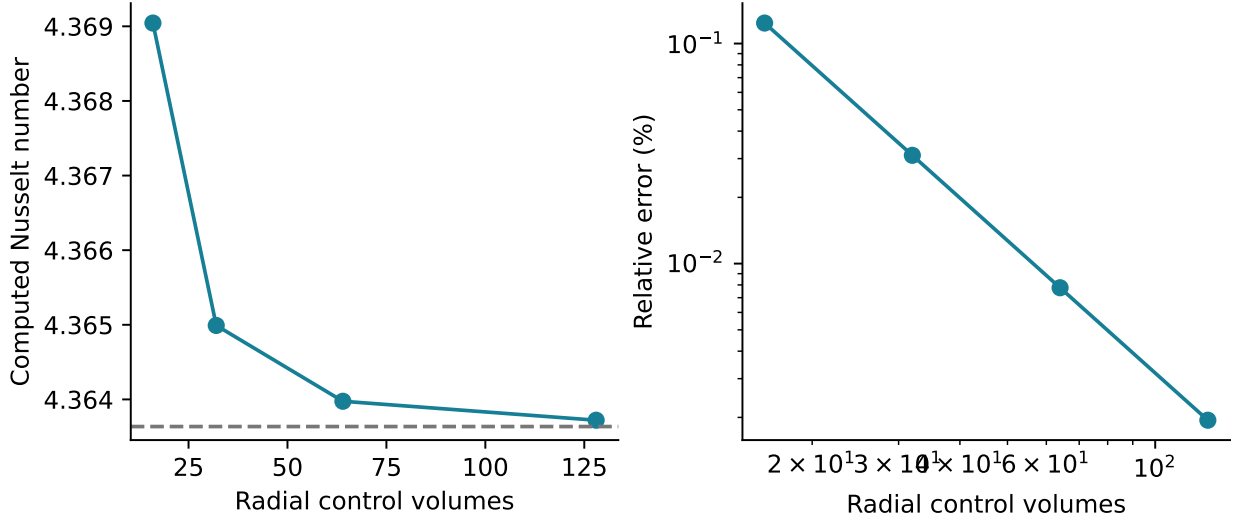


Figure 3. Radial energy discretization approaching 48/11. This is not an OpenFOAM grid-convergence study.

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

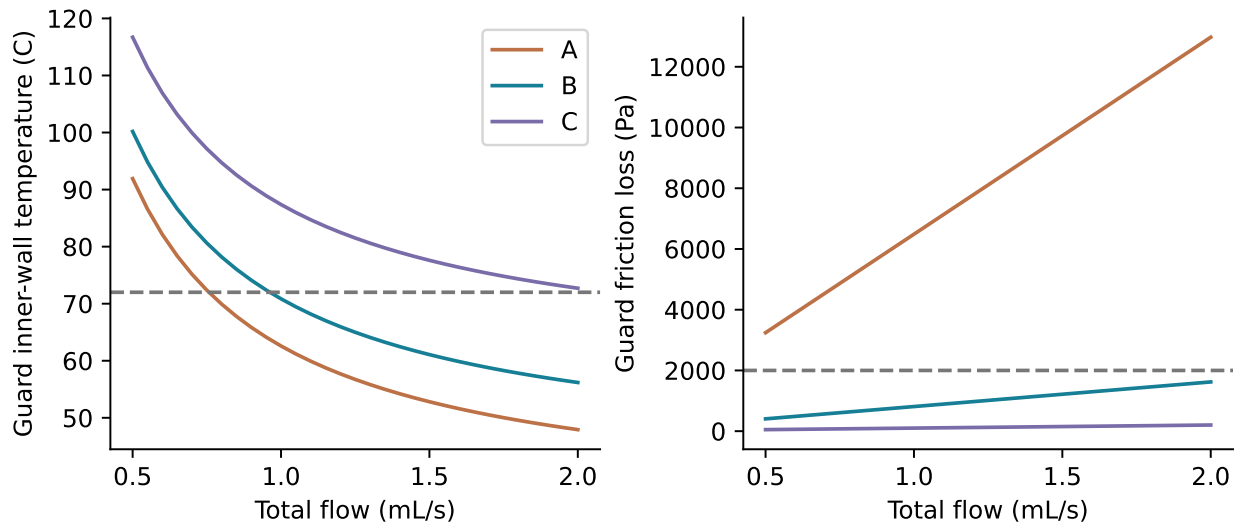


Figure 4. Separate adverse thermal and pressure scenarios versus total flow. Curves outside regime, width or entry-length gates must not be interpreted as valid designs.

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.

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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/>