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Validation-Led CFD Assessment and Development of the MCR-S2N/S2N+ Airbox Package

Master project

ACFB7003

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Confirmation of Own Work

I confirm that this dissertation is the result of my own work. All sources used have been acknowledged through appropriate academic references. Any figures, data, CAD geometry, CFD outputs, company information or test results produced by others are clearly identified and cited.

Signed:  _____

Date: 25-06-2026.

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Abstract

This dissertation evaluates the internal-flow performance of the MCR-S2N/S2N+ airbox package using steady Reynolds-averaged Navier-Stokes (RANS) CFD supported by experimental testing. The numerical method was assessed against a pressure-tapped 90-degree bend over 5-25 inH₂O, including volume-mesh, near-wall and turbulence-model sensitivity studies. The retained M3/L8/SST method reproduced the measured wall-pressure pattern with correlations of 0.897-0.943 and normalised RMSE values of 10.8-14.4%, supporting its use for comparative geometry assessment rather than exact local-pressure prediction. Pressure-tap testing of the updated airbox provided an additional spatial-pressure comparison and showed that CFD reproduced the regional pressure pattern but overpredicted suction magnitude.

The largest engineering improvement was obtained by the existing S2N+ updated airbox. Under common internal boundary conditions, D0 increased mean mass flow by 3.72% and reduced mean total-pressure loss by 37.2% relative to the original airbox, with all four separately simulated cylinder paths improving. D0 was therefore retained as the baseline for a D1-D15 design study covering wall profile, inlet shape and size, transition length, a central divider and trumpet-tray geometry. D9, with a 50 mm shorter lipless transition, gave the best individual design result: mean total-pressure loss was 4.10% lower than the matched D6 reference and all four paths improved. This gain is close to the 4.48% model-preparation sensitivity and is therefore not treated as a confirmed improvement.

At the +728.92 Pa inlet-pressure sensitivity condition, D9 retained a small mean advantage over D0 but introduced a C2 penalty. The study therefore identifies D9 as the preferred prototype for experimental comparison rather than a final validated design. Further work should manufacture and flow-bench D9 against D0, complete a matched airbox mesh-refinement check, and then assess the design on the vehicle under representative running conditions.

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Abbreviations and Symbols

Table 1. Abbreviations, identifiers and symbols used in the dissertation.

Abbreviation / symbol	Definition	Unit
Abbreviations		
BOI	Body of influence	—
CAD	Computer-aided design	—
CFD	Computational fluid dynamics	—
CV	Coefficient of variation	—
MAE	Mean absolute error	—
RANS	Reynolds-averaged Navier–Stokes	—
RMSE	Root mean square error	—
RNG	Renormalisation-group turbulence-model formulation	—
SD	Standard deviation	—
SST	Shear-stress transport	—
S2N / S2N+	Original and updated MCR Sports 2000 race car model	—
Project-specific identifiers		
C1–C4	Cylinder-path cases 1–4	—
D0–D15	D0 baseline and D1–D15 design concepts	—
M0–M4	Pipe-bend volume-mesh levels	—
L0, L4, L8, L12	Near-wall sensitivity cases with 0, 4, 8 and 12 prism layers	—
Tap 1–16	Pressure-tap identifiers	—

Abbreviation / symbol	Definition	Unit
Symbols		
C_D / C_L	Drag and lift coefficients	—
C_{pt}	External intake total-pressure coefficient, $(p_{t,intake} - p_{t,freestream}) / q^\infty$	—
K_t	Normalised internal total-pressure loss, $\Delta p_t / 4982 \text{ Pa}$	—
$\dot{m}$	Mass-flow rate	kg s^{-1}
p_s	Static pressure	Pa
p_t	Total pressure	Pa
Δp_t	Total-pressure loss	Pa
q	Dynamic pressure, $0.5\rho V^2$	Pa
q_∞	Freestream dynamic pressure	Pa
r	Pearson correlation coefficient	—
ρ	Air density	kg m^{-3}
V	Velocity magnitude	m s^{-1}
y^+	Dimensionless wall distance	—

1. Introduction

1.1 Background

The airbox of a naturally aspirated competition vehicle performs a more complex role than simply connecting an external intake opening to the engine. It must preserve as much recoverable pressure as possible, avoid unnecessary separation and recirculation, and distribute the incoming air consistently between the trumpet or runner entries. These requirements are important in a race-engine installation because relatively small changes in pressure loss or cylinder-to-cylinder air supply can affect volumetric efficiency, repeatability and the usefulness of subsequent engine calibration.

The present project focuses on the MCR-S2N/S2N+ airbox package, with particular attention to the S2N vehicle fitted with the updated airbox geometry. The downstream runner and trumpet geometry is retained throughout the principal comparisons. This controlled approach is intended to isolate the effect of the airbox body and inlet condition rather than mixing those effects with changes in runner length, bellmouth radius or downstream flow area.

The project is primarily an internal-flow investigation. A controlled pipe or bend test on the flow bench is used to evaluate the CFD procedure before that procedure is applied to the more complex airbox. A limited external CFD assessment is retained to establish the pressure and flow direction at the airbox mouth and to test whether the use of an idealised uniform inlet materially changes the internal predictions.

1.2 Problem statement

CFD can produce detailed pressure and velocity fields, but visually plausible contours do not demonstrate that a model is accurate. Internal-flow predictions may be affected by the selected turbulence model, near-wall treatment, mesh resolution, boundary conditions and the manner in which pressure loss is extracted. Applying an unvalidated method directly to the MCR airbox would therefore create uncertainty over whether the predicted differences between configurations are physical or numerical.

A second uncertainty concerns the airbox inlet boundary condition. An isolated internal model is commonly supplied with a uniform pressure or velocity profile, whereas the real intake may receive a non-uniform pressure field or a small flow-angle offset caused by surrounding bodywork. However, a complete whole-car aerodynamic development programme would be disproportionate to the main internal-flow research question. The external assessment is therefore restricted to determining whether applying an external-CFD-derived inlet pressure as a sensitivity condition produces a meaningful change relative to the idealised internal boundary condition.

The study uses a staged validation and comparison process. A controlled flow-bench bend test is first reproduced numerically to assess the steady RANS method. Limited whole-vehicle CFD is then used to estimate the pressure at the old and updated airbox openings and define an inlet-pressure sensitivity value. The same internal CFD method is applied to the old and updated airboxes using four separate one-cylinder-at-a-time cases under common boundaries. Finally, the updated airbox is pressure tapped and tested at the nominal 20 inH₂O bench condition to provide an airbox-specific pressure-correlation dataset.

1.3 Aim and objectives

The aim of this dissertation is to establish a steady-state CFD method suitable for comparing internal intake flow, apply it to the old and updated MCR-S2N/S2N+ airboxes under matched one-cylinder-at-a-time demand, and use the results to develop and assess a constrained airbox modification. The work is supported by pipe-bend validation, a limited external-inlet sensitivity study and pressure-tap correlation on the updated airbox.

- Design and conduct a controlled flow-bench pipe or bend test using controlled pressure measurements.
- Replicate the pipe flow-bench test numerically and quantify agreement in the measured wall-static-pressure distribution.
- Prepare the relevant MCR airbox internal fluid volumes while retaining the same runner and trumpet geometry across controlled comparisons.
- Conduct a limited external vehicle CFD pilot to assess the pressure profile presented to the airbox inlet.
- Use the limited whole-vehicle CFD results to extract area-averaged airbox-inlet static and total pressure, then assess the effect of the updated-airbox vehicle-derived total pressure on the internal solution.
- Evaluate internal total-pressure loss, mass flow, velocity and pressure distribution for Cylinders 1-4 using separate steady cases.
- Compare the selected MCR airbox configurations under consistent solver, mesh and boundary-condition definitions.
- Conduct volume-mesh, prism-layer and turbulence-model sensitivity assessments for selection of the numerical method, and apply consistent meshing controls to the subsequent airbox comparisons.
- Pressure tap the updated airbox and compare the measured 16-tap wall-static-pressure distributions with CFD at the matched nominal bench condition.
- Develop and evaluate a packaging-feasible airbox modification informed by the validated CFD results, targeting reduced total-pressure loss and improved consistency between the four separately simulated cylinder flow paths while retaining the fixed runner and trumpet geometry.

1.4 Research questions

- How accurately does the selected steady RANS method reproduce the measured wall-static-pressure redistribution in the controlled 90-degree pipe test?
- Does the vehicle-derived inlet-pressure condition materially change the internal response or design ranking relative to the controlled uniform inlet condition?
- How do the old and updated airboxes differ in total-pressure loss, mass flow and local pressure distribution when Cylinders 1-4 are simulated separately with common boundary conditions?
- How closely does the CFD reproduce the 16-tap updated-airbox wall-static-pressure distribution measured at 20 inH₂O, 10 mm valve lift and fully open throttle?
- Are the main conclusions supported by mesh quality, monitor stability, mass balance and the repeatability of the available airbox bench measurements?
- Can a constrained airbox modification improve total-pressure preservation and cylinder-path consistency by more than the observed model-preparation sensitivity without creating an unacceptable packaging or manufacturing penalty?

1.5 Scope and boundaries

The project focuses on steady internal CFD of the MCR-S2N/S2N+ airbox package followed by a constrained design study. The pipe bend is used to assess the numerical method, while the external calculation is limited to an inlet-boundary sensitivity study. Old and updated airboxes are compared using separate C1-C4 cases, and the updated-airbox pressure-tap test provides an additional spatial-pressure comparison. The design stage retains the existing runner and trumpet geometry and evaluates D1-D15 across wall profile, inlet shape and size, transition length, a central divider and trumpet-tray extension. The final concept is selected from the controlled 0 Pa cases before the higher-inlet-pressure and later whole-vehicle checks.

Full engine-cycle transient CFD, valve-motion modelling, acoustic or pressure-wave tuning, complete yaw and ride-height maps, rotating-wheel development and whole-vehicle aerodynamic optimisation are outside the core scope. A cycle-resolved intake calculation would require credible time-dependent engine boundary conditions, valve-motion or coupled engine modelling, substantially greater computational effort and a separate validation strategy. Within the approximately three-month university project period, introducing these requirements would have displaced the experimental validation and controlled geometry-development work that form the principal contribution of this study. The CFD domains do not include the cylinder head, port or moving valve, so absolute flow-rate agreement with the assembled-engine bench test is not claimed unless the pressure at the numerical outlet plane can be matched experimentally. The airbox pressure comparison is therefore reported as correlation of the spatial wall-static-pressure response at the nominal bench setting. No air-filter element was installed for the airbox flow-bench tests or represented in the internal CFD. The airbox can accommodate a filter element, but the present study therefore compares the geometries in a deliberately unfiltered configuration.

2. Literature Review

2.1 MCR Sports 2000 platform and airbox context

MCR Race Cars is the platform Welsh-built sports prototype used in Sports 2000 competition (MCR Race Cars, n.d.). The S2N vehicle was jointly developed by MCR Race Cars and the University of Wales Trinity Saint David (UWTSD), whereas S2N+ was subsequently developed by UWTSD as a university-led evolution of the platform. Both programmes received technical input from John Iley in his role as Professor of Practice at UWTSD, and the updated S2N+ airbox assessed in this dissertation was designed by Iley. At the time of this study, the UWTSD student race team operates two S2N+ cars in Sports 2000 competition, giving the airbox investigation direct practical relevance. This development history places the study within a continuing university–industry programme rather than a comparison of unrelated airbox geometries.

Inspection of the project CAD shows compact bodywork and a tightly packaged engine-cover region. The location of the intake relative to the roll hoop, engine cover and local boundary layer can therefore influence the pressure and flow direction reaching the airbox. The engineering question is consequently not only whether the external opening occupies a favourable region, but also whether the internal airbox preserves and distributes the available pressure effectively.

The dissertation treats internal airbox performance as its principal subject. The external vehicle model supplies a representative inlet boundary condition, while the flow-bench test supports assessment of the numerical method. This separation prevents the study from becoming an uncontrolled combination of whole-vehicle aerodynamics, manifold tuning and engine-cycle simulation.

2.2 Function of the airbox in a naturally aspirated race engine

In a naturally aspirated race engine, the airbox is both an aerodynamic diffuser/plenum and part of an unsteady gas-dynamic system (Heywood, 2018). It should preserve total pressure, limit separation and recirculation, and avoid a geometry that systematically favours one trumpet, throttle-body and runner path. A directly relevant race-car study by De Vita et al. (2002) used three-dimensional CFD to develop a naturally aspirated prototype-sports-car airbox with the objective of reducing fluid-dynamic losses and improving volumetric efficiency, with subsequent track testing used to assess the vehicle-level benefit. Brusiani, Bianchi and Bianchi d'Espinosa (2010) analysed a high-performance race-car airbox using LES because unsteadiness can affect pressure recovery, whereas Brennan et al. (2006) coupled one-dimensional engine simulation with three-dimensional CFD and measured pressure histories to identify airbox resonance and cylinder-to-cylinder effects. These studies show that mean pressure preservation and path consistency are important, but also that steady CFD cannot represent every mechanism governing the operating engine.

A useful airbox assessment therefore requires numerical metrics as well as flow visualisation. Average loss can conceal one poor throttle-body/runner path, while a smooth velocity contour can conceal irreversible dissipation. The present study consequently reports total-pressure loss, the normalised loss coefficient (K_t), mass flow and the spread across the four separately simulated cylinder paths. Because only one throttle-body/runner outlet is active in each case, the resulting coefficient of variation describes path consistency under matched steady demand; it is not a prediction of simultaneous four-cylinder flow distribution.

This distinction controls the level of claim made later in the dissertation. The steady cases are intended to isolate geometric resistance and determine whether the plenum presents materially different resistance to the four throttle-body/runner paths. Cycle-resolved filling, acoustic tuning and interactions between cylinders require transient pressure boundaries or coupled one-dimensional/three-dimensional modelling and are treated as future work rather than inferred from the steady results (Brennan et al., 2006; Ling and Tun, 2006)

Table 2. Airbox functions, engineering meaning and corresponding dissertation metrics.

Airbox function	Engineering meaning	Dissertation metric
Pressure supply	The inlet should receive useful pressure rather than strongly accelerated, separated or shielded flow.	External inlet p_s , p_t , velocity and flow angle.
Pressure preservation	The airbox should avoid dissipating recoverable pressure through separation and excessive turning.	Internal total-pressure loss and normalised loss coefficient K_t .
Cylinder-path consistency	The four separately opened runner paths should present comparable resistance under matched demand.	Mean, range, standard deviation and CV across four separate cases; worst-case result.
Flow quality	The internal field should limit recirculation, reverse flow and high-loss regions.	Velocity vectors, streamlines and low-momentum/recirculating regions.
Controlled comparison	Changes should be attributable to the airbox rather than simultaneous runner changes.	Fixed runner and trumpet geometry.

2.3 Fundamental intake-flow theory

The principal fluid-mechanics quantities are static pressure, dynamic pressure, total pressure and total-pressure loss. Static pressure is the local thermodynamic pressure, while dynamic pressure represents the kinetic-energy contribution and, for low-speed incompressible flow, is $q = 0.5 \rho V^2$. Total pressure approximates the recoverable mechanical energy of the stream and may be written $p_t = p_s + q$ under the low-Mach-number conditions of this study (Anderson, 2016; White, 2011).

Total pressure is more suitable than static pressure alone for comparing intake restriction because local deceleration can raise static pressure even after mechanical energy has been dissipated through viscous effects, turbulent mixing or separation (Anderson, 2016; White, 2011). The primary restriction metric is therefore $\Delta p_t = \text{mean}(p_{t,\text{in}}) - \text{mean}(p_{t,\text{out}})$. Since the inlet total gauge pressure is defined as 0 Pa in the internal model, the ratio $p_{t,\text{out}}/p_{t,\text{in}}$ is undefined and is not used. Instead, the dimensionless normalised loss is $K_t = \Delta p_t / 4982 \text{ Pa}$, where 4982 Pa is the selected reference depression used to normalise all internal airbox CFD cases.

Continuity requires the signed inlet and outlet mass flows of each converged steady case to balance (White, 2011), but it does not imply that four separately opened paths will have identical resistance. The mean, range, sample standard deviation and coefficient of variation of the four single-outlet mass-flow or loss results provide a dimensionless comparison of cylinder-path consistency. This statistic must be interpreted together with the worst case because an improved mean can still conceal deterioration of one path (ASME, 2018).

Table 3. Principal fluid-mechanics quantities and their use in the dissertation.

Quantity	Definition or expression	Use in this dissertation
Static pressure, p_s	Local thermodynamic pressure.	External pressure field and local internal pressure distribution.
Dynamic pressure, q	$0.5 \rho V^2$.	Theoretical pressure available from vehicle or bench flow speed.
Total pressure, p_t	Recoverable pressure, approximately $p_s + q$ at low Mach number.	Primary pressure-preservation quantity.
Total-pressure loss	Area-averaged $p_{t,\text{in}} - p_{t,\text{out}}$.	Primary internal restriction metric.
Normalised total-pressure loss	$K_t = \Delta p_t / 4982 \text{ Pa}$.	Dimensionless comparison using the selected 4982 Pa CFD reference depression; avoids an invalid gauge-pressure ratio.
Coefficient of variation	Sample standard deviation divided by the four-case mean.	Consistency of the four separate steady cylinder-path results.
Flow angle	Angle between the local velocity vector and the intended inlet axis.	External inlet alignment and boundary-condition sensitivity.

2.4 Airbox pressure loss and cylinder-path consistency

Airbox total-pressure loss is influenced by inlet-jet direction, expansion rate, turning radius, wall proximity and the clearance around the trumpet entries. A rapid expansion can form a separated shear layer; an inlet jet that impinges on a wall can create a stagnation region followed by mixing; and a low-momentum recirculation zone can obstruct one runner path. These mechanisms can simultaneously produce local static-pressure recovery and total-pressure loss, so static-pressure contours alone are insufficient for design selection (Anderson, 2016; Versteeg and Malalasekera, 2007).

A configuration with low mean loss is not automatically superior if one cylinder path deteriorates. Muckova et al. (2023) used transient CFD with time-shifted cylinder outlet pressures to identify uneven filling, screened four airbox variants and manufactured the preferred concept. Their subsequent flow-bench test opened each outlet sequentially while blanking the other three, and the original and modified airboxes showed the same broad comparative outcome. This supports sequential bench and CFD cases as evidence of path resistance, but the authors also state that stationary bench flow cannot reproduce the transient engine calculation exactly.

Montenegro et al. (2013) used a multilevel strategy ranging from simplified screening models to coupled 1D/3D analysis and experimental confirmation. Claywell and Horkheimer (2006) similarly combined 1D and coupled 1D/3D analysis for Formula SAE restrictor development, while Sathishkumar et al. (2024) varied convergent and divergent angles in CFD to reduce restrictor pressure loss. These component studies support targeted screening of transition geometry but do not resolve plenum-to-runner path balance. The present project therefore limits concept generation to modifications that address an observed mechanism. All four separate cylinder-path cases are used for each quantitative concept decision, while limiting or representative paths are used for mechanism-focused flow visualisation. This prevents a favourable response on one path from hiding deterioration elsewhere.

2.5 Bellmouths, runner entries and controlled downstream geometry

Bellmouth and trumpet-entry geometry can materially affect local contraction loss. Blair and Cahoon (2006) show that inlet profile and radius influence discharge behaviour: a sharp-edged entry encourages vena-contracta formation and separation, whereas a suitably shaped bellmouth allows smoother acceleration into the runner. Burnside (2025) also identified entry shape and manufacturing surface condition as relevant variables in UWTSD flow-bench work.

Those effects are relevant but are not the independent variable of the present study. Changing the trumpets at the same time as the airbox would make it impossible to determine whether an observed improvement came from the plenum or from the entry radius. The downstream runner and trumpet geometry is therefore held constant in the controlled airbox comparison. Any later trumpet optimisation should be treated as a separate study.

2.6 Experimental validation of internal-flow CFD

A numerical model should be validated using measured quantities that can be reproduced directly at matching locations and operating conditions. ASME V&V 20 distinguishes numerical verification from validation against experimental data and emphasises comparison for a specified response quantity at a specified validation point (ASME, 2009). Oberkampf and Trucano (2002) similarly distinguish solution verification, which assesses numerical error, from validation, which assesses agreement with physical reality for the quantities and conditions of interest. A canonical pipe-bend test is useful because the geometry, boundary conditions and pressure-tap locations are more controllable than those of the complete airbox. The present bend test therefore assesses the numerical prediction of the measured wall-static-pressure distribution rather than assuming that a plausible velocity contour demonstrates accuracy.

Roberts (2025) provides the closest local methodological precedent because a pressure-tapped 90° bend was used to assess an internal-flow CFD method before its application to an intake component. Peer-reviewed bend-flow evidence provides the physical interpretation. Sudo, Sumida and Hibara (1998) measured strong primary-flow distortion, secondary motion and Reynolds-stress redistribution through a circular 90° bend. These curvature-driven mechanisms explain why the largest local pressure gradients and the greatest CFD sensitivity are expected in and immediately downstream of the bend.

Validation should not be reduced to matching one tap or one contour. The experimental and numerical pressure references, tap coordinates, flow condition and averaging method must be consistent. Lin, Batista and Milesi-Ferretti (2022) also distinguish local static-pressure measurements from the irreversible pressure loss of a fitting: a wall-tap distribution is appropriate for local CFD comparison, whereas a complete bend-loss coefficient requires suitably developed upstream and downstream sections and an appropriate total-pressure or hydraulic-grade-line method. The present study therefore reports pressure-distribution agreement and treats total-pressure-loss validation separately.

The importance of matching the experimentally tested flow path is illustrated by Masi, Artico and Gobbato (2015), whose steady RANS assessment included the actual bellmouth, ports, valves, chamber roof and cylinder wall and compared discharge coefficients, in-cylinder motion and surface static pressure at several valve lifts. Nanni et al. (2008) similarly found that a finite-volume model could reproduce comparative mass-flow trends despite a persistent absolute bias that mesh and turbulence-setting changes did not remove. These studies indicate that CFD can remain useful for controlled comparative ranking when exact absolute agreement is limited, provided that the validation claim is restricted to the quantities and flow path actually represented.

2.7 Turbulence modelling and wall treatment

The pipe bend and airbox contain wall-bounded flow, streamline curvature, adverse pressure gradients and possible separation. These features make turbulence closure important because different RANS models can predict separation, shear-layer development and pressure redistribution differently. The $k-\omega$ shear-stress-transport (SST) model introduced by Menter (1994) combines near-wall $k-\omega$ behaviour with a blended outer-region formulation and a shear-stress limiter intended to improve predictions under adverse pressure gradients and separation. Its subsequent industrial development and application were summarised by Menter, Kuntz and Langtry (2003). SST is therefore a suitable engineering candidate for curved internal flow, but model suitability should be assessed rather than assumed. Although their application concerned external aerodynamics, Makowski and Kim (2000) showed that RANS closures giving similar pressure predictions in attached flow could diverge substantially downstream of separation. This reinforces the need for turbulence-model sensitivity testing rather than assuming that one closure is universally reliable.

Near-wall treatment is equally important because wall shear, boundary-layer development and separation depend on how the near-wall region is resolved. The dimensionless wall distance y^+ indicates the location of the first computational cell relative to the viscous near-wall region, while prism or inflation layers provide more controlled resolution normal to the wall than a tetrahedral-only mesh. Near-wall adequacy should therefore be judged using y^+ together with solution quantities such as pressure distribution and mass flow, rather than by layer count alone. Comparing plausible turbulence models under otherwise unchanged numerical conditions also helps distinguish model-form sensitivity from mesh-resolution effects (Ansys, 2024a; Ansys, 2024b).

2.8 Mesh sensitivity and numerical uncertainty

Mesh sensitivity is required because a numerically converged solution can still depend on spatial discretisation. Local wall pressure, separation, secondary-flow strength and integral quantities such as mass flow and total-pressure loss may change as surface, volume and near-wall resolution are refined. Celik et al. (2008) recommend systematic mesh refinement and explicit reporting of discretisation uncertainty so that numerical trends can be distinguished from physical configuration effects.

Where geometrically similar meshes and suitable refinement ratios are available, a formal Grid Convergence Index (GCI) can be used to quantify discretisation uncertainty. Roache (1994) proposed the GCI as a standardised way of reporting grid-refinement uncertainty using Richardson-extrapolation concepts, with later CFD reporting guidance formalised by Celik et al. (2008). When the refinement sequence is irregular or does not satisfy the assumptions required for a formal GCI, an engineering mesh-sensitivity assessment can instead compare the response quantities relevant to the study across progressively refined meshes. Such a comparison should consider both global and local quantities and the associated computational cost. The resulting claim should then be one of mesh sufficiency for the stated response quantities, rather than absolute mesh independence at every location (Celik et al., 2008).

2.9 External inlet condition, ram-air effect and boundary sensitivity

Vehicle motion creates a freestream dynamic pressure that may be partially converted into static pressure at an intake opening. In practice, bodywork acceleration, boundary layers, shielding, separation and misalignment reduce the recoverable pressure delivered to the airbox. Intake performance should therefore be described using total-pressure recovery and flow distortion rather than static pressure alone (Anderson, 2016).

The present dissertation does not require a complete external aerodynamic map. Its purpose is to determine whether an inlet-mouth pressure derived from the external vehicle model changes the internal airbox conclusion relative to the ideal uniform condition. One selected straight-line whole-vehicle case is therefore used to derive a sensitivity value. Area-averaged values provide a practical transfer method; a mapped non-uniform profile would be a stronger option if the software workflow were reliable and documented. Because the external model contains no engine through-flow, the transferred pressure is not treated as an installed-engine operating condition.

Table 4. External inlet-condition metrics and interpretation.

External metric	Extraction or expression	Interpretation
Freestream dynamic pressure	$q_{\text{inf}} = 0.5 \rho V_{\text{inf}}^2$	Maximum theoretical pressure associated with vehicle speed.
Intake static-pressure rise	$p_{\text{s,intake}} - p_{\text{s,freestream}}$	Local static-pressure change at the intake plane.
Intake total-pressure coefficient	$C_{\text{pt}} = (p_{\text{t,intake}} - p_{\text{t,freestream}}) / q_{\text{inf}}$	Total-pressure change normalised by freestream dynamic pressure; no gauge-pressure ratio is used.
Mean flow angle	Calculated from area-averaged velocity components.	Alignment of the incoming flow with the internal inlet axis.
Plane non-uniformity	Standard deviation or CV of pressure/velocity over the plane.	Whether a single averaged inlet condition is a strong simplification.

2.10 Steady-state assumptions and unsteady intake behaviour

A real four-stroke engine draws air intermittently as the intake valves open and close. Ling and Tun (2006) modelled the unsteady response of a non-symmetrical Formula SAE manifold to piston-driven pressure fluctuation, while Brennan et al. (2006) demonstrated that airbox-engine interaction can create resonance and cylinder-dependent pressure behaviour. Brusiani, Bianchi and Bianchi d'Espinosa (2010) further used LES to resolve mean and fluctuating pressure and velocity in a race-car airbox. These studies identify real phenomena that a steady RANS model cannot reproduce.

Steady CFD remains appropriate for the present objective because the research question is narrower: to compare geometric resistance and wall-pressure patterns under a common imposed demand. Muckova et al. (2023) provide a useful boundary on that interpretation because their transient CFD and sequential steady flow-bench tests agreed in comparative design direction even though the methods did not impose the same instantaneous outlet condition. Maynes et al. (2002) provide a more demanding race-engine example: their Formula 1 airbox assessment coupled a one-dimensional unsteady engine model to the three-dimensional airbox CFD on a time-step-by-time-step basis, allowing pressure waves and cylinder-specific induction effects to be represented. This demonstrates that a physically representative transient airbox calculation requires credible engine-breathing boundary information rather than simply changing a steady CFD case to a transient solver.

Cycle-resolved simulation would require substantially more information and computational effort. Rutland (2011) notes that LES engine studies may need several full 720-degree cycles before cycle-averaged behaviour is established. A credible transient extension here would therefore require time-dependent cylinder or port pressures, valve timing and temporal-resolution checks, or a coupled one-dimensional engine model. These requirements were outside the approximately three-month project period, so the present results are limited to separate steady cylinder-path responses; resonance, full-cycle filling and engine-power change remain future work.

2.11 Research gap

Published intake studies establish the importance of pressure preservation, cylinder-path consistency, transient engine demand and experimental assessment of CFD. However, validated evidence is not available for the specific MCR-S2N/S2N+ airbox package, and it is unclear whether the available vehicle-derived inlet condition changes the internal ranking or whether a practical geometry change can produce an improvement larger than the observed model-preparation sensitivity.

The study addresses this gap through pipe-bend validation, a limited external inlet assessment, common-boundary CFD of the old and updated airboxes, 16-tap correlation on the updated airbox and a controlled design study. The contribution is a traceable method for comparing internal geometric loss, checking the numerical trends against experimental evidence, testing sensitivity to the available vehicle inlet pressure and selecting a practical modification. It does not claim to reproduce cycle-resolved engine filling or validate the complete external vehicle flow field.

3. Methodology

3.1 Research design and controlled variables

The research has six linked stages. Pipe testing and matching CFD establish the numerical method, which is then assessed against the 16-tap updated-airbox dataset. A clean updated model is defined as D0 and compared with the old airbox. Limited external CFD provides the inlet-pressure sensitivity condition. The design stage then evaluates the D1-D15 concept families under common internal boundaries, including D15 as an explicit interaction check. After selection, clean D0 and the final geometry are repeated for C1-C4 using the same vehicle-derived inlet-pressure sensitivity value.

Table 5. . Research stages, purposes and principal outputs.

Stage	Main purpose	Primary outputs	Priority
1. Flow-bench pipe experiment	Generate a controlled wall-pressure dataset.	Tap pressures, operating-point sweep and measurement limitations.	Validation evidence
2. Matching pipe CFD	Assess mesh, wall layers and turbulence model.	RMSE, correlation, mass balance, mesh sensitivity and turbulence-model sensitivity.	Method validation
3. Updated-airbox correlation	Transfer the retained method to the pressure-tapped updated airbox.	Bench-CFD tap correlation metrics, pressure-bias assessment and regional flow pattern.	Airbox evidence
4. Clean D0 and production comparison	Establish the clean design reference and compare it with the old airbox.	Tapped-to-clean model-preparation sensitivity; D0-versus-old mass flow, velocity and total-pressure loss.	Baseline selection
5. External inlet-condition assessment	Define an external-CFD-derived inlet-pressure sensitivity value without contaminating the controlled geometry ranking.	Old/updated inlet-mouth pressure extraction and definition of a common internal sensitivity condition.	Supporting study
6. Evidence-led design	Diagnose D0, complete the D1-D3 wall-profile study, assess D4/D5 inlet shape, isolate D6-D8 inlet size, D9-D11 transition length and D13 divider behaviour, then complete the lower-tray and combined-design stages.	Controlled D0-D15 comparison, final-design selection, matched inlet-pressure robustness check and whole-vehicle compatibility assessment.	Design contribution

The principal controlled variables are the downstream runner and trumpet geometry, air properties, solver settings, SST k- ω turbulence model, convergence requirements, tap definitions and nominal pressure difference. For each numerical case only one cylinder outlet is active; the other three are walls. The same mesh is reused for all four cylinder cases of a given airbox, and the old and updated geometries retain matched local-sizing logic.

3.2 Experimental flow-bench pipe test

3.2.1 Test piece and instrumentation

A stainless-steel 90° elbow with a nominal internal diameter of 50.8 mm was installed on the SuperFlow SF-600E flow bench. Sixteen wall-static-pressure taps were distributed around the bend and connected to an Evolution Measurement EvoScann P16-A pressure scanner. Taps 1–8 were located along the upper wall of the pipe and Taps 9–16 along the lower wall; the two groups are therefore interpreted as separate wall-pressure paths rather than as one continuous Tap 1–16 sequence. The measured test piece and tap layout were reproduced in the validation CAD. Figure 1 identifies the pressure-tap numbering used throughout the pipe-validation analysis.



Figure 1. Pressure-tap layout on the 90° bend.

The pressure scanner reported absolute pressure for each channel. A common atmospheric reading was therefore recorded for all sixteen channels and retained as a channel-specific zero. The bench was operated at 5, 10, 15 and 20 inH₂O using range 4 and at 25 inH₂O using range 5. The pipe dataset contains one stabilised reading at each operating condition; repeated pipe runs were not available, so no experimental standard deviation is calculated for this dataset.

Table 6. Flow-bench pipe-test equipment and recorded specifications.

Item	Recorded specification / current status
Flow bench	SuperFlow SF-600E; suction-flow operation; 5–20 inH ₂ O on range 4 and 25 inH ₂ O on range 5.
Pressure scanner	Evolution Measurement EvoScann P16-A, 16 channels.
Test piece	Stainless-steel 90° elbow; nominal internal diameter 50.8 mm.
Pressure taps	Sixteen physical pressure taps represented by matched split-face patches in CFD.
Connecting tubes	Sixteen scanner lines connected the pressure taps to the scanner.
Ambient condition	Channel-specific atmospheric zero recorded before testing. Ambient temperature was not recorded and is retained as a stated limitation.

3.2.2 Test procedure

Before testing, the pressure-tap lines and scanner connections were inspected and all sixteen scanner channels were exposed to atmosphere to establish channel-specific zero references. The pipe and flow-bench adaptor were then sealed before testing commenced.

The flow bench was operated sequentially at 5, 10, 15, 20 and 25 inH₂O. The 5–20 inH₂O conditions were tested using bench range 4, while 25 inH₂O used range 5. At each condition, the flow and pressure readings were allowed to stabilise before the sixteen pressure channels were recorded.

One stabilised dataset was recorded at each operating condition. Consequently, the pipe experiment provides an operating-point pressure-distribution dataset but does not provide repeated measurements from which experimental repeatability or standard deviation can be determined. This limitation is considered in the uncertainty assessment.

3.2.3 Experimental data reduction and uncertainty

Each gauge pressure was calculated from the channel-specific atmospheric reading using $p_{g,i} = (p_{\text{test},i} - p_{\text{atm},i}) \times 100$, where the scanner readings are in mbar and the result is in Pa. For comparison of distribution shape, each case was also referenced to Tap 1 using $\Delta p_i = p_i - p_1$. This normalisation removes a constant pressure-datum offset but preserves every tap-to-tap pressure difference.

The present experimental evidence is therefore a pressure-distribution dataset rather than a fully repeatability-qualified loss measurement. In accordance with the separation of instrument, repeatability and test-condition contributions recommended by ASME (2018), the uncertainty assessment distinguishes pressure-scanner accuracy, zero drift, repeatability, tap-position uncertainty and bench-setting uncertainty. Static wall-pressure range is not described as irreversible bend total-pressure loss because developed upstream and downstream total-pressure data were not measured (Lin, Batista and Milesi-Ferretti, 2022).

Table 7. Experimental pressure-data reduction and reporting methods.

Experimental output	Calculation or reporting method
Gauge pressure at each tap	$(p_{\text{test},i} - p_{\text{atm},i}) \times 100 \text{ Pa/mbar}$.
Pressure relative to Tap 1	$\Delta p_i = p_i - p_1$; used to compare pressure-distribution shape while removing a constant datum offset.
Wall-static-pressure range	Maximum minus minimum wall-static pressure among Taps 1–16.
Tap 9–10 pressure difference	$p_{10} - p_9$; lower-wall pressure difference used to assess CFD sensitivity in a region of strong local pressure redistribution.
Repeatability	Not calculated for the pipe dataset because only one stabilised reading was recorded at each operating condition.
Validation limitation	The wall-static-pressure distribution is not equivalent to irreversible bend total-pressure loss.

3.3 CFD replication and validation of the pipe test

3.3.1 Geometry and boundary conditions

The CFD fluid volume reproduced the internal pipe and bend geometry and included sixteen split-face wall patches at the experimental tap locations. The pressure patches were retained at a nominal surface size of 0.5 mm in every mesh so that sampling resolution did not change during the volume-mesh study. The atmospheric end was defined as a 0 Pa gauge pressure boundary, while the bench end used a negative gauge pressure corresponding to the applied suction; at 20 inH₂O this was -4982 Pa. Boundary signs and zone definitions were checked before solution. The scanner data were converted separately from absolute pressure to atmospheric-referenced gauge pressure for experimental comparison.

The principal validation condition was 20 inH₂O. Air was treated as a constant-property fluid, the pipe walls were smooth and no-slip, and static pressure was extracted as an area-weighted average over each tap patch. The pressure curves were compared both in their reported Fluent gauge form and after reference alignment. The aligned comparison is used only for curve shape; absolute-pressure validation requires the same physical datum in the experiment and CFD.

Table 8. Pipe CFD settings and implementation rationale.

CFD setting	Implementation and rationale
Solver	Pressure-based, steady solver with coupled pressure-velocity solution, selected for robust low-Mach internal-flow convergence.
Spatial schemes	Second-order discretisation was used to reduce first-order numerical diffusion through the bend.
Turbulence model	SST k-omega was used for the retained production simulations after a sensitivity comparison against Realizable k-epsilon and RNG k-epsilon; the comparison procedure is described in Section 3.3.3 and the results are reported in Section 4.3.4.
Fluid	Air was modelled with constant reference properties stored in the Fluent case; the same property definition was retained across comparable simulations.
Pressure boundaries	The inlet used a 0 Pa gauge atmospheric reference and the outlet reproduced the bench depression from 5 to 25 inH ₂ O; at 20 inH ₂ O the outlet pressure was -4982 Pa. Boundary signs and zone names were checked before each run.
Walls and near-wall mesh	Smooth no-slip walls. Near-wall sensitivity compared L0, L4, L8 and L12; the prism cases used a 0.05 mm first-layer height and 1.2 growth ratio. The retained production setting is reported in Section 4.3.3.
Pressure extraction	Area-weighted average static pressure was extracted from the Tap 1-16 wall patches to match the measured wall-static-pressure quantity.

3.3.2 Pipe mesh and convergence

Five volume meshes, M0–M4, were generated in ICEM CFD using systematic reductions in global, body-of-influence, pipe-wall and bend-wall sizing. The meshes were compared at 20 inH₂O using the complete Tap 1–16 pressure distribution, the Tap 9–10 local pressure change, the reported inlet–outlet pressure difference, mass flow, mass balance and runtime. A formal Grid Convergence Index was not calculated because the refinement sequence did not provide sufficiently uniform refinement ratios. A separate near-wall study was then performed so that prism-layer count could be varied independently of the volume-mesh refinement. The dedicated cases comprised a tetrahedral-only L0 reference and L4, L8 and L12 prism meshes. The prism cases used a 0.05 mm first-layer height and a growth ratio of 1.2, and their Fluent cell-type reports were checked before solution.

Each pipe case was advanced to a common 1000-iteration endpoint for consistent comparison. Convergence acceptance was based on inlet/outlet mass-flow balance and stability of the sixteen tap monitors together with the residual histories, rather than on residual magnitude alone. Numerical convergence outcomes are reported with the mesh-sensitivity results in Section 4.3.

3.3.3 Validation assessment

The validation assessment was conducted at 20 inH₂O using a 0 Pa gauge pressure inlet and a –4982 Pa gauge bench-side pressure outlet. Volume-mesh, near-wall and turbulence-model effects were assessed sequentially so that only one numerical choice was varied at each stage. The near-wall comparison used L0, L4, L8 and L12; turbulence-model sensitivity compared SST $k-\omega$, Realizable $k-\epsilon$ and RNG $k-\epsilon$ with the selected mesh and near-wall treatment otherwise held fixed. Selection was based on the complete 16-tap pressure distribution, RMSE, MAE, correlation, Tap 9–10 pressure change, mass flow, mass balance, monitor stability and computational cost. The numerical outcomes and production selection are reported in Section 4.3. The selected method was then applied across the 5–25 inH₂O pressure sweep.

Table 9. Pipe-validation metrics and interpretation.

Validation metric	Expression or interpretation
Tap-1-referenced pressure	$\Delta p_i = p_i - p_1$; compares pressure-distribution shape without a constant datum offset.
Aligned RMSE / MAE	Error after applying one common constant pressure offset; used as a shape metric, not absolute-pressure proof.
Correlation coefficient	Agreement in the rise, fall and recovery pattern across all sixteen taps.
Tap 9–10 pressure difference	Local lower-wall pressure difference, $p_{10} - p_9$; used to expose model sensitivity in the strongly redistributed bend region.
Mesh contribution	Change in tap distribution, reported boundary pressure difference and engineering interpretation between meshes.
Validation claim	Restricted to the tested 90° bend, operating range and wall-static-pressure distribution until total-pressure-loss data are available.

3.4 Limited external inlet-condition CFD

A limited whole-vehicle CFD assessment was completed for the old- and updated-airbox installations at the same 35 m/s straight-line vehicle condition. Its purpose was to estimate the local pressure field at each airbox opening for a boundary-condition sensitivity study, not to develop the whole-car aerodynamics or reproduce engine induction. External mesh sizing was informed by the available SABE vehicle work. Area-weighted static and total pressure were extracted at each airbox inlet face, while vehicle and airbox force coefficients were retained as supporting installation-level quantities.

The area-averaged total pressure obtained at the updated-airbox inlet face was subsequently transferred uniformly to clean D0 and, after design selection, to the selected final geometry as a matched internal inlet-pressure robustness condition. The numerical pressure values and comparisons are reported in Section 4.7. Because the external model contained no engine through-flow, the transferred pressure was treated as a boundary-condition sensitivity value rather than as the actual pressure available during engine induction. A later matched whole-vehicle comparison between the updated-airbox installation and the selected final geometry was performed separately to assess installed-geometry compatibility.

Table 10. Whole-vehicle external CFD boundary conditions and modelling assumptions.

Boundary or setting	Implemented choice	Reason / limitation
Inlet	35 m/s velocity inlet	Characterise the local airbox-opening pressure field; no engine through-flow is represented.
Outlet	Pressure outlet / atmospheric reference	Allows external flow to leave the domain.
Ground	Stationary no-slip wall	Recorded simplification; no moving-ground validation claimed.
Vehicle body	No-slip wall	Captures body boundary layers and local separation.
Wheels	Stationary/simplified	Not a core development variable.
Turbulence model	SST k-omega	Consistent with the retained engineering model.
Sampling plane	Airbox inlet face	Provides area-weighted static and total pressure for boundary transfer.

The external study is restricted to the tested straight-line condition and a uniform pressure transfer. It does not establish sensitivity to speed, yaw, ride height or inlet distortion; a measured or mapped inlet profile would be required if those effects became a primary research question.

3.5 Internal steady airbox CFD

The main CFD application uses cleaned internal fluid domains for the old and updated airboxes, including the inlet, plenum, trumpet and runner passages. No air-filter element was included in these internal domains. This matched the airbox flow-bench configuration used for the present comparison and avoided introducing an additional porous-media or filter-pressure-loss model whose characteristics were not measured in this project. Four separate outlet face zones were retained so that the same mesh could be reused for every cylinder case. For each solution, the selected cylinder outlet was defined as a pressure outlet and the other three outlets were changed to walls. The solution was reinitialised before each cylinder case. Two updated-airbox model preparations are distinguished in the results. The pressure-tapped model contains the sixteen split-face sampling patches required for the CFD-to-bench wall-pressure correlation. After that correlation was completed, a clean updated-airbox model was prepared for design comparison by omitting the tap split faces and applying a small approximately 0.5 mm rearward correction to the lower opening. Because both changes were made together, the resulting comparison is treated as a model-preparation sensitivity check rather than as a pure pressure-tap effect. The clean model is designated D0 and is the formal reference geometry for the D1–D15 design programme.

For every internal airbox CFD case, the inlet was defined as a pressure inlet with 0 Pa total gauge pressure and the active cylinder outlet was set to -4982 Pa gauge, numerically equivalent to 20 inH₂O; the other three cylinder outlets were closed. The same outlet pressure was retained for Cylinders 1-4 on the old, tapped-updated, clean D0 and design airboxes to provide a controlled comparative demand. Although this value matched the nominal bench setting numerically, it did not reproduce the experimental pressure distribution across the complete intake assembly because the cylinder head, intake port and partially open valve were excluded from the CFD domain. The boundary condition was therefore used to compare relative geometric performance and pressure-distribution trends, not to represent the pressure drop across the airbox alone or a specific operating-engine condition. This steady one-cylinder-at-a-time condition also does not represent valve motion, cycle-resolved pressure waves or interaction between cylinders (Ling and Tun, 2006; Brennan et al., 2006).

Table 11. Internal airbox CFD case groups, geometry definitions and purposes.

Case group	Geometry	Boundary definition	Purpose
Tapped updated C1-C4	S2N+ updated airbox with 16 tap split faces	0 Pa total-pressure inlet; one active -4982 Pa pressure outlet	CFD-to-bench spatial wall-pressure correlation
Clean D0 C1-C4	Clean S2N+ updated-airbox design-study model; tap faces omitted; ~0.5 mm lower-opening correction	Same 0 Pa inlet and -4982 Pa active outlet	Controlled old/new comparison and design baseline
Old airbox C1-C4	Older airbox; matched runners/trumpets	Same inlet and active-outlet pressure difference	Controlled geometry comparison
D1-D15 design concepts	Targeted modifications derived from clean D0: D1-D3 wall profile; D4/D5 inlet-opening shape; D6-D8 inlet size; D9-D11 transition length; D12/D14 trumpet-tray extension/volume; D13 central divider; D15 combined D9 + D12.	Same 0 Pa inlet and -4982 Pa active outlet	Controlled single-variable screening followed by a combined-feature interaction check.
Vehicle-derived inlet robustness cases	Clean D0 and the selected final geometry	Uniform inlet total gauge pressure derived from the updated-airbox external pilot; one active -4982 Pa pressure outlet. Numerical inlet value reported in Section 4.7.	Matched inlet-pressure robustness comparison after controlled design selection; sensitivity condition only, not an installed-engine operating point.
Bench correlation	Pressure-tapped updated airbox on engine/head	20 inH ₂ O; 10 mm valve lift; fully open throttle; no air-filter element installed	Compare the 16-tap pressure pattern for each cylinder

3.6 Boundary-condition definition and transfer

The external and internal models are separate computational domains. Area-averaged pressure at the updated-airbox opening was transferred as a uniform total-pressure inlet only for the vehicle-derived sensitivity cases. This transfer tests whether the internal ranking is robust to the external-CFD-derived inlet-mouth pressure; it does not define a complete operating-engine boundary condition because the external model contains no engine through-flow and the internal outlet pressure is a controlled numerical demand. The production old-versus-updated and subsequent design comparisons therefore used the same 0 Pa total-pressure inlet so that changes in outlet velocity, mass flow and total-pressure loss could be attributed directly to internal geometry rather than to different external pressure supply.

For every cylinder case, report definitions recorded inlet and active-outlet mass flow, area-weighted total pressure, area-weighted velocity magnitude and area-weighted wall static pressure in accordance with the averaging definitions in the Fluent documentation (Ansys, 2024b). Total-pressure loss was calculated as $\Delta p_t = \text{mean}(p_{t,\text{in}}) - \text{mean}(p_{t,\text{out}})$, and the normalised loss coefficient as $K_t = \Delta p_t / 4982 \text{ Pa}$. A total-pressure ratio based on gauge pressures was not used because the inlet total gauge pressure is 0 Pa. Mass imbalance was calculated from the signed inlet and outlet mass-flow rates and checked together with residuals and monitor stability (White, 2011).

3.7 Meshing, convergence and verification

The pressure-tapped updated-airbox mesh used 3 mm main surfaces, 2 mm trumpet/runner surfaces, a 3 mm body of influence, 0.5 mm tap faces, a 6 mm maximum volume-cell length and eight prism layers (0.05 mm first layer; 1.2 growth). The old-airbox mesh used matched local-sizing logic. Clean D0 and the wall concepts omitted tap-face refinement. D4/D5 required 1 mm inlet faces and a 3 mm inlet-region body of influence. Because D0 was not regenerated with this inlet refinement, D4/D5-versus-D0 is treated as a screening comparison, whereas D4-versus-D5 provides the more directly matched mesh comparison. Final cell counts and achieved mesh-quality values are retained in Appendix E.1.

Table 12. Production CFD mesh definitions and decisions.

Model	Mesh definition	Methodological role
Pipe validation CFD	Production mesh and near-wall definition selected through the staged sensitivity procedure in Section 3.3.	Applied to the pressure sweep and transferred as the numerical-method basis for the airbox study.
Tapped updated correlation model	6 mm max volume; 3 mm BOI/main surfaces; 2 mm trumpets/runners; 0.5 mm tap faces; eight prism layers.	Used for the 16-tap CFD-to-bench correlation. Achieved cell count and quality are reported in Appendix E.1.
Clean D0 / design models	Matched 6 mm max-volume, 3 mm main/BOI and 2 mm trumpet logic; tap-face refinement omitted.	Common design-study reference and concept mesh logic.
Old airbox	Matched production sizing logic relative to the updated-airbox model.	Used for the controlled old-versus-updated geometry comparison; achieved mesh data are reported in Appendix E.1.

Production meshes were checked and improved against an orthogonal-quality threshold of 0.05 before transfer to the solver. Negative-volume checks and clipped views of the prism layers were also reviewed, following the quality checks described by Ansys (2024b). Each solution recorded residual history, inlet/outlet mass balance and stability of mass flow, total pressure and tap-pressure monitors. Achieved mesh-quality values are reported in Appendix E.1.

3.8 Validation, correlation and data reduction

Direct validation is claimed only for the wall-static-pressure distribution of the pressure-tapped pipe at matching tap locations and bench depressions. The updated-airbox measurements provide an additional correlation dataset, but the assembled bench system includes the cylinder head, port and 10 mm-open valve whereas the CFD outlet is located at the runner end. The airbox comparison therefore assesses pressure-pattern agreement and relative cylinder trends, not exact absolute flow-rate validation.

For the updated-airbox bench test, each tap pressure was referenced to the corresponding replacement-test ambient channel: $\Delta p_i = (p_{\text{test},i} - p_{\text{ambient},i}) \times 100 \text{ Pa}$. Tap means and sample standard deviations were calculated from three valve-open repeats for every cylinder, with repeatability and datum effects kept separate in the uncertainty interpretation (ASME, 2018). The CFD-to-bench assessment used tap-wise error, MAE, RMSE, mean error and Pearson correlation for each cylinder. Figure 2 records the individual tap numbering and the three regional groups used in the subsequent tap-wise and regional comparisons.

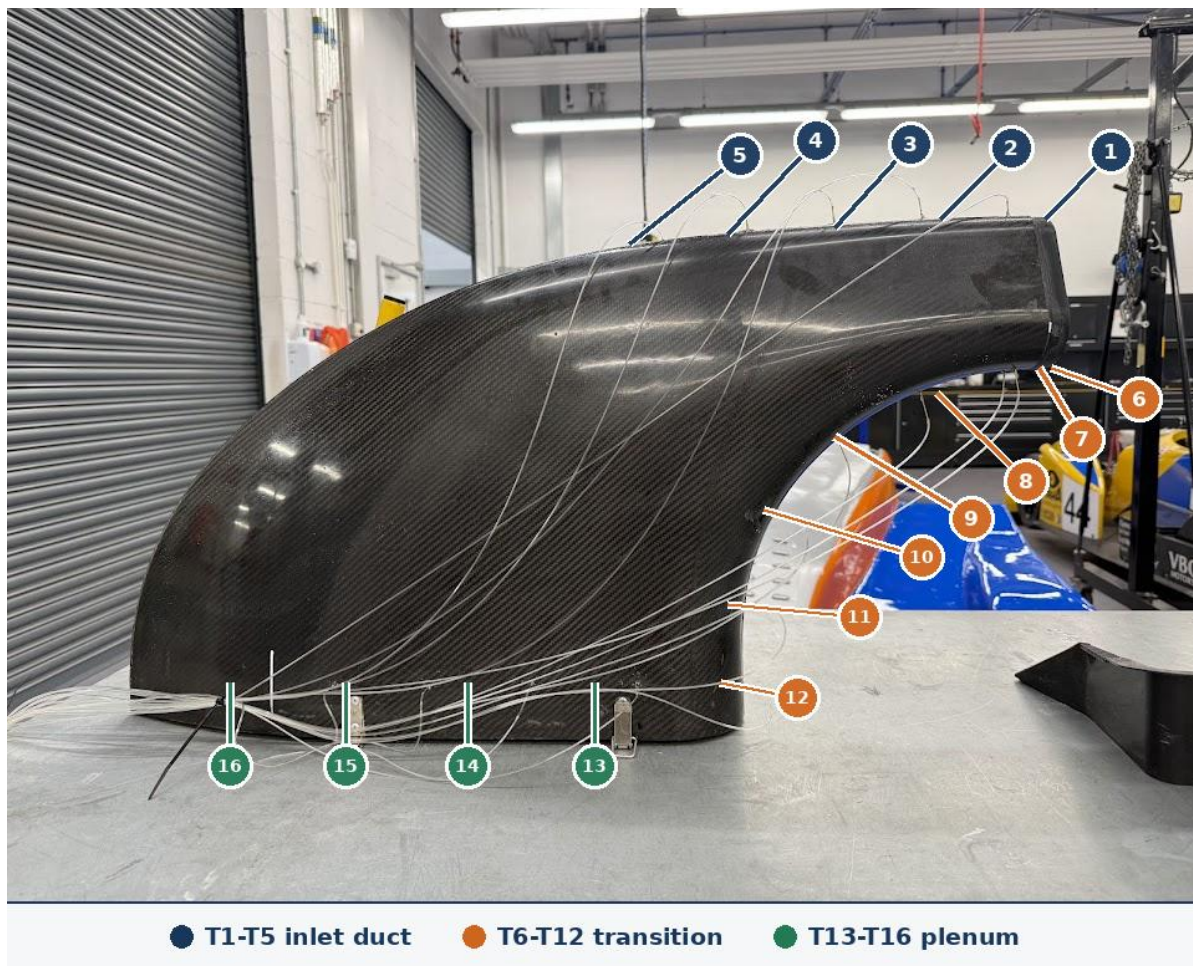


Figure 2. Pressure-tap layout on the updated airbox (author's photograph).

Table 13. Validation, correlation and verification evidence levels.

Evidence level	Example	Permitted claim
Direct validation	Pipe CFD compared with measured tap pressures at matched depressions	Validated for the tested bend and wall-static-pressure distribution.
Airbox correlation	Updated-airbox CFD tap pressures compared with the 16-tap bench pattern	Spatial and cylinder-trend support; not exact absolute-flow validation.
Verification	Mesh quality, mass balance and monitor stability	Numerical consistency, not experimental accuracy.
Controlled comparison	Old versus updated airbox under identical one-cylinder boundary definitions	Relative performance within the steady modelling assumptions.

3.9 Numerical and experimental uncertainty

The final updated-airbox dataset contains three open-valve repeats for every cylinder, with all spark-plug holes sealed. One ambient reading was recorded for each cylinder set, so slow within-set zero drift cannot be quantified using before-and-after references. The repeat ranges are therefore reported directly.

Table 14. Principal experimental and numerical uncertainty sources.

Uncertainty source	Potential effect	Mitigation or reporting
Scanner zero and display resolution	Bias or quantisation in ambient-referenced tap pressure	Channel-specific ambient correction; 0.01 mbar equals 1 Pa.
One ambient per cylinder set	Unremoved zero drift within a set	Report as a limitation and use repeat range/SD.
Three repeats per cylinder	Limited estimate of repeatability	Report mean, sample SD and run-mean range for each cylinder.
Valve/head absent from CFD	Different restriction downstream of the runner	Treat airbox result as spatial correlation rather than exact flow validation.
Tap position and numerical patch	Non-equivalent pressure sampling	Match tap coordinates and use 0.5 mm numerical tap faces.
Steady one-cylinder demand	No pulsation or cylinder overlap	Limit conclusions to comparative mean behaviour.
External-profile simplification	May omit non-uniform real inlet flow	Use pilot evidence and reserve mapped/experimental profile for further work.
Air-filter element omitted	Installed-system restriction and inlet flow conditioning are not represented	Interpret results as unfiltered airbox behaviour; repeat the final comparison with the intended filter in future work.

3.10 Design development methodology

The design stage begins after the tapped correlation model has been separated from clean D0, the old-versus-D0 comparison is complete and the external pilot has supplied an inlet-pressure sensitivity condition. All concept screening uses the same 0 Pa inlet total pressure and -4982 Pa active-cylinder outlet as D0 so that changes are attributed to geometry rather than pressure supply or tap-specific model preparation. After selection, clean D0 and the chosen geometry are repeated for C1-C4 using the same external-derived inlet-pressure condition.

3.10.1 Baseline selection

The old and updated airboxes were compared using mean and worst-case total-pressure loss, mass flow and the spread of the four separately simulated cylinder-path results. The design baseline was selected from the combination of hydraulic performance, package compatibility and manufacturing practicality rather than a single best cylinder or the lowest local wall-static pressure. Where rankings were mixed, the result was recorded as a trade-off rather than forcing an unsupported overall winner.

3.10.2 Flow-mechanism diagnosis

The retained baseline was diagnosed using common-scale total-pressure contours, velocity vectors, streamlines and reverse-flow or low-momentum regions for the limiting and representative cylinder paths. The diagnosis linked measurable penalties to physical mechanisms such as inlet-jet impingement, excessive turning, abrupt expansion, restricted trumpet clearance or recirculation. Static-pressure recovery was not treated as proof of low loss unless the total-pressure field supported the same interpretation (Anderson, 2016; Versteeg and Malalasekera, 2007).

3.10.3 Design requirements and constraints

Requirements were defined before concept generation so that an aerodynamically attractive but unusable geometry could not become the preferred design. Packaging and manufacturing constraints were assessed using the available CAD package and manufacturing discussion. Where an exact dimensional limit was not independently quantified, the existing interfaces and practical access envelope were retained rather than introducing an unconstrained geometry change.

Table 15. Airbox design requirements, constraints and acceptance conditions.

Requirement	Design rule	Evidence / measurement	Acceptance condition
Fixed flow hardware	Retain the production runner and trumpet geometry.	Matched CAD and named faces.	No change to controlled downstream geometry.
Package envelope	Remain within the available engine-cover and service envelope.	CAD clearance check.	No interference; required access retained.
Inlet interface	Retain the existing interface unless the baseline diagnosis directly implicates it.	Inlet dimensions and external-pilot evidence.	Boundary remains comparable or change is explicitly justified.
Manufacture	Use feasible wall thickness, radii, splits and tool/print access.	Proposed process and supervisor/manufacture r review.	Concept can be manufactured and assembled.
Instrumentation	Preserve required tap, fastener and inspection access.	CAD access review.	No essential access is blocked.

3.10.4 Concept generation

Concepts were derived from the Section 4.8 flow diagnosis. D1-D3 changed the inlet-to-plenum wall profile; D4/D5 changed inlet shape; D6-D8 changed inlet size; and D9-D11 changed transition length. D13 added a short central divider. Roberts (2025) had found that a longer turning-vane arrangement in a different but broadly similar plenum could create blockage and recirculation, which supported testing only a short local divider here. D12 and D14 extended the trumpet tray by 50 and 100 mm, and D15 combined D9 with D12 to test interaction. Selection used total-pressure loss, worst-path behaviour, mass flow, consistency, numerical sensitivity and manufacturability; acoustic and pressure-wave effects remained outside the steady CFD scope.

3.10.5 Concept screening

Concept screening used all four separate cylinder-path cases with the same inlet and outlet pressures, turbulence model, discretisation, convergence checks and matched local-meshing rules. The limiting or most representative paths were then used for detailed flow-field visualisation so that the mechanisms behind the quantitative results could be interpreted without overloading the main chapter. A concept could be rejected for increased mean or worst-path loss, deterioration of path consistency, migration of the loss region to another critical area, poor packaging or lack of a plausible mechanism even if one local contour appeared improved. This staged use of quantitative screening and mechanism-focused visualisation is consistent with the multilevel approach advocated by Montenegro et al. (2013).

3.10.6 Final-design verification and decision rule

D0 and each design concept were compared for Cylinders 1-4 using the controlled 0 Pa inlet total pressure and the common -4982 Pa active-cylinder outlet. The mean and worst-case results were evaluated together with the four-case range and coefficient of variation. A modified design was eligible for selection only if the mean performance improved, the worst cylinder path did not deteriorate materially, the change was interpreted against the relevant numerical sensitivity (Celik et al., 2008), and the package/manufacturing review was acceptable. Combined features were assessed using the same decision rule rather than assumed to be additive. After final design selection, clean D0 and the selected geometry were repeated using the same external-CFD-derived inlet-pressure sensitivity condition and the common -4982 Pa active-cylinder outlet. The selected geometry was also assessed separately in a later whole-vehicle external calculation.

Table 16. Final design decision measures.

Decision measure	Calculation	Reason for inclusion
Mean total-pressure loss	Mean Delta p_t across C1-C4.	Overall restriction under matched steady demand.
Worst-case loss	Maximum Delta p_t among C1-C4.	Prevents an improved mean from hiding one poor path.
Mean outlet velocity / mass flow	Mean area-weighted active-outlet velocity and mass flow across C1-C4.	Flow delivery under the common imposed pressure difference.
Cylinder-path consistency	Range, sample SD and CV of four separate case results.	Quantifies spread without claiming simultaneous distribution.
Numerical significance	Design change compared with the available model-preparation and numerical-sensitivity evidence.	Checks whether a small geometry effect could be within the observed numerical variation.

Decision measure	Calculation	Reason for inclusion
Practical feasibility	Package, access and manufacturing review.	Converts CFD improvement into an engineering recommendation.
Vehicle-derived inlet-pressure robustness check	Repeat clean D0 and the selected final geometry for C1-C4 using the same uniform inlet total pressure derived from the external updated-airbox pilot.	Checks whether the selected geometry retains its improvement direction under the external-CFD-derived sensitivity condition; does not represent fully coupled engine flow.

3.11 Methodological limitations and research ethics

Agreement in the simpler pipe bend does not remove uncertainty in the more complex airbox. The external study is only a straight-line inlet-condition pilot, and the internal model omits the cylinder head, port, valve motion and cycle-resolved demand. Its outlet pressure is therefore a controlled numerical boundary rather than a measured runner/head pressure. No air-filter element was fitted in the bench test or CFD, so the reported losses and design ranking apply to the unfiltered configuration.

The final airbox test series contains three open-valve repeats for every cylinder, with all spark-plug holes sealed. The project also uses CAD and vehicle information supplied within the MCR/UWTSD programme. These materials were used only for the academic purposes of this project and within the permissions under which they were provided. Full proprietary CAD geometry is not distributed as part of the dissertation; only the views and derived results required to explain the analysis are presented. Company, SABE, MIRA, previous-student, dyno and track information is acknowledged where used and is not presented as independently generated data. The study involved no human participants or personal data.

4. Results

This chapter presents the experimental and numerical results in the order established by the methodology. It begins with pipe-method validation and airbox CFD-to-bench correlation, followed by the controlled old-versus-updated airbox comparison and external inlet-condition assessment. The subsequent design results evaluate the effect of the investigated geometric modifications before identifying and verifying the final selected geometry.

4.1 Test-rig and geometry preparation

The pipe-validation rig comprised the SuperFlow SF-600E, a stainless-steel 90-degree elbow, sixteen wall-static-pressure taps and the EvoScann P16-A scanner. The S2N+ updated airbox was later fitted with a separate set of sixteen labelled taps. It was tested on the assembled intake with the throttles fully open, the selected intake valve fixed at 10 mm lift and each cylinder drawn separately at 20 inH₂O. No air-filter element was installed during these airbox tests, matching the unfiltered internal-CFD configuration. The other three flow paths remained closed during each cylinder test. Figures 1 and 2 provide the photographic record of the two pressure-tapped test pieces.

4.2 Experimental flow-bench pipe results

Table 17. Experimental pipe wall-static-pressure results across 5–25 inH₂O.

Bench setting	Maximum gauge pressure [Pa]	Tap	Minimum gauge pressure [Pa]	Tap	Tap pressure range [Pa]
5 inH ₂ O	–1121	5	–2763	12	1642
10 inH ₂ O	–1973	3	–5546	12	3573
15 inH ₂ O	–3089	4	–8182	12	5093
20 inH ₂ O	–4432	4	–11341	12	6909
25 inH ₂ O	–5658	4	–14203	12	8545

The five experimental datasets retained the same broad wall-pressure pattern as the bench depression increased. After reference to Tap 1, the correlation with the 20 inH₂O dataset was greater than 0.97 for every other operating point. Along the upper wall (Taps 1–8), wall static pressure generally increased through Taps 1–4/5 before decreasing towards Taps 7–8. Along the lower wall (Taps 9–16), pressure decreased from Tap 9 towards a minimum at Tap 12 and then increased through Taps 13–16. Tap 8 and Tap 9 are on different wall paths and are therefore not interpreted as adjacent points in a continuous pressure progression. The minimum pressure occurred at Tap 12 for all five cases, while the overall measured wall-static-pressure range increased from 1.64 kPa at 5 inH₂O to 8.55 kPa at 25 inH₂O.

At 20 inH₂O, Tap 4 was the maximum-pressure location and Tap 12 was the minimum-pressure location, giving a 6.91 kPa wall-static-pressure range. The Tap 9–10 lower-wall pressure difference was –1.559 kPa. The measured pressure range should not be interpreted as the irreversible total-pressure loss across the bend, because the taps measure local wall static pressure rather than developed upstream and downstream total pressure.

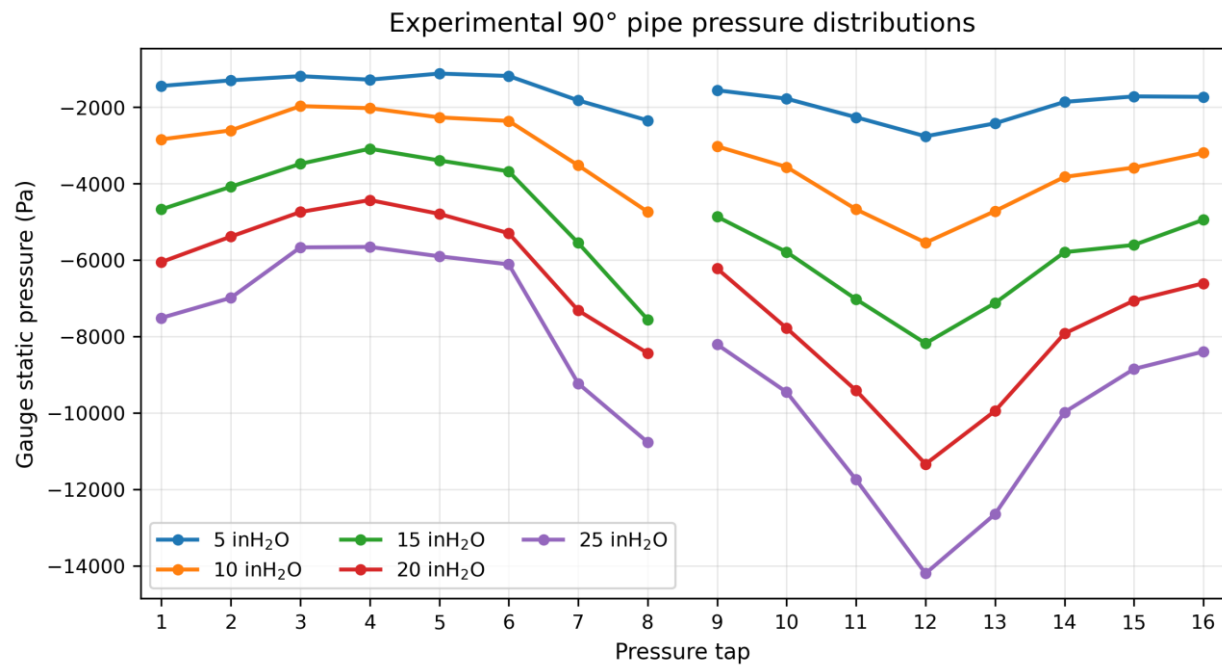


Figure 3. Measured bend wall-static-pressure distributions at 5–25 inH₂O.

4.3 Pipe CFD validation and mesh sensitivity

4.3.1 Volume-mesh sensitivity and M3 selection

Table 18. Pipe volume-mesh sensitivity at 20 inH₂O.

Mesh	Total ICEM elements	Runtime	Reported inlet-outlet static-pressure difference [Pa]	Difference from M4 [%]	Assessment
M0	345,449	15 min	726.5	30.3	Intentionally coarse; visibly mesh-sensitive.
M1	667,868	25 min	1017.2	2.4	Large improvement from M0; pressure curve close to refined cases.
M2	1,308,147	55 min	1028.9	1.3	Refined pressure pattern; similar total count to M3.
M3	1,369,206	57 min	1021.9	2.0	Selected volume mesh: refined local sizing with low additional cost.
M4	6,729,579	5 h	1042.3	Reference	Fine reference; large computational increase for small change.

At 20 inH₂O, the inlet-to-outlet static-pressure difference changed markedly from M0 to M1, while M1–M3 remained within 2.4% of the M4 result. M3 required 57 minutes, only two minutes more than M2, whereas M4 required approximately five hours and contained almost five times as many ICEM elements as M3. The Tap-1-referenced M3 pressure curve was also close to M4, with an inter-mesh RMSE of 38.8 Pa and a maximum difference of 101.2 Pa. M3 was therefore retained as the production volume mesh based on numerical sufficiency, local resolution and computational cost rather than the closest experimental RMSE alone.

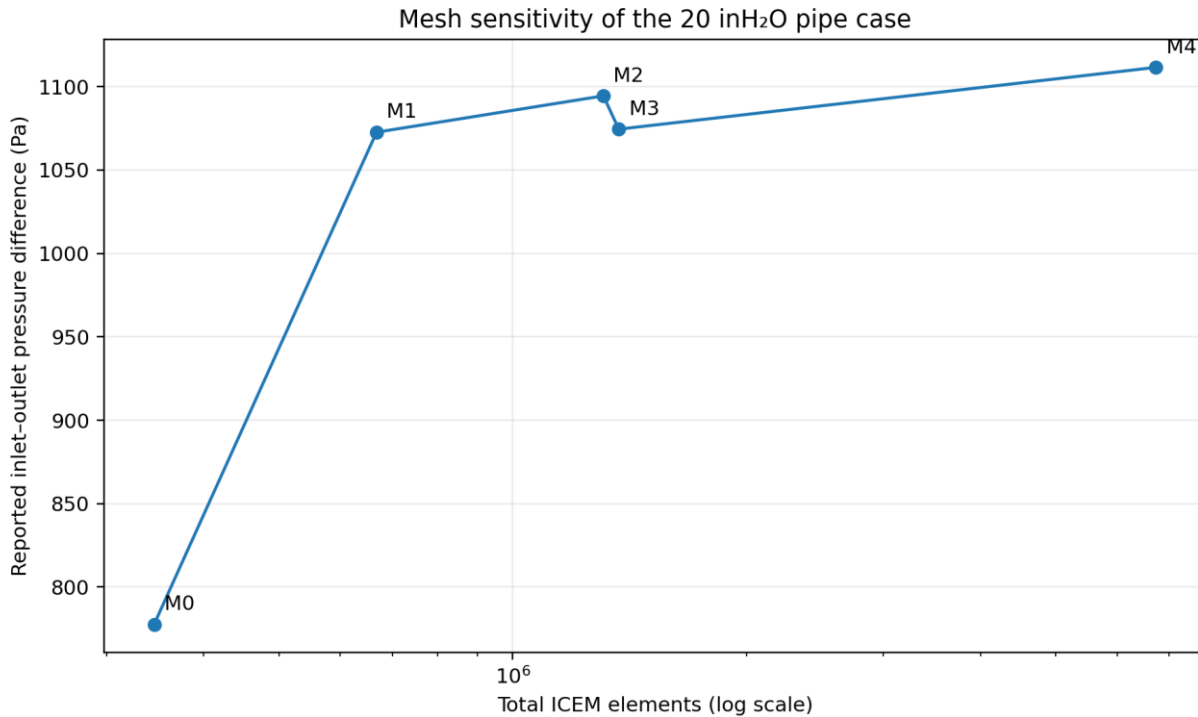


Figure 4. Mesh sensitivity of the inlet–outlet pressure difference at 20 inH₂O.

4.3.2 20 inH₂O pressure-distribution validation

Table 19. Pipe CFD-to-bench pressure-distribution comparison at 20 inH₂O.

Mesh	RMSE [Pa]	MAE [Pa]	Correlation r	Tap 9–10 Δp [Pa]	Mass flow [kg s ⁻¹]
M0	1064.1	748.6	0.924	–1033.3	0.18085
M1	723.9	542.0	0.946	–968.2	0.17512
M2	742.9	532.9	0.942	–968.4	0.17488
M3	745.1	542.6	0.943	–976.1	0.17501
M4	739.6	532.8	0.943	–972.4	0.17471

The M1–M4 solutions reproduced closely grouped pressure-distribution shapes. M1 gave the lowest experimental RMSE, but the refined cases differed only modestly. M3 was retained because its local surface and bend sizing was finer than M1 while its result remained close to M2 and M4. For M3, the Tap-1-referenced RMSE was 0.745 kPa, the MAE was 0.543 kPa and the correlation coefficient was 0.943. The measured Tap 9–10 pressure change was –1.559 kPa, whereas M3 predicted –0.976 kPa, underpredicting the local reduction by approximately 37.4%. Both the experiment and M3 solution placed the minimum pressure at Tap 12. The method therefore captures the overall wall-static-pressure redistribution but retains a material local discrepancy in the strongly curved-flow region. The concentration of the remaining error in the curved-flow region is consistent with the strong secondary-flow redistribution measured in circular 90-degree bends (Sudo, Sumida and Hibara, 1998).

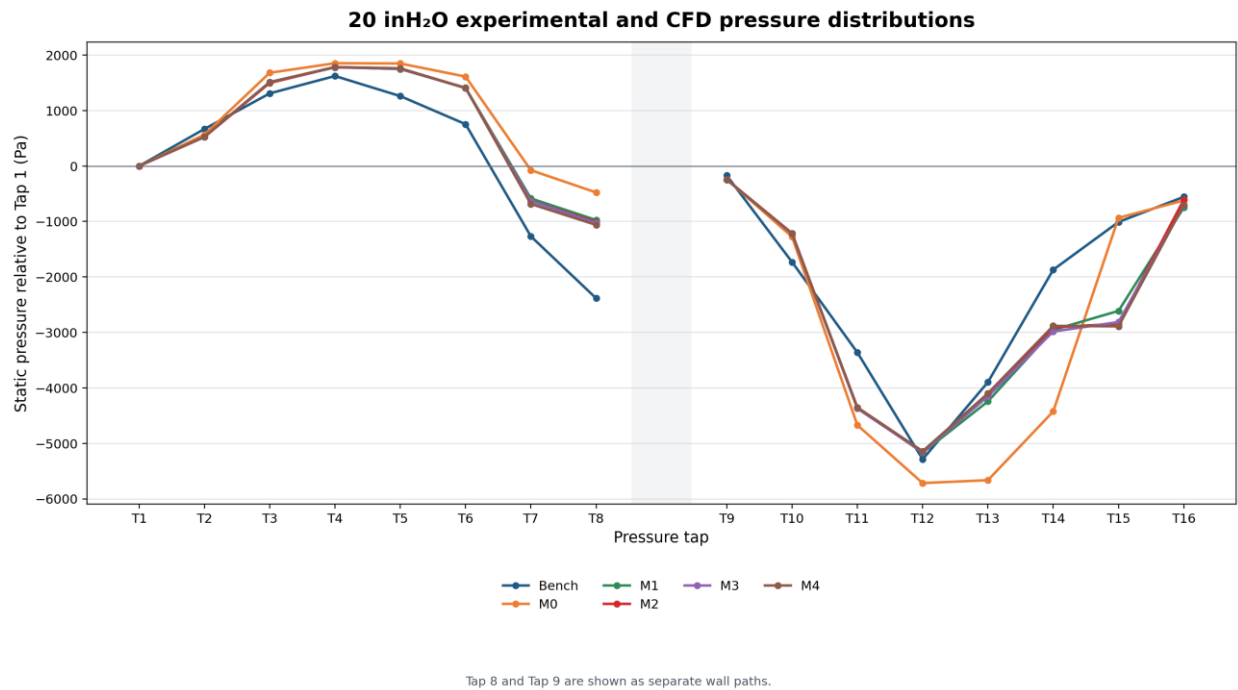


Figure 5. Bench–CFD wall-static-pressure comparison for the 90° bend at 20 inH₂O.

4.3.3 Prism-layer sensitivity and near-wall resolution

Near-wall sensitivity was assessed on the M3 mesh using zero, four, eight and twelve prism layers (L0, L4, L8 and L12). The prism cases retained a 0.05 mm first-layer height and 1.2 growth ratio, with all other numerical settings fixed at the 20 inH₂O condition. Tap pressures were referenced to Tap 1 to compare changes in distribution shape.

Table 20. Prism-layer sensitivity on the M3 pipe mesh.

Case	Layers	Mass flow [kg s ⁻¹]	RMSE vs L12 [Pa]	Max difference vs L12 [Pa]	RMSE vs bench [Pa]
L0	0	0.17960	649.5	1804.6	972.7
L4	4	0.17756	177.1	550.1	782.0
L8	8	0.17500	96.4	289.5	746.7
L12	12	0.17503	Reference	Reference	758.2

Relative to L12, pressure-distribution RMSE decreased from 649.5 Pa for L0 to 177.1 Pa for L4 and 96.4 Pa for L8. The L8-L12 inlet mass-flow difference was only 0.016%. The Tap 9–10 lower-wall pressure difference was -0.975 kPa for L8 and -0.974 kPa for L12. L8 also gave the lowest bench RMSE among the prism cases, although experimental agreement was not the only basis for selection. This response-based comparison is consistent with using engineering mesh-sensitivity evidence when the assumptions required for a formal GCI are not satisfied (Celik et al., 2008).

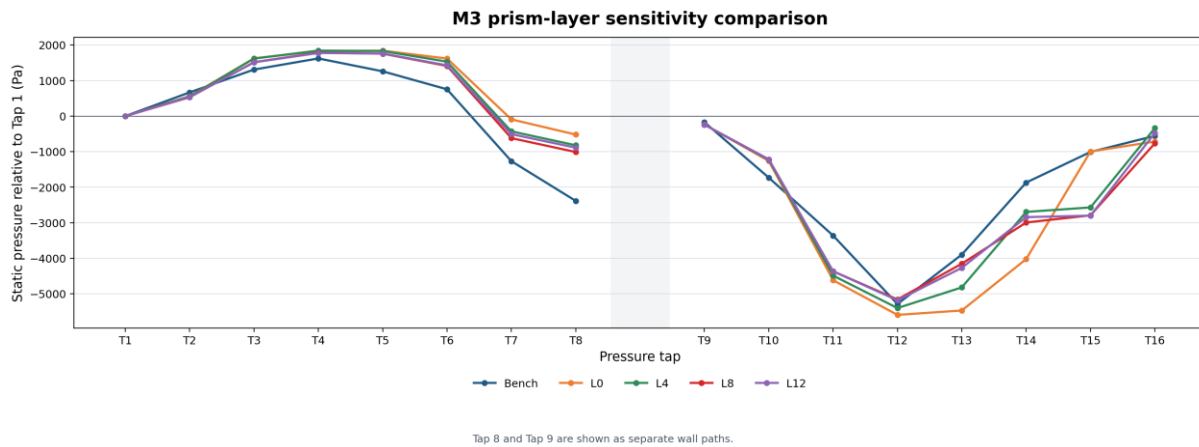


Figure 6. Prism-layer sensitivity of bend wall-static-pressure at 20 inH₂O.

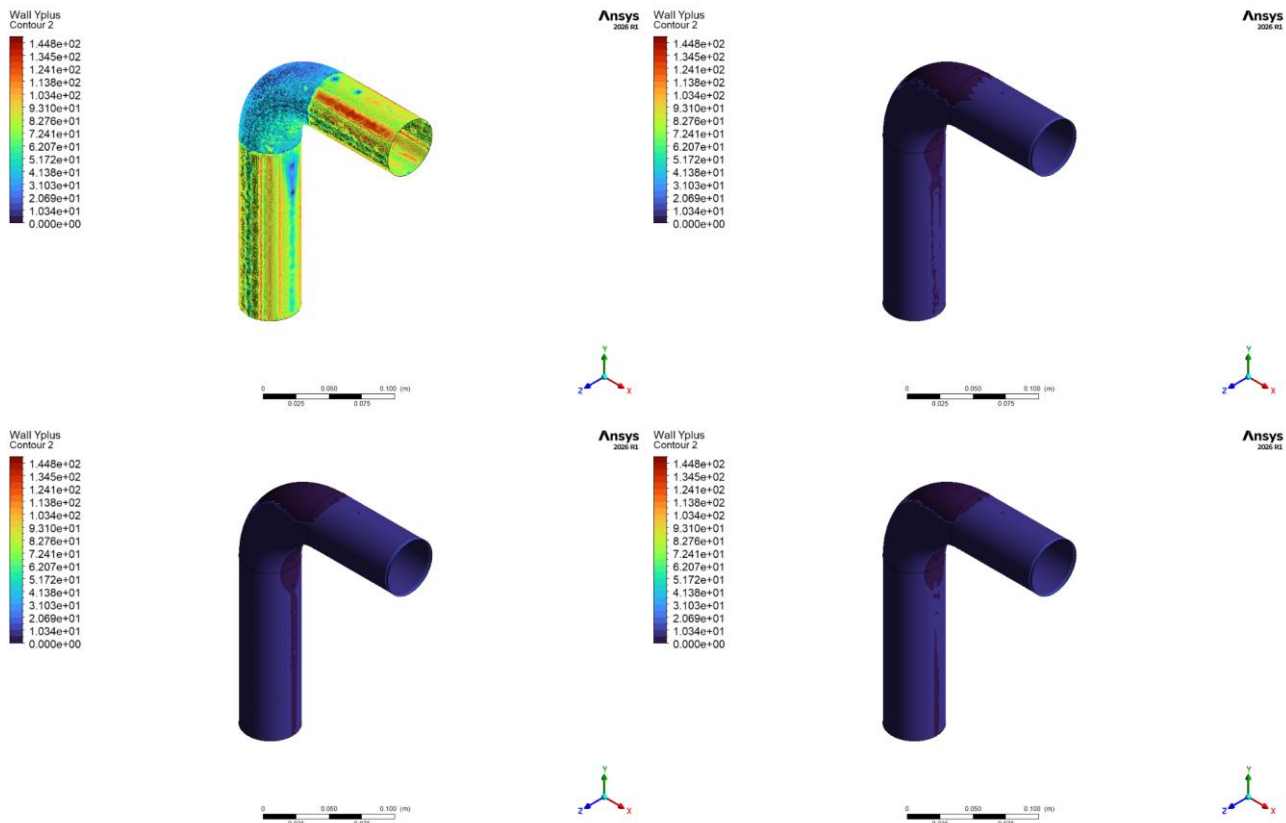


Figure 7. Wall y+ across the L0, L4, L8 and L12 near-wall meshes.

The common-scale wall-y+ contours confirm that introducing prism layers substantially improves near-wall resolution relative to L0; the expanded comparison shows similar L4-L12 spatial patterns because their first-layer height is shared.

Eight layers were selected as the lowest sufficiently resolved configuration. L12 did not materially change mass flow, the complete pressure distribution or the Tap 9–10 local pressure difference, whereas L4 retained a larger downstream difference. Figures 6 and 7 support this decision; the detailed L4/L8/L12 wall-y+ comparison is retained in Appendix C.

4.3.4 Turbulence-model sensitivity and model selection

Turbulence-model sensitivity was assessed at 20 inH₂O on the M3/L8 mesh using SST k- ω , Realizable k- ϵ and RNG k- ϵ . Boundary conditions, air properties, solver controls, tap definitions and convergence assessment were held constant.

Table 21. Turbulence-model sensitivity on the M3/L8 pipe mesh.

Case / model	Mass flow [kg s ⁻¹]	RMSE [Pa]	MAE [Pa]	Correlation r	Tap 9–10 Δp [Pa]
T1 SST k- ω	0.175001	746.7	548.3	0.9424	-975.0
T2 Realizable k- ϵ	0.177106	752.6	566.2	0.9526	-1008.5
T3 RNG k- ϵ	0.176586	765.8	581.1	0.9500	-1001.0

All three models reproduced the broad upper- and lower-wall pressure patterns and placed the minimum at Tap 12. SST gave the lowest complete-distribution RMSE and MAE. Realizable and RNG k-epsilon predicted Tap 9–10 lower-wall pressure differences of -1.008 and -1.001 kPa, compared with -0.975 kPa for SST and -1.559 kPa experimentally, but neither alternative removed the local discrepancy or improved the complete distribution consistently. Figure 8 provides the quantitative comparison, and Figure 9 shows the corresponding centre-plane velocity contours. Retaining SST is also consistent with its established use for adverse-pressure-gradient and separating flows (Menter, 1994; Menter, Kuntz and Langtry, 2003).

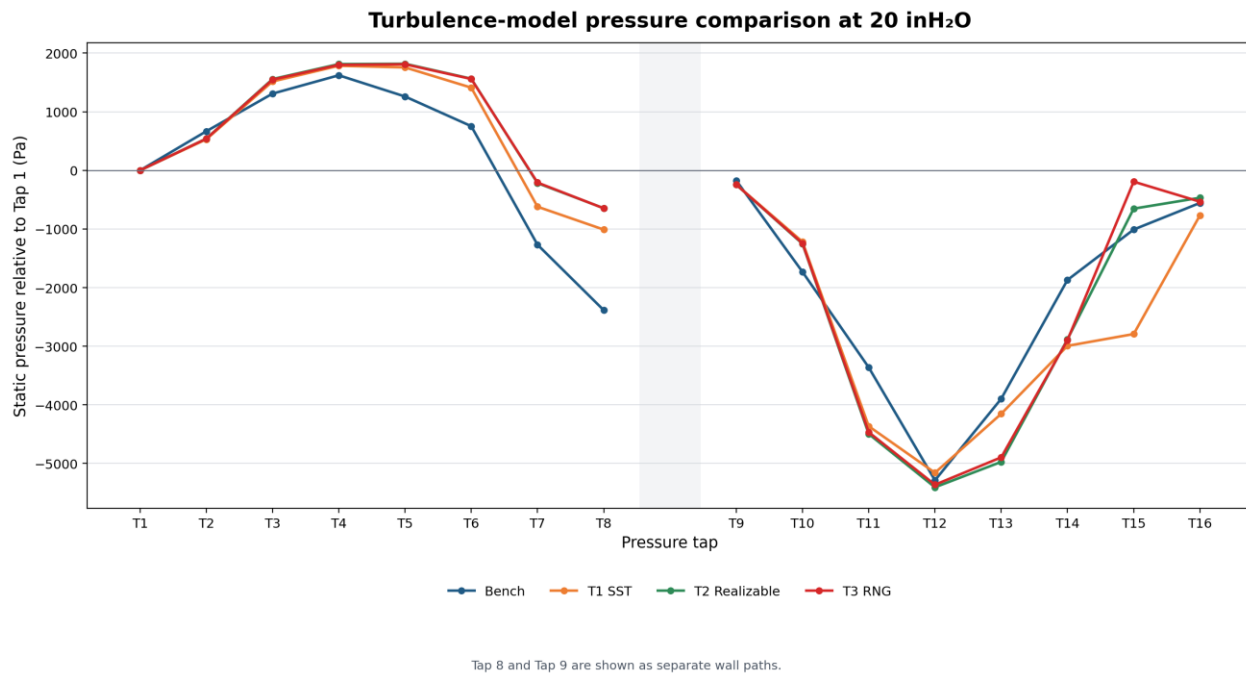


Figure 8. Turbulence-model comparison of bend wall-static-pressure at 20 inH₂O.

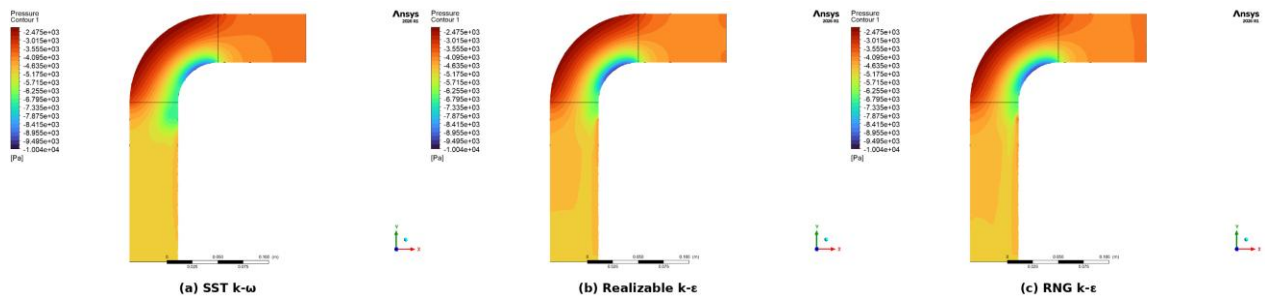


Figure 9. Centre-plane velocity contours by turbulence model at 20 inH₂O.

SST k- ω was retained for the pressure sweep and airbox simulations because it combined the lowest overall error, stable convergence and consistency with the external vehicle model.

4.3.5 Full pressure-sweep validation

The M3/L8/SST method was applied at 5-25 inH₂O using matched tap locations and pressure boundaries. Experimental and CFD curves were independently referenced to Tap 1, enabling comparison of distribution shape while not demonstrating absolute-pressure agreement or irreversible total-pressure loss.

Table 22. Full 5–25 inH₂O pipe-validation pressure sweep.

Bench setting	CFD mass flow [kg s ⁻¹]	RMSE [Pa]	MAE [Pa]	Correlation r	Bench Tap 9–10 Δp [Pa]	CFD Tap 9–10 Δp [Pa]
5 inH ₂ O	0.085804	234.4	167.0	0.8968	-220	-237.4
10 inH ₂ O	0.122443	427.2	269.0	0.9197	-540	-480.4
15 inH ₂ O	0.150866	735.3	477.9	0.8969	-926	-726.6
20 inH ₂ O	0.175001	746.7	548.3	0.9424	-1559	-975.0
25 inH ₂ O	0.197203	945.8	664.2	0.9430	-1248	-1235.6

Across all five conditions, CFD reproduced the measured upper- and lower-wall pressure patterns and placed the minimum at Tap 12 (Figure 10). Correlation remained between 0.897 and 0.943. Although absolute RMSE rose with pressure scale, normalised RMSE remained within 10.8–14.4% of the measured tap-pressure range, so the agreement did not systematically worsen as flow increased. Agreement is therefore judged for the specified wall-pressure response and operating range rather than as universal model validation (ASME, 2009).

Flow-bench and CFD pressure-distribution comparison across the test range

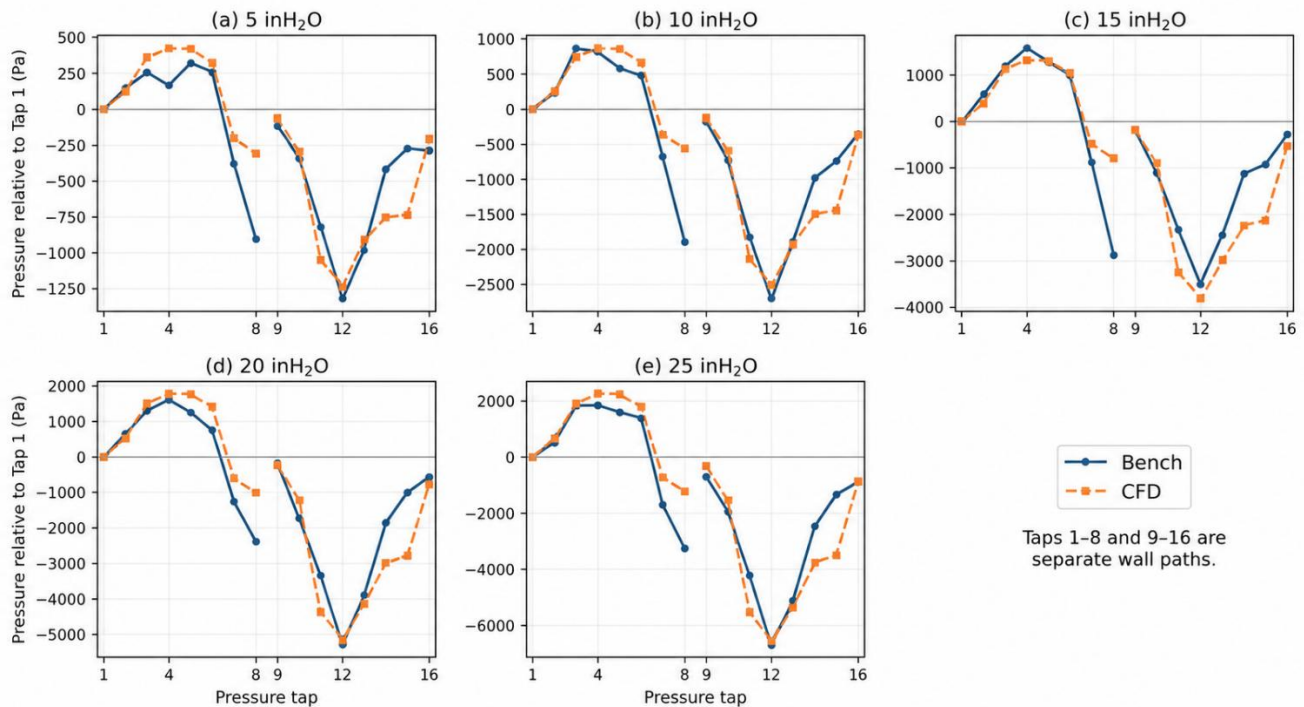


Figure 10. Bench-CFD bend wall-static-pressure comparison across 5–25 inH₂O.

The largest Tap 9-10 discrepancy occurred at 20 inH₂O rather than at the highest depression, indicating a local flow-structure mismatch rather than a simple scaling error. The sweep therefore supports M3/L8/SST for comparative steady airbox assessment, while the validation claim remains limited to the tested bend wall-pressure distribution and operating range.

4.4 Airbox bench and internal CFD correlation

4.4.1 Updated-airbox flow-bench pressure results

The S2N+ updated airbox was fitted with sixteen wall-static-pressure taps. Taps 1-5 sampled the upper inlet, Taps 6-12 the lower inlet/transition and Taps 13-16 the plenum above the runners; because Tap 6 is offset and Tap 9 lies below Tap 8, the middle group is interpreted regionally rather than as one continuous line. Each channel was referenced to its cylinder-set ambient reading. Three repeats were recorded for each cylinder at 20 inH₂O, 10 mm valve lift and fully open throttle, with one path active, all spark-plug holes sealed and no air filter. The nominal bench depression acted across the complete intake path, including the head, port and partially open valve, so it was not the pressure drop across the airbox alone.

Table 23. Updated-airbox flow-bench pressure results for Cylinders 1-4.

Cylinder	Repeated runs	Mean wall-static pressure [Pa]	Run-mean range [Pa]	Spatial SD [Pa]	Most negative tap	Highest-pressure tap
1	3	-43.94	4.06	15.97	T6 (-71.7 Pa)	T16 (-8.0 Pa)
2	3	-39.44	2.75	13.57	T1 (-68.0 Pa)	T16 (-5.3 Pa)
3	3	-32.56	2.13	10.65	T6 (-51.0 Pa)	T16 (-15.3 Pa)
4	3	-22.15	6.00	7.44	T6 (-35.7 Pa)	T5 (-10.3 Pa)

Cylinder 1 produced the strongest mean wall-pressure reduction, while Cylinder 4 produced the smallest. Cylinder 4 also showed the greatest repeat-to-repeat movement, although all three repeats still gave a clear pressure reduction below its ambient reference. The full cylinder-by-cylinder values are reported in Table 23.

4.4.2 Updated-airbox CFD-to-bench wall-pressure comparison

The area-weighted CFD wall-static pressures were compared with the experimental mean at all sixteen taps. The CFD pressures were generally more negative than the bench measurements. Cylinder 1 gave the lowest magnitude error, Cylinder 3 gave the strongest spatial correlation, and Cylinder 4 gave the largest magnitude error. Correlation across the four cylinder cases ranged from 0.742 to 0.866. The consistently negative mean errors show that the principal disagreement is a systematic suction-magnitude bias rather than random scatter alone; the full error metrics are reported in Table 24.

Table 24. Updated-airbox CFD-to-bench wall-pressure error metrics.

Cylinder	MAE [Pa]	RMSE [Pa]	Mean error, CFD - bench [Pa]	Pearson r
1	51.0	61.1	-51.0	0.844
2	58.5	66.2	-58.5	0.742
3	70.3	76.6	-70.3	0.866
4	87.7	92.1	-87.7	0.742

Negative mean error indicates that CFD predicted a more negative wall pressure than the bench measurement.

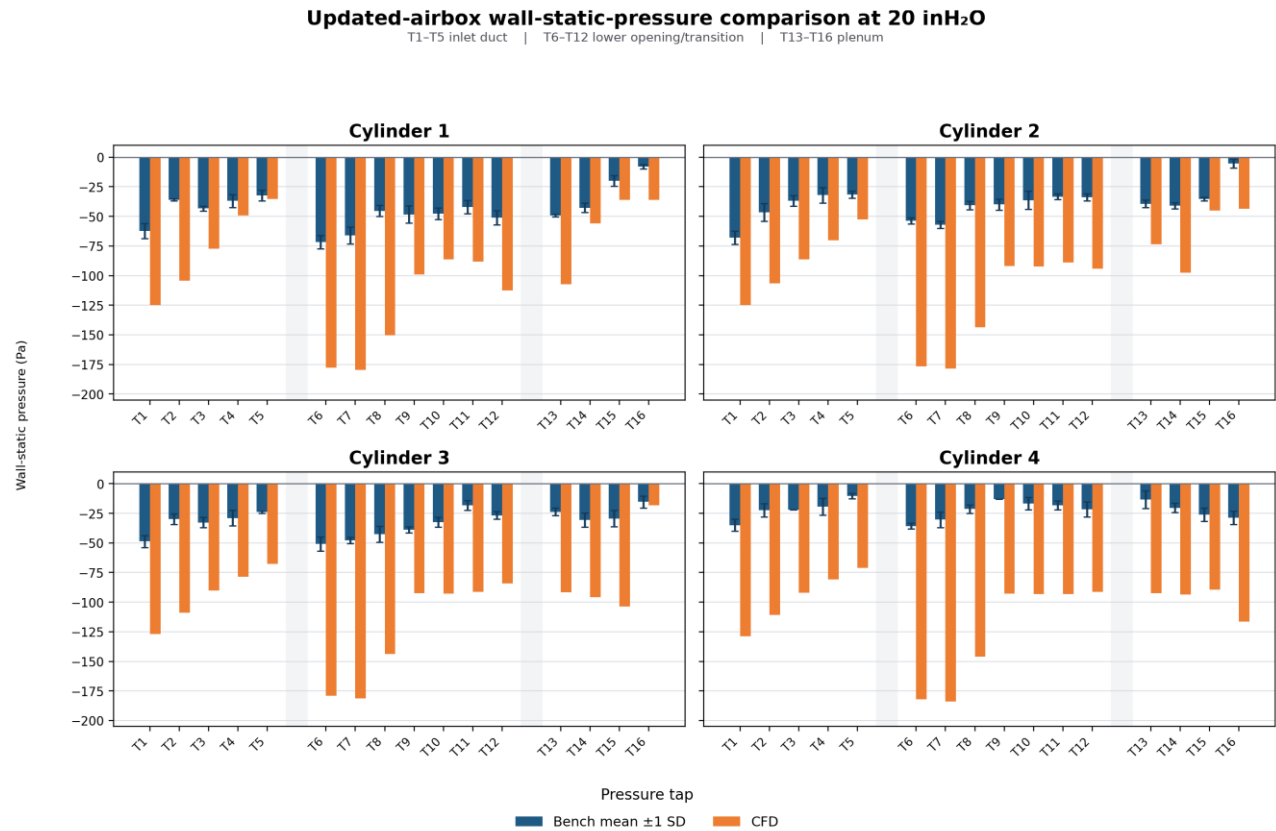


Figure 11. Updated-airbox bench–CFD wall-static-pressure comparison for C1–C4 at 20 inH₂O.

Regional pressure trend averaged across Cylinders 1-4

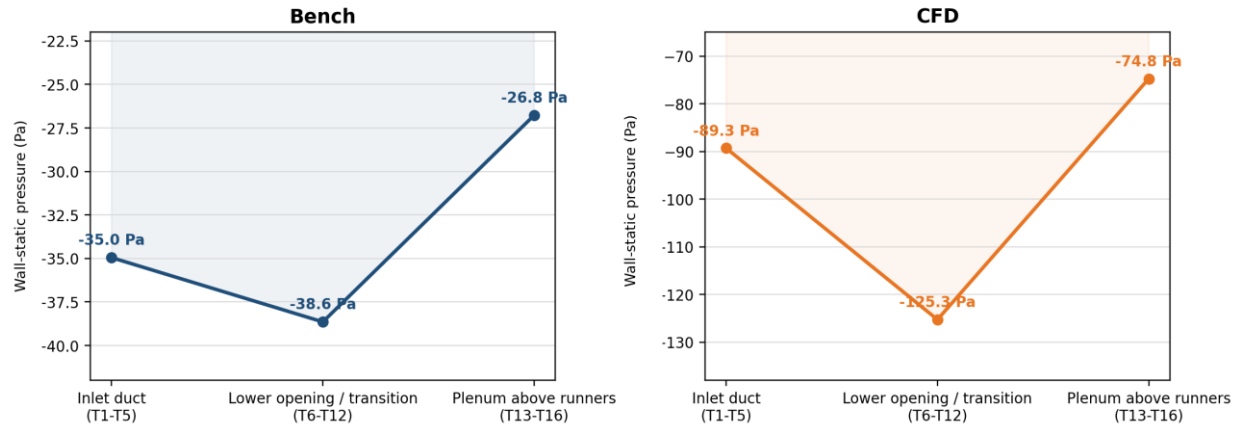


Figure 12. Bench-CFD regional wall-static-pressure trends in the updated airbox.

The updated-airbox tap comparison gave additional support for using the pipe-tested CFD method on the airbox. CFD reproduced the broad wall-static-pressure pattern, including the stronger reduction through the lower inlet transition and the later rise in the plenum, with positive spatial correlations of $r = 0.742$ - 0.866 . The same SST k-omega model, coupled solver, second-order discretisation and near-wall treatment were therefore retained for D0 and the design cases. Because suction magnitude was consistently overpredicted, the tapped model is used for spatial correlation only; clean D0 is the design baseline. The remaining absolute bias is consistent with published intake studies showing that downstream port, valve and cylinder representation can materially affect agreement with experiment (Masi, Artico and Gobbato, 2015; Nanni et al., 2008).

Supporting tabulations for the matched airbox methodology, the tapped-versus-clean model-preparation check and the final design evidence are provided in Appendix E. The clean D0 rerun is introduced next so that the design concepts are compared against a reference model prepared in the same manner.

4.5 Pressure-tapped versus clean-model sensitivity and D0 definition

The pressure-tapped updated-airbox model was required to reproduce the sixteen experimental sampling locations, but the tap split faces and associated local refinement were not required for the subsequent design study. A clean rerun was therefore completed with the tap split faces omitted and the lower opening shifted approximately 0.5 mm rearward. Since both model-preparation changes were made together, the comparison cannot isolate a pure pressure-tap effect; it is reported as a sensitivity check between the correlation model and the clean design-study model.

Table 25. Pressure-tapped versus clean D0 model-preparation sensitivity.

Metric	Tapped correlation model	Clean D0	Change
Mean mass flow [kg s^{-1}]	0.161575	0.161706	+0.08%
Mean outlet velocity [m s^{-1}]	85.25	85.43	+0.20%
Mean total-pressure loss [Pa]	457.2	436.8	-4.48%
Mass-flow CV [%]	0.67	0.45	-0.22 percentage points
Total-pressure-loss CV [%]	8.85	6.01	-2.84 percentage points

Mean mass flow changed by only +0.08%, and the individual cylinder mass-flow changes remained within approximately -0.20% to +0.42%. This indicates that the overall flow-delivery level was essentially unchanged by the clean model preparation. Total-pressure loss was more sensitive: the mean decreased by 4.48%, with the largest cylinder-specific change occurring for C1. The clean model is therefore adopted as D0 so that D1, D2 and D3 are ranked against a reference prepared without the correlation-only tap faces.

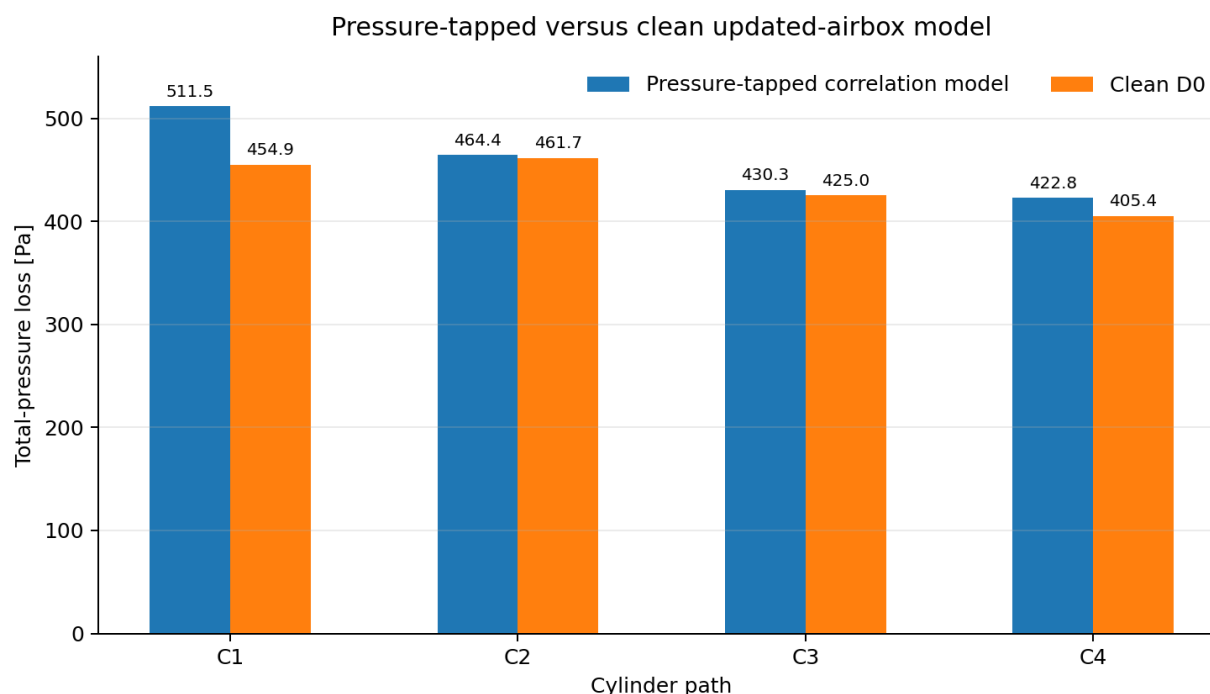


Figure 13. Pressure-tapped correlation model versus clean D0 total-pressure loss for the four steady cylinder paths.

The sensitivity check also changes the interpretation of the design stage. The earlier tapped-model loss of 457.2 Pa is retained for CFD-to-bench correlation, whereas clean D0, with a mean loss of 436.8 Pa, is the only reference used for D1, D2 and D3 percentage changes. This separation prevents a correlation-specific CAD/mesh treatment from artificially inflating the apparent benefit of a design concept.

4.6 Old-versus-updated airbox comparison and baseline selection

The old-airbox Fluent exports were generated using the same 0 Pa inlet and -4982 Pa active-outlet boundary definition. The pressure-tapped S2N+ updated-airbox outputs are retained here as supporting matched cases linked to the 16-tap correlation model. They are not used as the D0 reference for the subsequent design ranking.

Table 26. Supporting pressure-tapped S2N+ updated-airbox and old-airbox CFD output summary.

Case	Mass flow [kg s ⁻¹]	Inlet total pressure [Pa]	Outlet total pressure [Pa]	Total-pressure loss [Pa]	Mass imbalance [%]	Comment
Tap-C1	0.160784	0.000	-511.5	511.5	0.023	Tap r=0.844; RMSE=61.1 Pa; tapped correlation model
Tap-C2	0.160695	0.000	-464.4	464.4	0.107	Tap r=0.742; RMSE=66.2 Pa; tapped correlation model
Tap-C3	0.161800	0.000	-430.3	430.3	0.064	Tap r=0.866; RMSE=76.6 Pa; tapped correlation model
Tap-C4	0.163021	0.000	-422.8	422.8	0.030	Tap r=0.742; RMSE=92.1 Pa; tapped correlation model
Old-C1	0.155772	0.000	-729.0	729.0	0.032	Controlled old-airbox case
Old-C2	0.154428	0.000	-782.2	782.2	0.041	Controlled old-airbox case
Old-C3	0.158629	0.000	-571.1	571.1	0.003	Controlled old-airbox case
Old-C4	0.154796	0.000	-700.8	700.8	0.038	Controlled old-airbox case

Under the matched boundaries, clean D0 increased mean outlet mass flow from 0.15591 to 0.16171 kg s⁻¹ (+3.72%) and improved every cylinder path relative to the old airbox.

Mean total-pressure loss decreased from 695.8 to 436.8 Pa (-37.2%). The largest reductions occurred on C1, C2 and C4, which were the more strongly penalised old-airbox paths.

Path consistency also improved: mass-flow CV fell from 1.22% to 0.45% and loss CV from 12.89% to 6.01%. These values describe four separate steady paths, not simultaneous cylinder filling. Together with the mean-loss reduction, they justify retaining clean D0 as the quantitative design baseline.

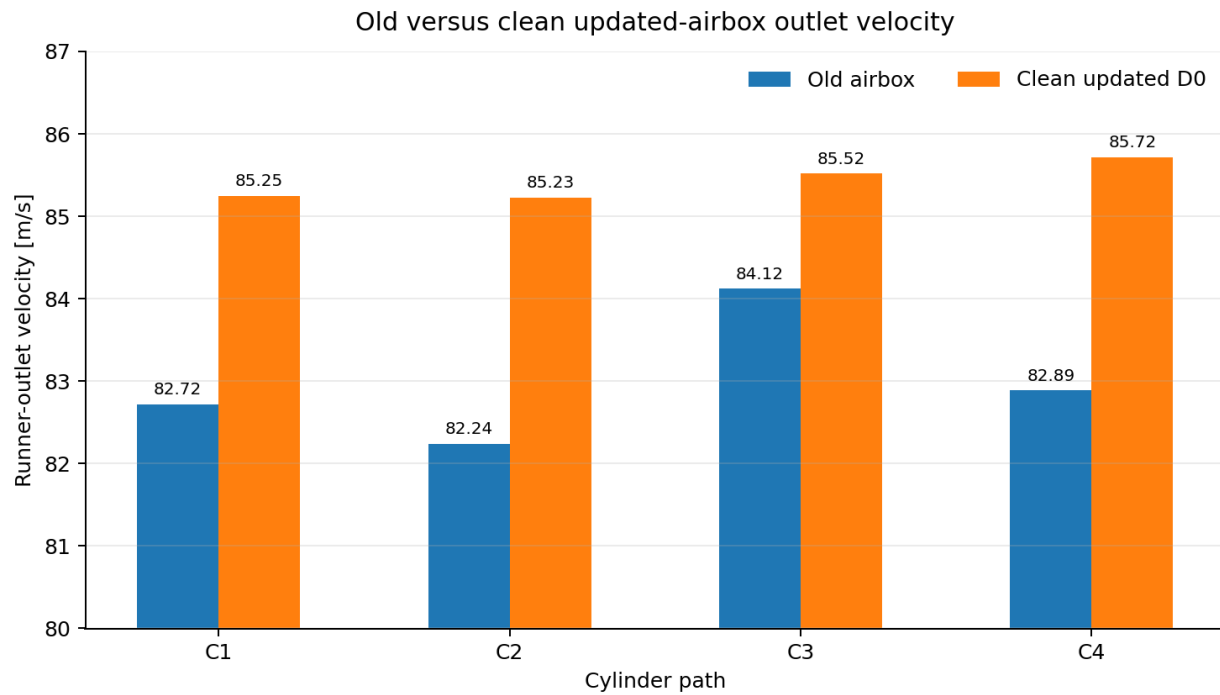
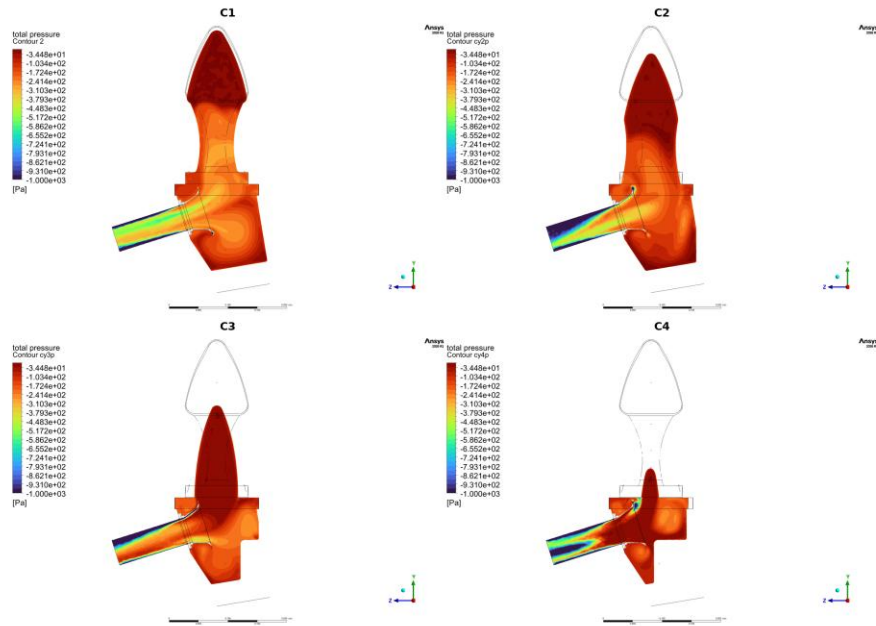


Figure 14. Old-airbox versus clean D0 runner-outlet velocity for the four matched steady cylinder paths.

Table 27. Old-airbox versus clean S2N+ updated-airbox (D0) performance comparison.

Cylinder	Clean D0 mass flow [kg s ⁻¹]	Old-airbox mass flow [kg s ⁻¹]	Mass-flow change [%]	D0 total-pressure loss [Pa]	Old-airbox total-pressure loss [Pa]	Loss change [%]	Assessment
1	0.161456	0.155772	+3.65	454.9	729.0	-37.6	Clean D0: higher flow and lower loss
2	0.160962	0.154428	+4.23	461.7	782.2	-41.0	Clean D0: higher flow and lower loss
3	0.161715	0.158629	+1.95	425.0	571.1	-25.6	Clean D0: higher flow and lower loss
4	0.162691	0.154796	+5.10	405.4	700.8	-42.1	Clean D0: higher flow and lower loss

(a) Original airbox: runner-centred sections



(b) Updated airbox: common plenum/runner section

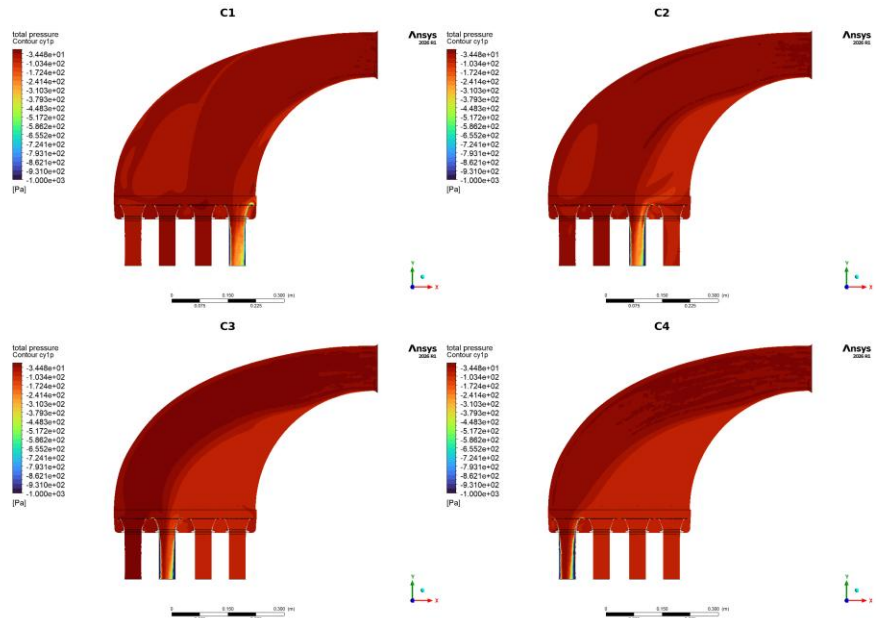


Figure 15(a). Total-pressure fields for the original and pressure-tapped S2N+ updated airboxes under the matched 0 Pa inlet and -4982 Pa active-outlet condition.

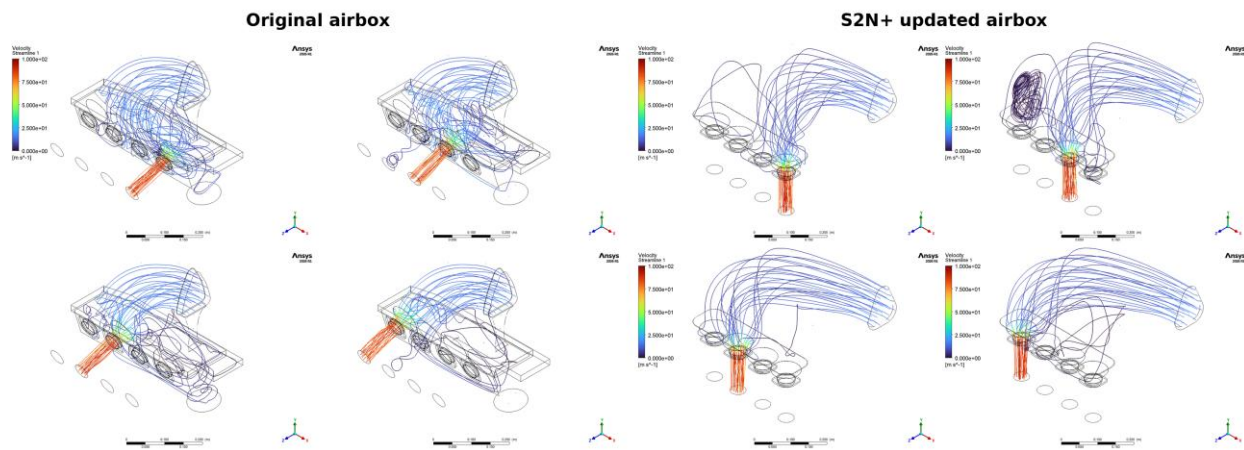


Figure 15(b). Inlet-seeded streamline comparison for the original and S2N+ updated airboxes across C1–C4 at the controlled 0 Pa inlet condition.

The flow fields in Figure 15(a) and the streamline comparison in Figure 15(b) help explain why the numerical improvement is much larger in total-pressure loss than in outlet velocity. The original airbox shows more extensive cross-plenum turning and looping before the flow reaches the active runner, together with larger low-total-pressure regions. In the updated geometry, the main flow paths are generally more direct and the strongest total-pressure reduction is concentrated closer to the trumpet and runner approach. This agrees with the lower mean loss and smaller path-to-path loss spread reported above.

The section locations and streamline seeding are not identical between the two airboxes, so Figure 15 is used only to compare the general flow mechanism. The updated field in Figure 15(a) is from the pressure-tapped model, whereas the quantitative old-versus-updated percentages use clean D0. The small mass-flow change between the tapped and clean preparations supports this qualitative use, while the 4.48% loss change is why clean D0 remains the numerical baseline.

4.7 External inlet-condition assessment

Table 28. Whole-vehicle external inlet-condition comparison.

External configuration	Inlet static pressure [Pa]	Inlet total pressure [Pa]	Drag coefficient, C_D	Lift coefficient, C_L	Inlet total-pressure change vs old [%]	Role in study
Old airbox	600.3	792.6	0.4816	-1.4953	Reference	External pressure-supply reference
Updated airbox	569.2	728.9	0.4805	-1.4328	-8.04	Vehicle-derived internal sensitivity condition

The matched whole-vehicle cases at 35 m/s estimated the local static and total pressure at the two airbox openings in the external model. Their force coefficients were used only as a vehicle-level reasonableness check, not as validation of the local inlet field. Because no engine through-flow was represented, these pressures are not treated as measured or fully coupled pressure supply to the running engine.

The old-airbox inlet total pressure was 792.6 Pa and the updated-airbox value was 728.9 Pa, an 8.04% lower value for the updated installation, while drag coefficient changed by only -0.24% (Figure 16). This does not show that the updated airbox has poorer internal performance: the external model contains no engine through-flow and the two inlet geometries sample different local vehicle pressure fields. The result instead shows that external pressure capture and internal airbox resistance are separate effects. Using configuration-specific inlet pressures in the design comparison would combine those effects, so the controlled internal study retains a common inlet boundary.

The original airbox also had a larger inlet-face area than the S2N+ updated opening. This larger surface sampled more of the local high-pressure region around the intake and is likely to have contributed to the higher area-averaged static and total pressures reported for the original geometry. Opening area alone does not determine the result, because the position, shape and orientation of the inlet also affect the local pressure field.

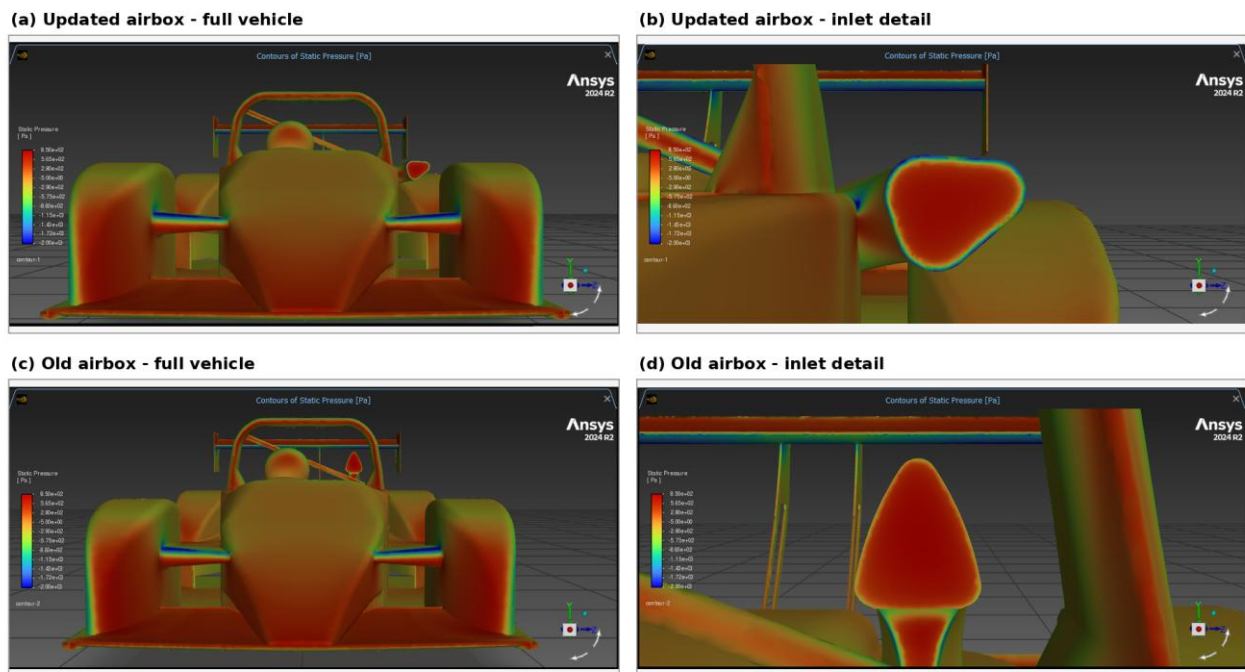


Figure 16. Common-scale external static-pressure contours for the updated and old airbox installations: (a, c) full-vehicle views and (b, d) inlet details.

4.7.1 Vehicle-derived inlet-pressure sensitivity

Clean D0 was repeated for Cylinders 1-4 using a uniform 728.92 Pa inlet total gauge pressure and the same -4982 Pa active outlet. This was a boundary-condition sensitivity case rather than an engine operating point: 728.92 Pa was taken from the external-model inlet face, while the external model contained no engine through-flow. Relative to the controlled 0 Pa cases, mean outlet velocity increased from 85.43 to 91.63 m/s (+7.26%) and mean mass flow from 0.16171 to 0.17333 kg s⁻¹ (+7.19%). The increase was similar for all four paths, so the higher imposed inlet pressure raised the overall CFD flow level without changing the basic cylinder-path ordering (Figure 17).

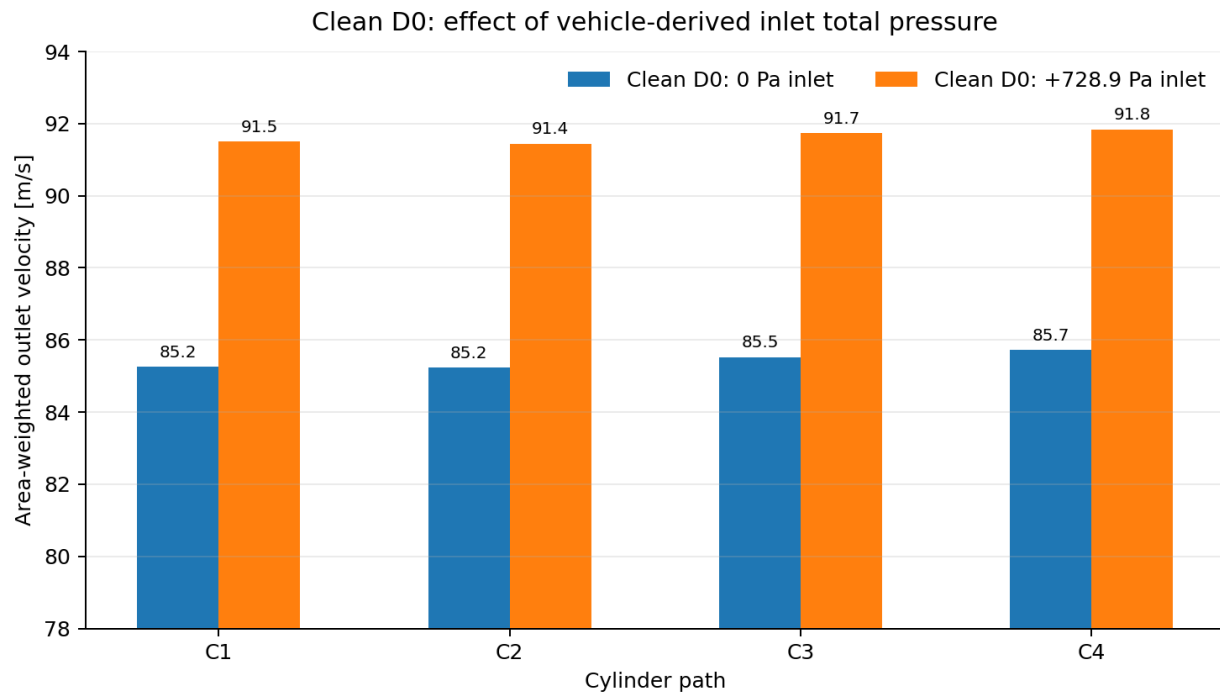


Figure 17. Clean D0 runner-outlet velocity comparison between the controlled 0 Pa inlet and the uniform +728.9 Pa vehicle-derived inlet total-pressure sensitivity condition.

Table 29. Clean D0 sensitivity to the +728.9 Pa vehicle-derived inlet condition.

Metric	0 Pa controlled inlet	+728.9 Pa uniform inlet	Change
Mean outlet velocity, C1-C4	85.43 m s ⁻¹	91.63 m s ⁻¹	+7.26%
Mean outlet mass flow, C1-C4	0.16171 kg s ⁻¹	0.17333 kg s ⁻¹	+7.19%
Mean internal total-pressure loss	436.8 Pa	482.1 Pa	+10.38%*

Mean total-pressure loss increased from 436.8 to 482.1 Pa (+10.38%) as the flow level increased. Because the two conditions are not matched-flow cases, this larger absolute loss is not interpreted as deterioration of D0. A rise in loss is expected when the same passages carry more flow because turning, mixing and wall-shear losses increase with velocity. The loss ordering remained C2, C1, C3 and C4 from highest to lowest, matching the controlled 0 Pa result, while mass-flow CV remained essentially unchanged at 0.45-0.46%. All four cases also retained mass imbalance below approximately 0.04%.

Total-pressure-loss CV decreased from 6.01% to 4.72% at +728.92 Pa, while the path ordering remained unchanged. The higher inlet pressure therefore increased through-flow without introducing a clearly different steady flow pattern. Supporting fields in Appendix G, Figure G20, show the same broad plenum circulation. This supports use of the 0 Pa condition for controlled geometry ranking while showing that D0 behaves similarly at the higher imposed inlet pressure.

4.8 D0 baseline flow-mechanism diagnosis

With the old-versus-updated comparison established in Section 4.6, the remaining purpose of the baseline diagnosis was to identify where clean D0 still lost pressure and why some cylinder paths were less favourable than others. The direct old-versus-updated comparison in Figure 15 provides the overall context, while Figure 18 shows the clean D0 streamlines at a larger scale for the four controlled 0 Pa cases. The grouped S2N+ baseline supporting fields are retained in Appendix G, Figure G1, for additional qualitative detail.

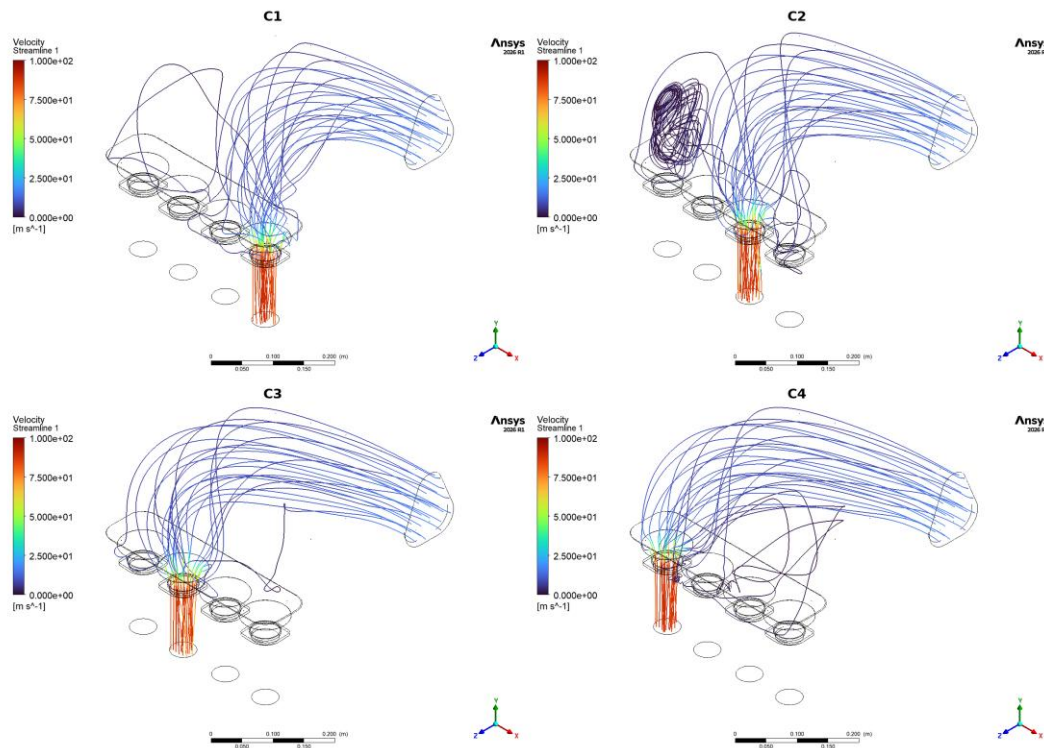


Figure 18. Clean D0 inlet-seeded streamline fields for Cylinders 1-4 at the controlled 0 Pa inlet condition.

The D0 streamlines remain strongly three-dimensional. C1 and C2 show longer cross-plenum loops and less direct runner-feeding paths than C3 and C4, particularly around the inactive trumpet region. This agrees with the quantitative loss ranking: C2 and C1 have the highest D0 losses at 461.7 and 454.9 Pa, compared with 425.0 Pa for C3 and 405.4 Pa for C4. The visual evidence therefore supports the measured path ranking rather than introducing a separate performance measure.

Mass-flow variation remains small (CV 0.45%) despite the larger 6.01% spread in total-pressure loss. This indicates that the main remaining difference between D0 paths is pressure preservation through the turning and runner-approach region rather than an inability to pass similar bulk flow under the imposed pressure boundary. The longer C1/C2 trajectories and the detached inner inlet-transition region therefore became the main targets for the first geometry changes, leading directly to the D1-D3 wall-profile study.

4.9 Wall-profile study: D1-D3

Designs D1-D3 formed the wall-profile study and were assessed against clean D0 using the same 0 Pa inlet condition, -4982 Pa active-cylinder outlet, SST k- ω model, report definitions and matched clean-model meshing controls. D1 modified the diagnosed inlet-to-plenum inner wall, while D2 and D3 tested alternative profiles on the lower half of the lower inlet wall. The family was judged primarily by total-pressure loss and path consistency because the changes in mass flow were small.

Figure 19 gives the quantitative comparison. D1 was the only wall-profile concept to reduce total-pressure loss on all four paths. Mean loss fell from 436.8 Pa for D0 to 429.7 Pa for D1 (-1.61%), although C2 remained the worst path and improved by only 0.51%. Mean mass flow increased by 0.09%, mass-flow CV changed from 0.45% to 0.41%, and loss CV increased slightly from 6.01% to 6.35%. The result therefore indicates a small improvement in pressure preservation rather than a major change in flow capacity.

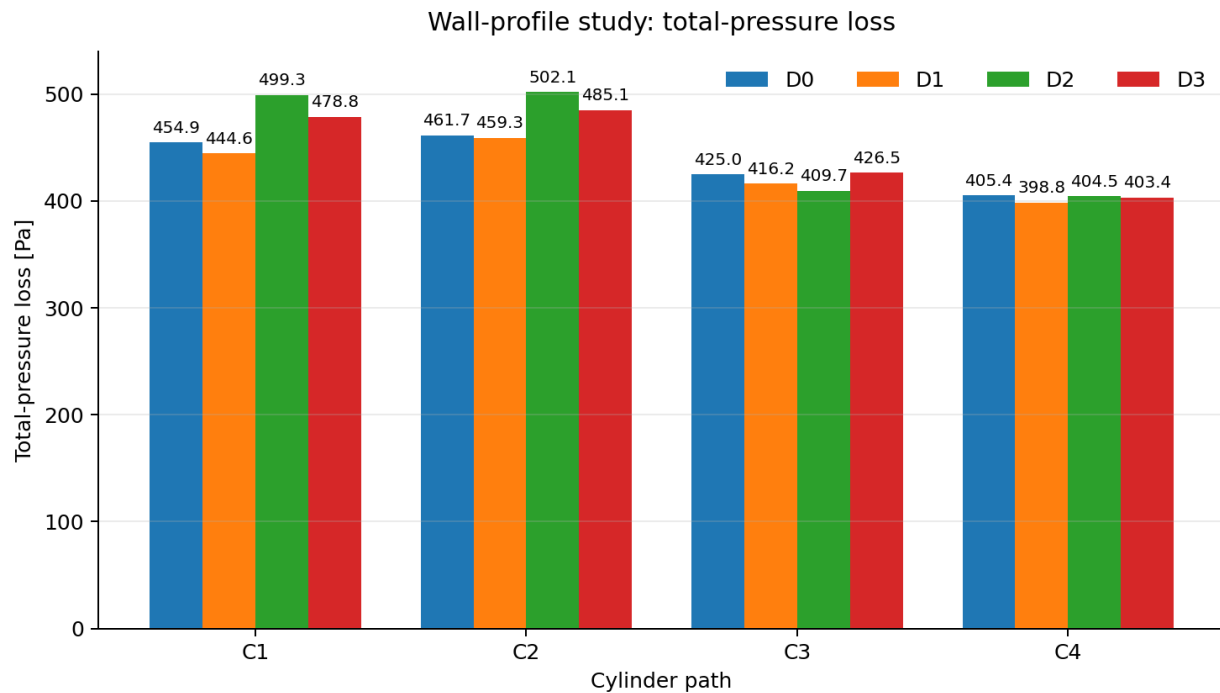


Figure 19. Clean D0, D1, D2 and D3 total-pressure loss for the four matched steady cylinder paths.

D2 produced the opposite response. Mean mass flow increased by only 0.24%, while mean total-pressure loss rose to 453.9 Pa (+3.93%) and the worst-path loss rose to 502.1 Pa (+8.75%). Loss CV increased to 11.91%. The change therefore shifted the flow distribution towards C3/C4 while increasing the penalty on C1/C2, rather than giving a uniform efficiency gain.

D3 reduced the severity of the D2 penalty but did not reverse it. Mean mass flow changed by only +0.29% versus D0, while mean loss increased by 2.68% and the worst-path loss by 5.06%. The same C1/C2 penalty direction seen for D2 remained, so similar outlet velocities did not translate into similar pressure preservation.

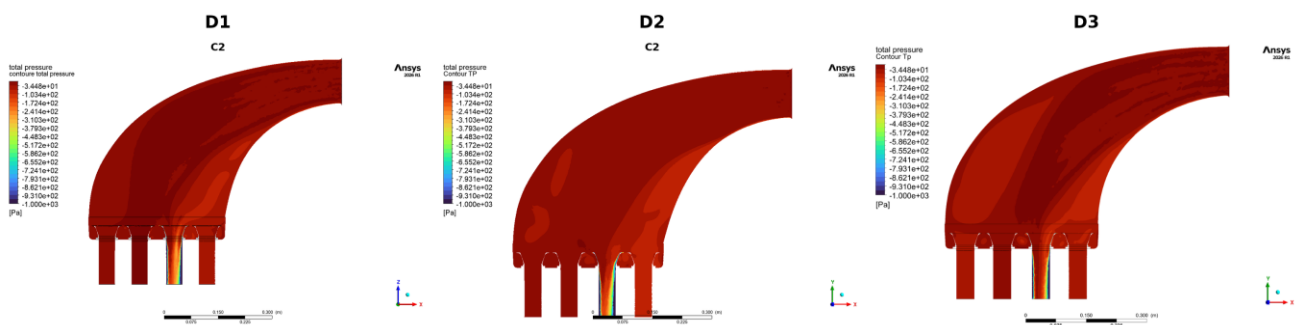


Figure 20. Cylinder 2 total-pressure contours for D1, D2 and D3 under the controlled 0 Pa inlet condition. All three panels use the same -1000 to -34.48 Pa contour scale.

The matched C2 contours in Figure 20 support the numerical ranking. D1 retains the same broad lower-plenum loss region but without the larger penalty produced by the two lower-wall profiles. D2 shows the strongest deterioration around the lower transition and active-runner approach, while D3 changes the distribution but still leaves C2 worse than D0. The comparison also shows why streamline attachment or local velocity magnitude alone is not enough for selection: the important difference is how much total pressure is preserved along the complete path to the active runner.

D1 was the best wall-profile concept, but its 1.61% mean-loss reduction is smaller than the 4.48% model-preparation sensitivity in Section 4.5, so it is reported as a small favourable trend rather than a confirmed gain. D2 and D3 are rejected because both increase mean and worst-path loss and worsen path consistency. Clean D0 remains the baseline for the following inlet-shape study.

4.10 Inlet-shape study

D4 used a round inlet and D5 a capsule-shaped inlet, each within 0.4% of D0 area and blended to unchanged downstream geometry. Both omit the D0 lip, so lipless D6 is the direct shape reference; D0 is retained only for project context.

Table 30. Inlet-shape study performance comparison for D0, D6, D4 and D5.

Metric	Clean D0	D6 lipless reference	D4	D5
Mean outlet mass flow [kg s ⁻¹]	0.161706	0.162178	0.162419 (+0.15% vs D6)	0.162697 (+0.32% vs D6)
Mean outlet velocity [m s ⁻¹]	85.43	85.49	85.59	85.52
Mean total-pressure loss [Pa]	436.8	432.6	423.2 (-2.17% vs D6)	427.5 (-1.16% vs D6)
Worst total-pressure loss [Pa]	461.7 (C2)	459.6 (C1)	473.9 (C1, +3.12% vs D6)	471.3 (C1, +2.55% vs D6)
Loss CV [%]	6.01	6.99	8.45	7.38

Relative to D6, D4 reduced mean loss by 2.17% and D5 by 1.16%, but C1 loss increased by 3.12% and 2.55%. The mean improvement therefore came from lower losses on the other paths rather than a uniform reduction in resistance. Figure 21 shows the path-by-path loss response; the corresponding mass-flow changes were small and are retained in Appendix G.

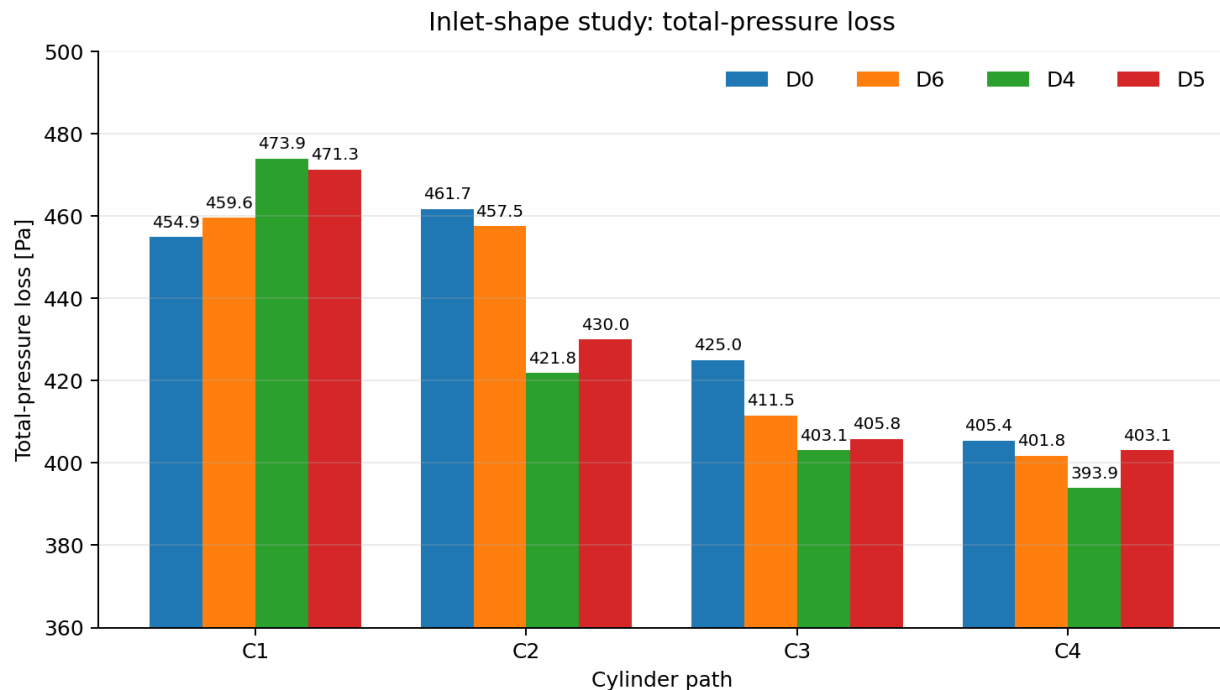


Figure 21. Clean D0, lipless D6, D4 and D5 total-pressure loss for the four matched steady cylinder paths.

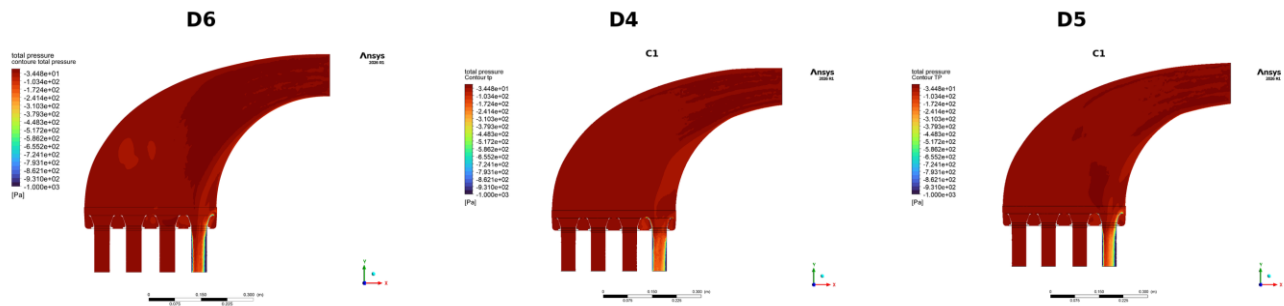


Figure 22. Matched C1 total-pressure contours for D6, D4 and D5 at the controlled 0 Pa inlet condition. All three panels use the same -1000 to -34.48 Pa total-pressure scale.

4.10.1 Design 4 assessment

D4 reduced mean loss by 2.17% relative to D6, but C1 loss increased by 3.12% and loss CV rose from 6.99% to 8.45%. Figure 22 shows that the low-total-pressure region feeding C1 remains while the other paths improve. The result is a redistribution of loss rather than a uniform reduction, so D4 was not selected. Supporting fields are grouped in Appendix G, Figure G5.

4.10.2 Design 5 assessment and inlet-shape decision

D5 reduced mean loss by 1.16% relative to D6 but increased C1 loss by 2.55% and loss CV to 7.38%. Figure 22 shows a similar C1 total-pressure pattern to D4: C2/C3 improve most while C1 remains limiting. The change is small and does not remove the path imbalance, so D5 was not selected. Supporting fields are grouped in Appendix G, Figure G6.

D4 and D5 therefore give the same overall outcome: a small reduction in mean loss with a penalty on C1. Changing inlet shape alone did not solve the path imbalance, so D6 was retained as the lipless reference for the inlet-size study.

4.11 Inlet-size study

Inlet size was isolated using the D0 circular form. Because its lip could not be scaled consistently, D6 removes the lip, D7 offsets the opening outward by 5 mm and D8 by 10 mm; downstream geometry and controlled boundary conditions are unchanged.

Table 31. Inlet-size study performance comparison for D0 and D6-D8.

Metric	Clean D0	D6 lipless reference	D7 +5 mm	D8 +10 mm
Mean outlet mass flow [kg s ⁻¹]	0.161706	0.162178 (+0.29% vs D0)	0.162555 (+0.23% vs D6)	0.162610 (+0.27% vs D6)
Mean outlet velocity [m s ⁻¹]	85.43	85.49 (+0.07% vs D0)	85.51 (+0.02% vs D6)	85.59 (+0.12% vs D6)
Mean total-pressure loss [Pa]	436.8	432.6 (-0.96% vs D0)	431.2 (-0.31% vs D6)	421.4 (-2.59% vs D6; -3.53% vs D0)
Worst total-pressure loss [Pa]	461.7 (C2)	459.6 (C1)	459.2 (C2)	458.6 (C2)
Loss CV [%]	6.01	6.99	6.93	7.42

D6 changed the mean result only slightly relative to D0 and did not improve every path, so it is used as the lipless geometric reference rather than as a selected design. Figures 23 and 24 focus on the pressure-loss response; mass-flow changes are retained in Appendix G.

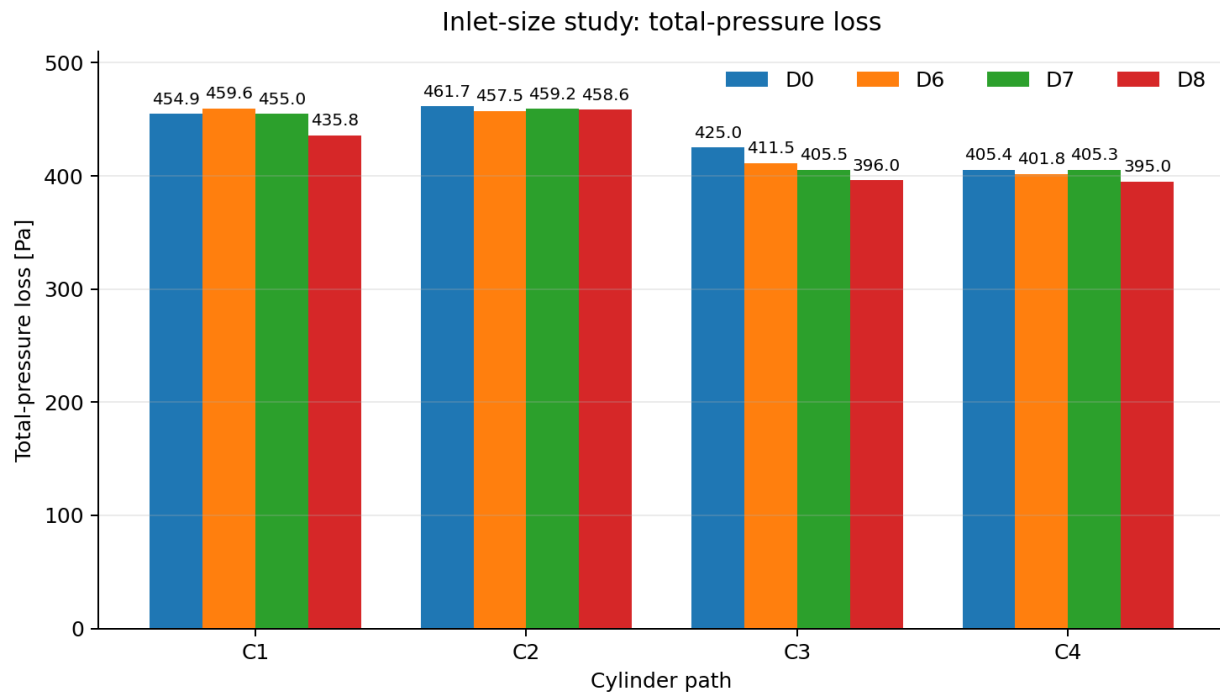


Figure 23. Clean D0, D6, D7 and D8 total-pressure loss for the four matched steady cylinder paths.

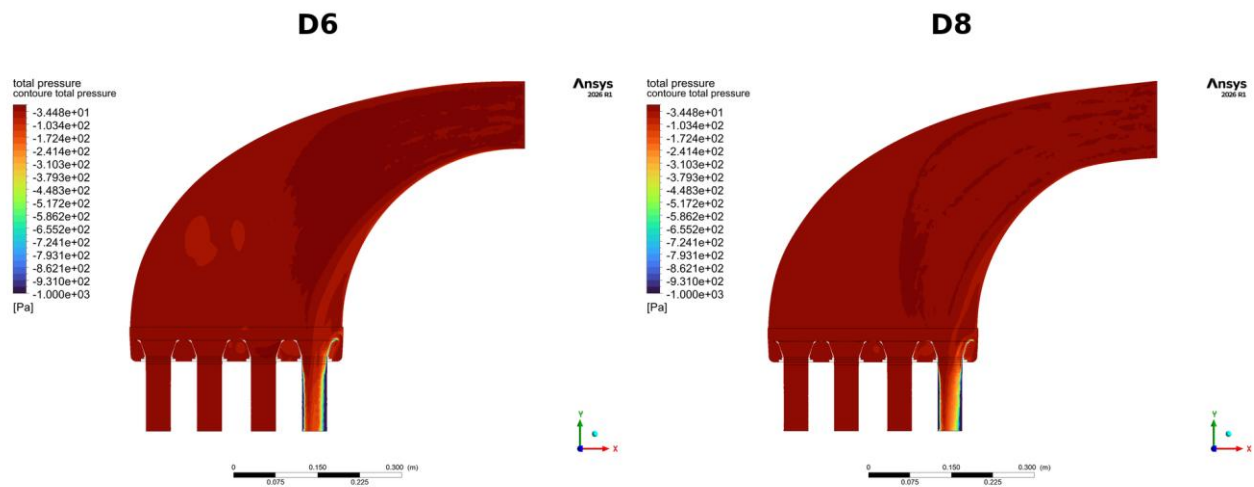


Figure 24. Matched C1 total-pressure contours for D6 and D8 at the controlled 0 Pa inlet condition. Both panels use the same -1000 to -34.48 Pa total-pressure scale.

Relative to D6, D7 changed mean loss by only -0.31%. D8 gave the clearer response, reducing mean loss by 2.59%, mainly through C1 and C3, while C2 changed little. Figure 24 provides the matched C1 total-pressure comparison.

4.11.1 D6 lip-removal reference

D6 mean loss was 432.6 Pa and its worst path shifted to C1. This confirms its role as the lipless size-study reference rather than an aerodynamic improvement.

4.11.2 D7/D8 size comparison and inlet-size decision

D7 produced no useful additional size benefit and slightly penalised C2/C4, so it was not taken forward.

D8 reduced mean loss by 2.59% relative to D6 and 3.53% relative to D0. All four D8 path losses remained below D0, although the worst-path improvement versus D0 was only 0.67% and loss CV rose to 7.42%. It was the best inlet-size case but was not preferred over D9.

4.12 Inlet-transition-length study

Transition length was isolated using D6 rather than enlarged D8. D9 shortened and D10 lengthened the lipless transition by 50 mm. D11 then shortened the D9 transition by a further 25 mm, giving a total reduction of 75 mm relative to D6. Downstream geometry and boundary conditions were unchanged.

Table 32. Inlet-transition-length study performance comparison for D6, D9, D10 and D11.

Metric	D6 reference	D9 -50 mm	D10 +50 mm	D11 -75 mm
Mean outlet mass flow [kg s ⁻¹]	0.162178	0.162633 (+0.28%)	0.161621 (-0.34%)	0.162235 (+0.04%)
Mean total-pressure loss [Pa]	432.6	414.7 (-4.10%)	432.6 (+0.01%)	427.6 (-1.15%)
Worst total-pressure loss [Pa]	459.6 (C1)	444.4 (C1, -3.31%)	481.7 (C1, +4.80%)	454.0 (C2, -1.21%)
Loss CV [%]	6.99	5.84	10.63	5.36

D9 improved all four paths relative to D6. Mean total-pressure loss fell by 4.10%, worst-path loss by 3.31% and loss CV from 6.99% to 5.84%, while mean mass flow changed by only +0.28%. Mean loss was also 5.05% below D0. The selection is therefore based on pressure preservation rather than flow-rate change.

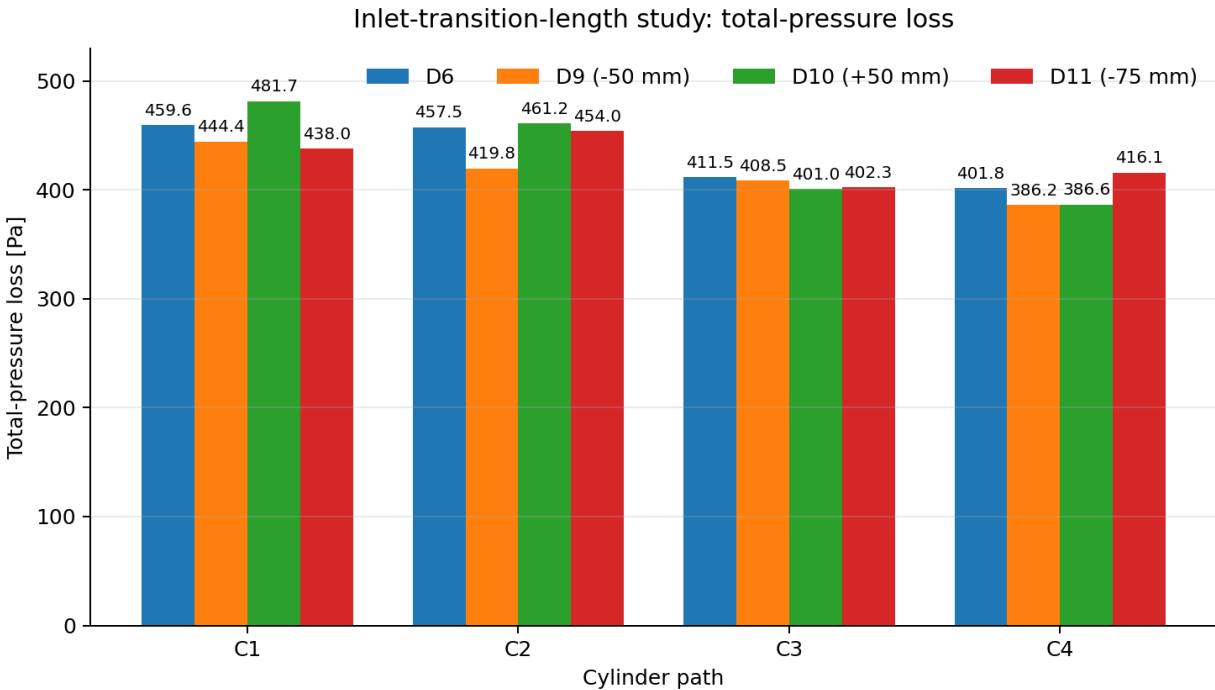


Figure 25. D6, D9, D10 and D11 total-pressure loss for the four matched steady cylinder paths in the inlet-transition-length study.

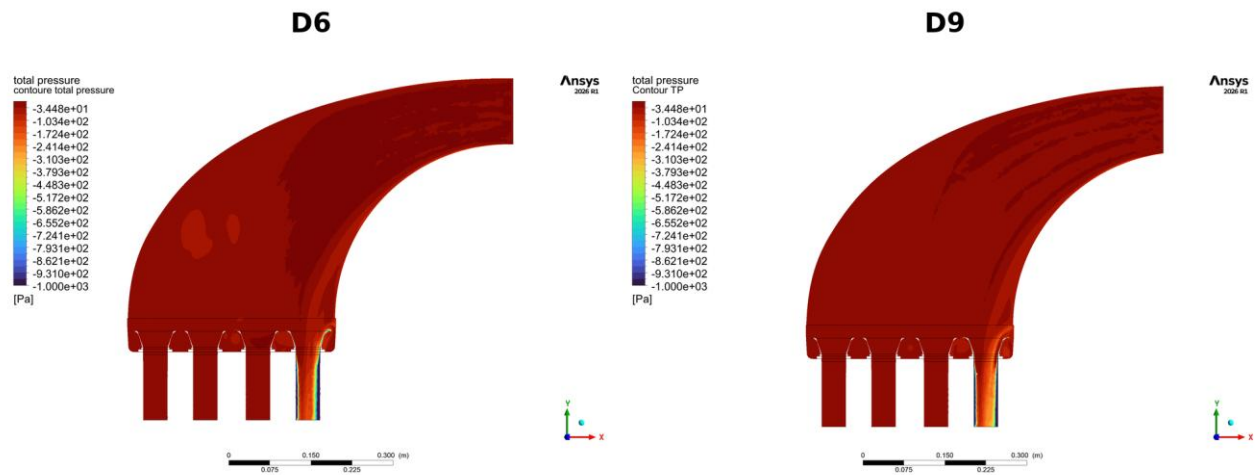


Figure 26. Matched C1 total-pressure contours for D6 and D9 at the controlled 0 Pa inlet condition. Both panels use the same -1000 to -34.48 Pa total-pressure scale.

Figure 26 shows a smaller low-total-pressure region along the limiting C1 path in D9, while Figure 27 shows reduced, but not eliminated, plenum circulation.

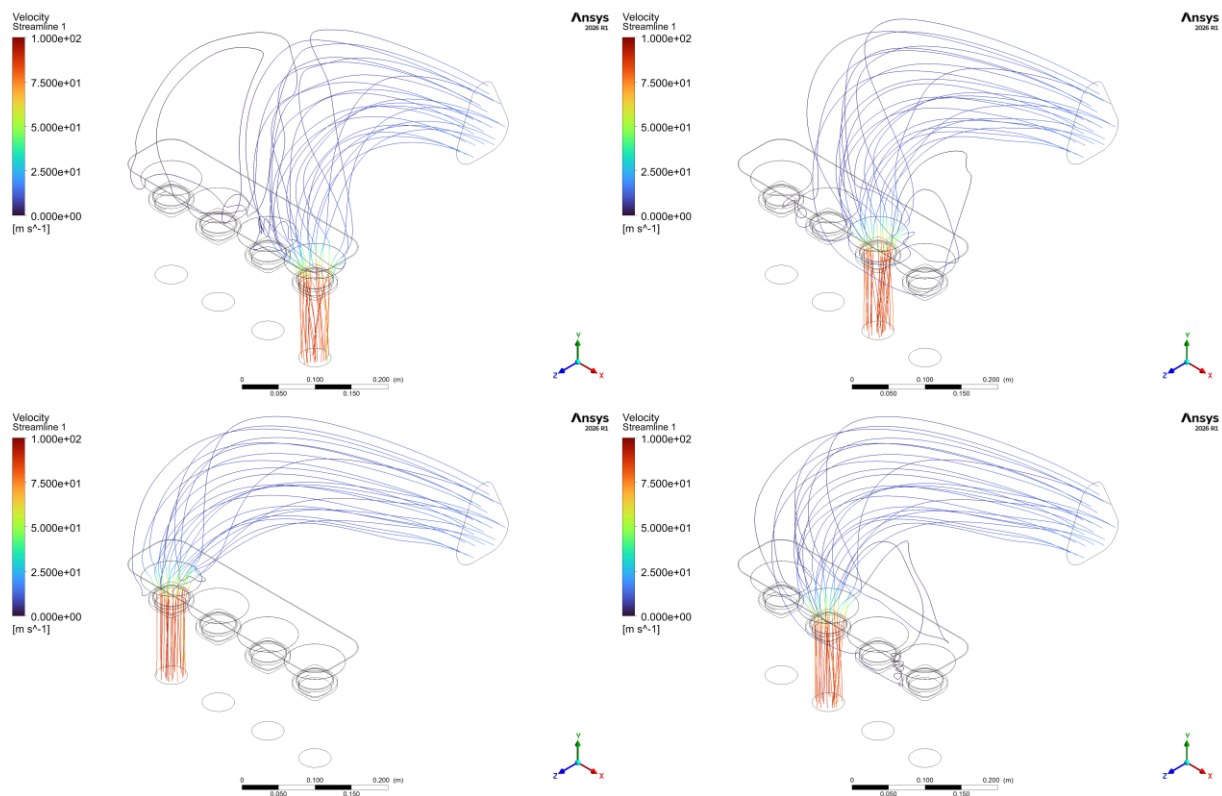


Figure 27. D9 (-50 mm transition) inlet-seeded streamline fields for C1–C4 at the controlled 0 Pa inlet condition.

D10 gave no mean-loss benefit and increased C1 loss by 4.80%, with loss CV rising to 10.63%. The longer transition was therefore rejected.

D11 tested a further 25 mm shortening. Relative to D6, mean loss fell by only 1.15% and worst-path loss by 1.21%; both were worse than D9, although loss CV improved to 5.36%. Appendix G, Figure G15, shows the same broad loss regions and path-dependent circulation. Further shortening therefore did not continue the D9 benefit. D9 remains the best tested transition length, but its 4.10% isolated gain is close to the 4.48% model-preparation sensitivity and is not claimed as mesh independent.

Length also affects pressure-wave behaviour. Because the steady-state one-path CFD approach cannot resolve engine-speed-dependent resonance, D9 is selected only for steady pressure preservation and consistency; transient or coupled 1D/3D analysis is required for acoustic claims (Brennan et al., 2006; Ling and Tun, 2006).

4.13 Central-divider concept (D13)

D13 retained the D6 top, inlet and standard trumpet tray but introduced a short central divider between C2 and C3 in the trumpet-tray region. The divider was intentionally limited rather than extended through the plenum. Roberts (2025) investigated a different, but broadly similar, intake-plenum geometry and found that a longer turning-vane concept could act as an obstruction and generate a large recirculation region. Although the geometries were not directly comparable, this provided useful qualitative guidance and was one reason a shorter local divider was used for D13. The aim was to influence lateral cross-plenum motion while limiting added blockage.

Table 33. D6 versus D13 central-divider performance summary.

Metric	D6 reference	D13 short divider	Change
Mean total-pressure loss [Pa]	432.6	429.1	-0.80%
Worst-path loss [Pa]	459.6	452.8	-1.48%
Loss CV [%]	6.99	6.00	-0.99 pp
Mean outlet mass flow [kg s ⁻¹]	0.162178	0.162091	-0.05%

Table 33 shows that D13 produced only a small improvement relative to D6. Mean and worst-path total-pressure loss both decreased slightly, while mean mass flow was effectively unchanged. Loss CV also improved from 6.99% to 6.00%. The small pressure-loss benefit was concentrated in C1/C2; C3 changed little and C4 became slightly worse.

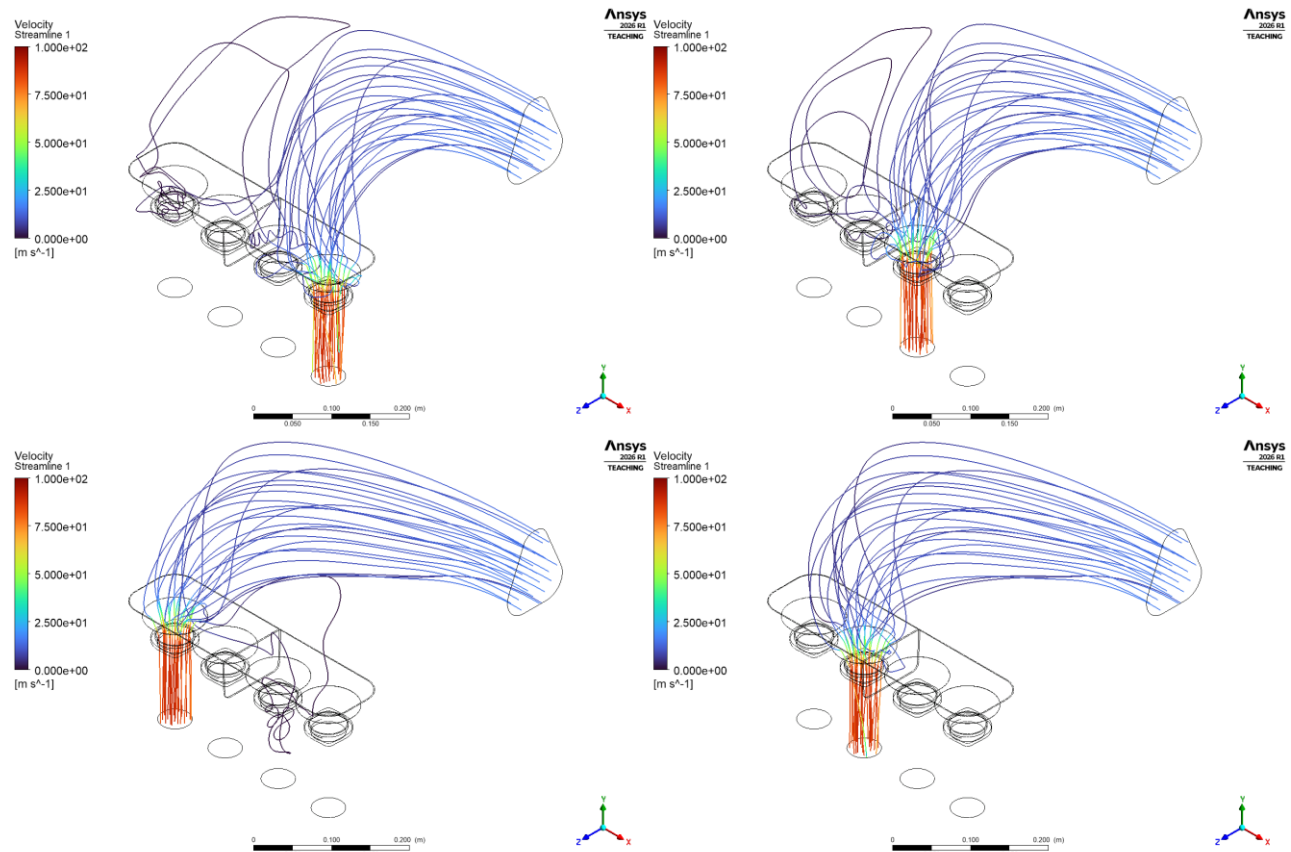


Figure 28. D13 central-divider inlet-seeded streamline fields for the four separately simulated cylinder paths at the controlled 0 Pa inlet condition.

Figure 28 shows that the short divider changes the local flow around the centre of the trumpet tray but does not remove the broader cross-plenum circulation. With only a 0.80% reduction in mean loss, the benefit is too small to justify the additional obstruction and manufacturing complexity, so D13 was rejected. The remaining supporting fields are grouped in Appendix G, Figure G17.

4.14 Trumpet-tray extension study (D12 and D14)

The trumpet tray was varied independently of the inlet-transition-length study while retaining the D6 top, inlet and runner/trumpet geometry. D12 extended the tray by 50 mm and D14 by 100 mm relative to D6. The modification increases both geometric clearance below the plenum and total lower-plenum volume, so the comparison is treated as a trumpet-tray extension/volume sensitivity rather than a pure length-only study.

Table 34. Trumpet-tray extension performance comparison for D6, D12 and D14.

Metric	D6 reference	D12 +50 mm	D14 +100 mm
Mean outlet mass flow [kg s ⁻¹]	0.162178	0.162735 (+0.34%)	0.162432 (+0.16%)
Mean outlet velocity [m s ⁻¹]	85.49	85.56	85.28
Mean total-pressure loss [Pa]	432.6	423.9 (-2.01%)	451.4 (+4.35%)
Worst total-pressure loss [Pa]	459.6 (C1)	448.3 (C1, -2.46%)	517.7 (C1, +12.65%)
Mass-flow CV [%]	0.32	0.32	0.67

Metric	D6 reference	D12 +50 mm	D14 +100 mm
Loss CV [%]	6.99	5.17	10.51

D12 gave a small improvement relative to D6: mean and worst-path loss both decreased and loss CV fell to 5.17%, with no material penalty on another path. D14 reversed this trend; the larger extension increased mean loss and strongly worsened C1.

Figure 30 adds the D6 reference to the two tray extensions. D12 retains the same broad cross-plenum flow topology as D6, and the visual change is subtle, consistent with its modest 2.01% mean-loss reduction rather than a wholesale change in the flow mechanism. D14, by contrast, shows longer recirculating paths in the enlarged lower-plenum region, consistent with its 12.65% C1 penalty and higher loss CV. D12 was therefore the better tray case and was carried into D15, but it is not treated as an independently confirmed gain.

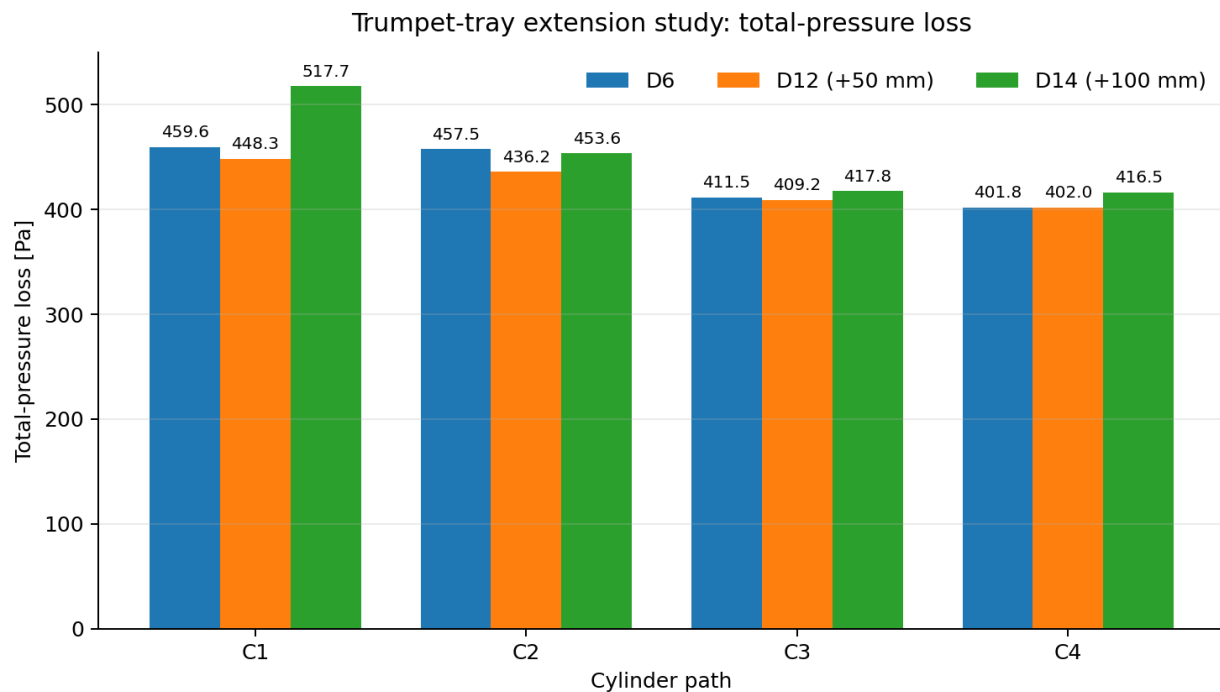


Figure 29. D6, D12 and D14 total-pressure loss for the four matched steady cylinder paths in the trumpet-tray extension study.

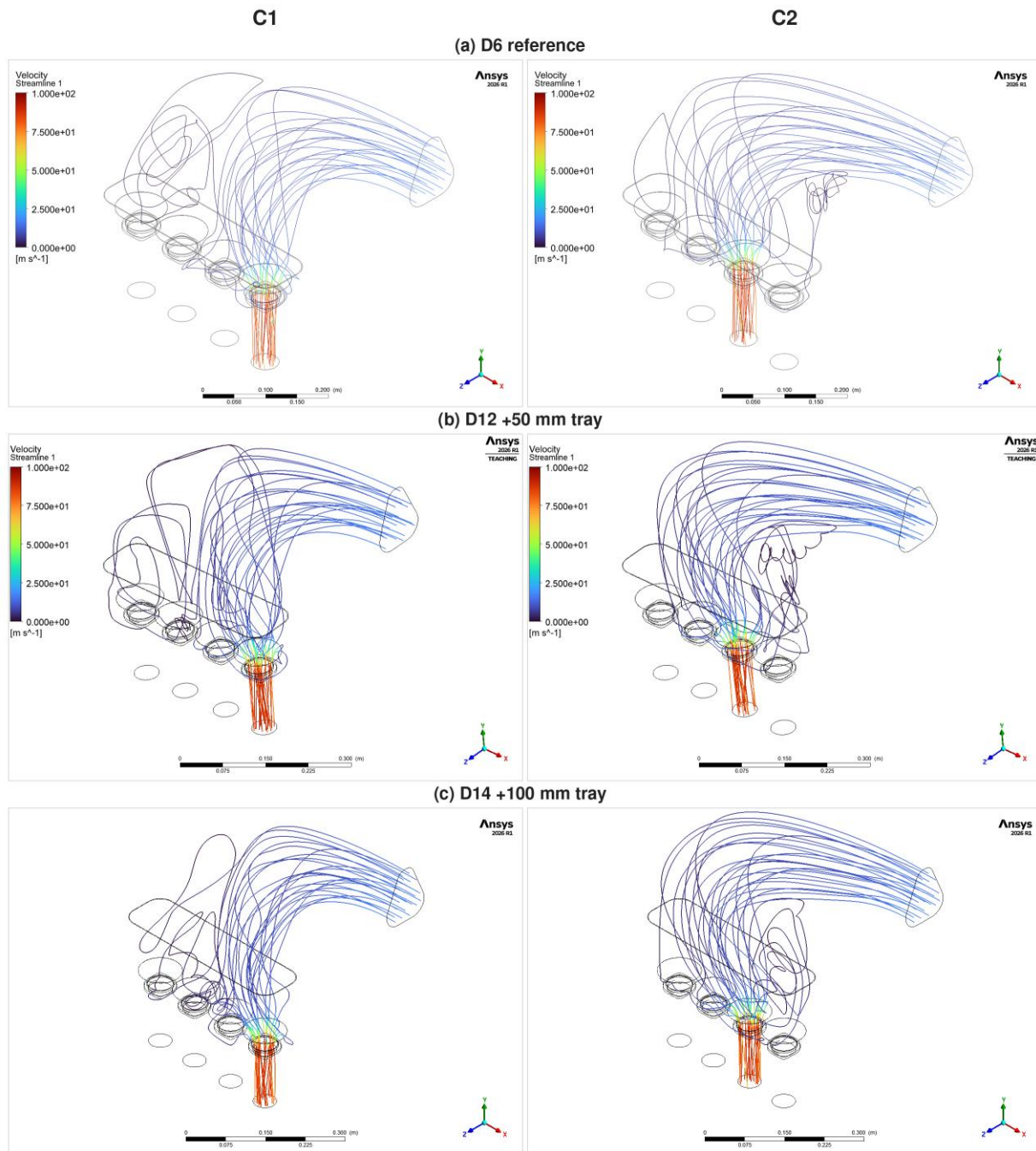


Figure 30. D6 reference, D12 (+50 mm) and D14 (+100 mm) inlet-seeded streamlines for the C1 and C2 paths at the controlled 0 Pa inlet condition. Rows compare the tray lengths and columns compare the two cylinder paths.

4.15 Combined D9 + D12 concept assessment (D15)

D15 combined the two preferred independently tested features: the D9 inlet transition shortened by 50 mm relative to D6 and the D12 +50 mm trumpet-tray extension. The D6 top/opening and fixed runner/trumpet geometry were otherwise retained. The purpose was to test whether the pressure-loss reductions obtained independently from inlet-trajectory and lower-plenum changes were additive under the same controlled 0 Pa inlet and -4982 Pa active-outlet condition.

Table 35. Combined-concept performance comparison for D6, D9, D12 and D15.

Metric	D6 reference	D9 -50 mm inlet	D12 +50 mm tray	D15 D9 + D12
Mean outlet mass flow [kg s ⁻¹]	0.162178	0.162633	0.162735	0.162398
Mean outlet velocity [m s ⁻¹]	85.49	85.70	85.56	85.28
Mean total-pressure loss [Pa]	432.6	414.7	423.9	452.7
Worst total-pressure loss [Pa]	459.6 (C1)	444.4 (C1)	448.3 (C1)	506.0 (C1)
Mass-flow CV [%]	0.32	0.42	0.32	0.46
Loss CV [%]	6.99	5.84	5.17	8.78

D15 did not retain either parent concept's benefit. Mean loss rose to 452.7 Pa, above D9, D12 and D6, while C1 became the worst of the carried-forward cases and loss CV increased to 8.78%. Mass imbalance remained below 0.11%, so the adverse ranking is not explained by a continuity error.

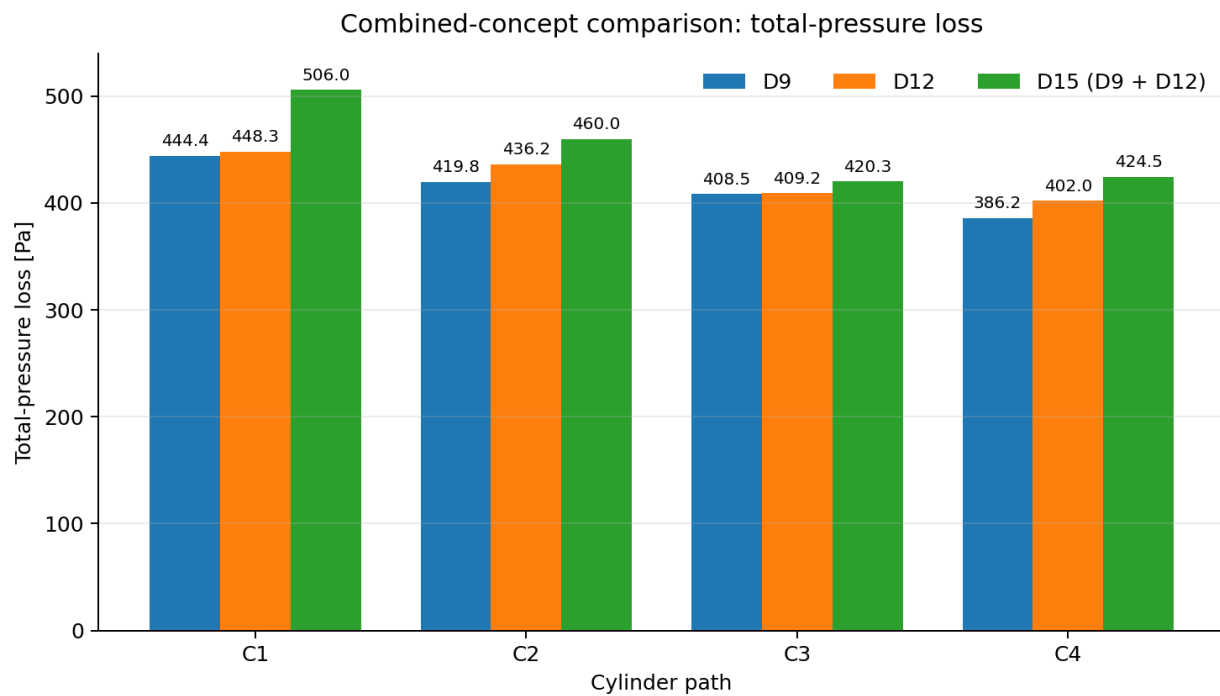


Figure 31. D9, D12 and D15 total-pressure loss for the four matched steady cylinder paths, showing the non-additive combined response.

Figure 32 shows long cross-plenum loops and recirculating paths around the inactive trumpets, indicating interaction between the shortened inlet trajectory and enlarged lower plenum. The combined geometry therefore changes the flow mechanism instead of adding the two separate gains. Supporting fields are grouped in Appendix G, Figure G19.

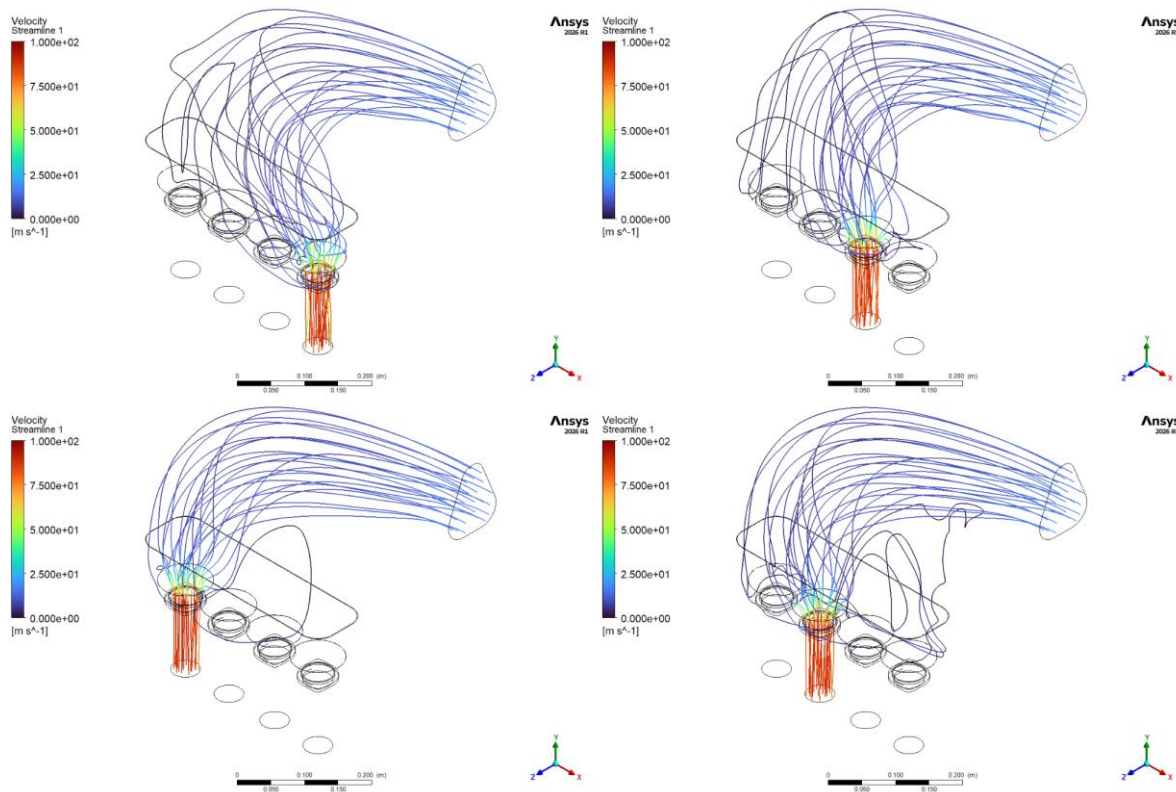


Figure 32. D15 (D9 + D12) inlet-seeded streamline fields for C1–C4 at the controlled 0 Pa inlet condition.

D15 was rejected. D9 remains the preferred controlled design because it has the lowest mean and worst-path loss among the carried-forward candidates and a lower loss CV than D6. It was selected for the +728.92 Pa robustness check and later whole-vehicle assessment.

4.16 Vehicle-derived inlet verification

The final internal check compared clean D0 and D9 at the controlled 0 Pa inlet and at a common +728.92 Pa inlet total gauge pressure, with the active outlet fixed at -4982 Pa. Applying the same higher inlet pressure to both geometries isolates the internal response at the higher flow level; it remains a sensitivity condition rather than an installed-engine operating point.

Figure 33 shows the path-level comparison at both matched inlet conditions. At 0 Pa, D9 reduced mean loss by 5.04% and worst-path loss by 3.75% relative to D0, with all four paths improving. At +728.92 Pa, its mean-loss advantage reduced to 0.68% and mean mass flow remained 0.45% higher, but C2 loss increased by 2.89% and loss CV rose from 4.72% to 7.15%. The higher-pressure result therefore supports only a small mean advantage and shows a less favourable C2 response.

D0 versus D9 total-pressure loss at matched inlet conditions

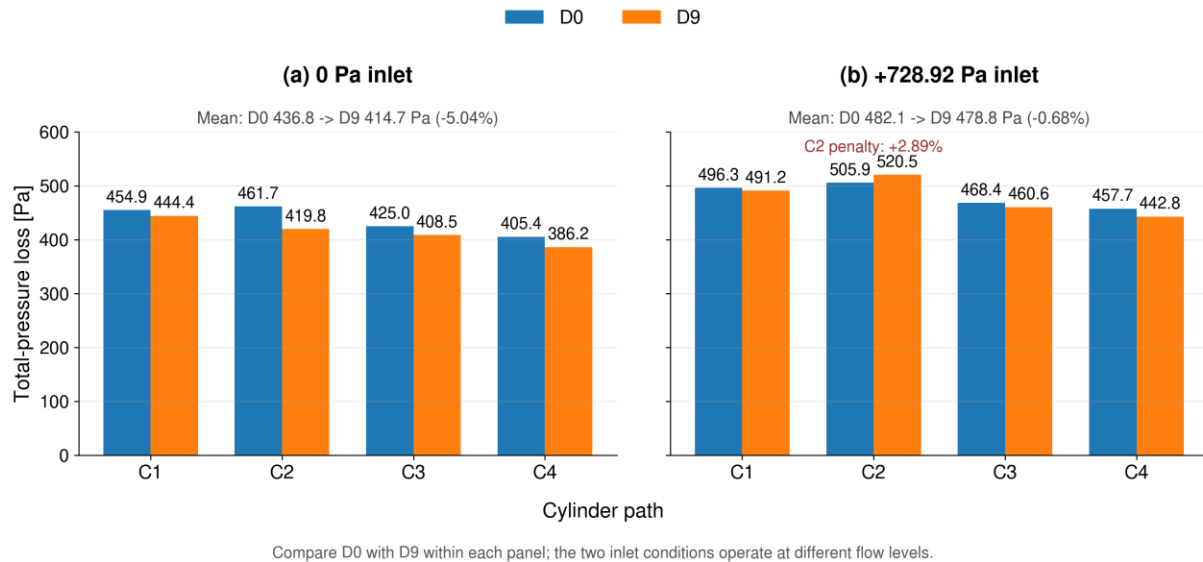


Figure 33. Clean D0 and D9 total-pressure loss for the four matched cylinder paths at (a) the controlled 0 Pa inlet and (b) the uniform +728.92 Pa vehicle-derived inlet-pressure sensitivity condition. Comparisons are made within each inlet condition.

The higher inlet pressure raised D9 mean mass flow by about 7.06% relative to its own 0 Pa case, so the larger absolute loss at +728.92 Pa is not interpreted as a direct deterioration. The useful comparison is the matched D0-D9 difference at each inlet condition.

Table 36 summarises the separate matched whole-vehicle comparison: D9 reduced vehicle and airbox drag coefficients by 1.61% and 8.13%, respectively, with a more negative vehicle lift coefficient. The earlier inlet-pressure pilot and this later external study are separate result sets, so their absolute coefficients are not cross-compared. The final export did not contain D9 inlet-mouth static or total pressure; +728.92 Pa therefore remains the earlier pilot sensitivity value rather than a revised D9 pressure supply.

Table 36. Drag and lift coefficients for the matched S2N+ and D9 external CFD cases at 35 m/s.

External configuration	Vehicle C_D	Vehicle C_L	Airbox C_D
S2N+ airbox	0.56139	-1.9018	0.00846
D9 airbox	0.55235	-1.9756	0.00777

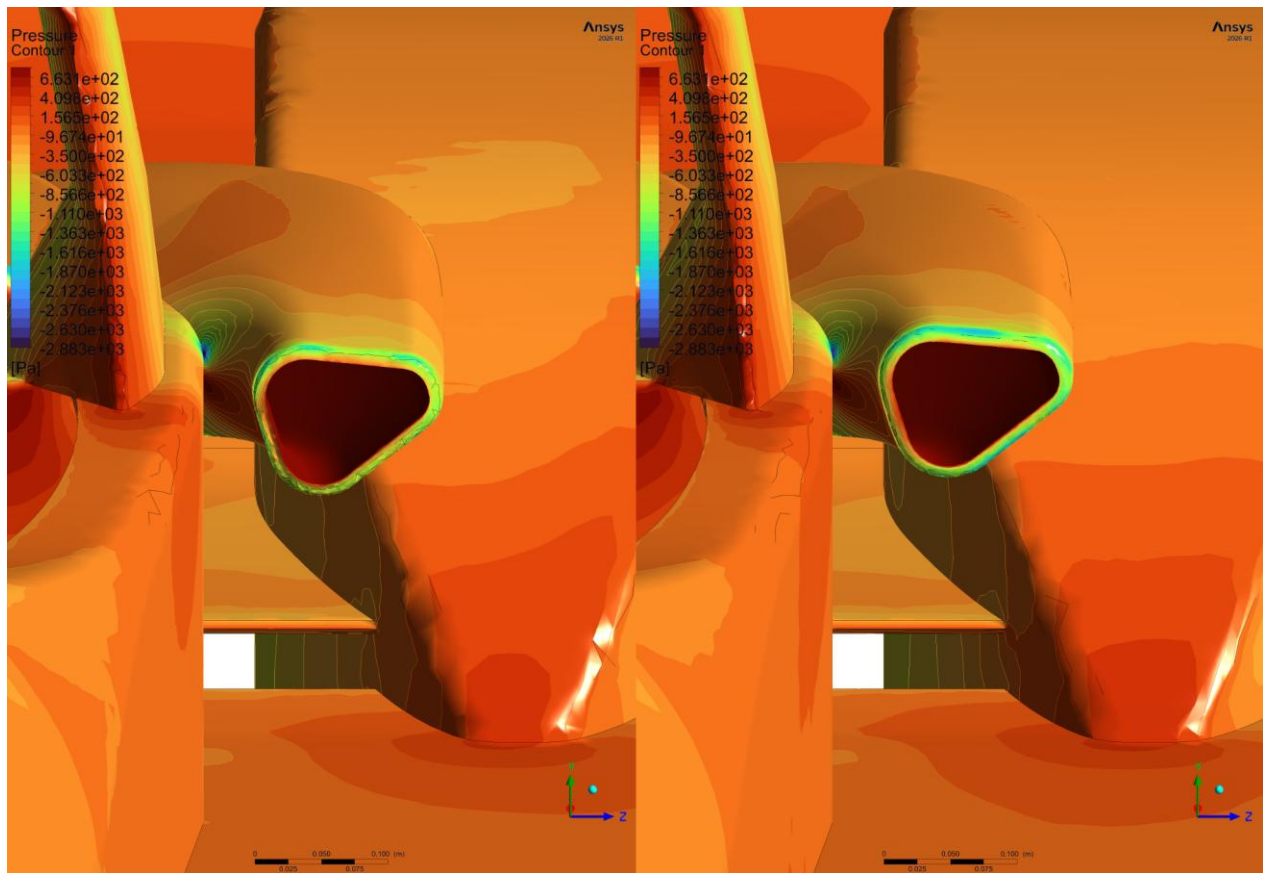


Figure 34. Matched whole-vehicle static-pressure field around the airbox opening for the (left) S2N+ updated-airbox and (right) D9 installations. A common pressure scale is used.

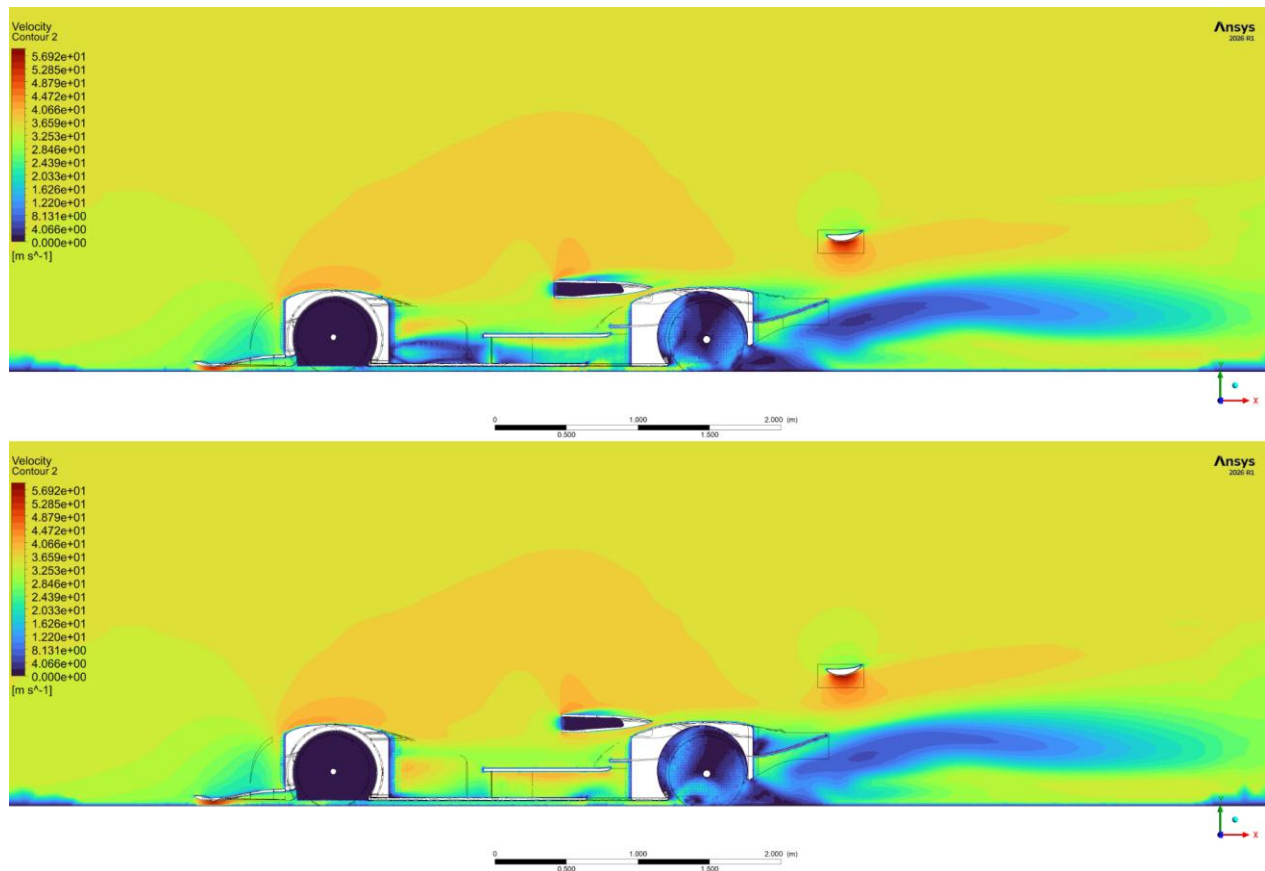


Figure 35. Matched whole-vehicle centre-plane velocity-magnitude field for the (top) S2N+ updated-airbox and (bottom) D9 installations. A common velocity scale is used.

Figures 34 and 35 show that the local pressure field around the airbox opening and the large-scale external velocity field remain broadly similar after the D9 geometry change. No new large-scale separation or wake feature is evident in the matched centre-plane view. This qualitative evidence is consistent with the within-set force-coefficient comparison and supports installation compatibility of D9. The contours are not used to infer a revised inlet-pressure boundary because the final D9 export did not contain area-averaged inlet-mouth static or total pressure.

Table 37. D0 versus D9 performance at the uniform +728.9 Pa vehicle-derived inlet-pressure sensitivity condition.

Cylinder	D0 outlet velocity [m s ⁻¹]	D9 outlet velocity [m s ⁻¹]	D0 mass flow [kg s ⁻¹]	D9 mass flow [kg s ⁻¹]	Total-pressure loss [Pa], D0 / D9
C1	91.499	91.59	0.173159	0.173612	496.3 / 491.2
C2	91.442	91.30	0.172385	0.173342	505.9 / 520.5
C3	91.738	91.85	0.173483	0.174421	468.4 / 460.6
C4	91.841	91.93	0.174296	0.175076	457.7 / 442.8
Mean / CV	91.630 / 0.21%	91.668 / 0.31%	0.173331 / 0.46%	0.174113 / 0.45%	482.1 / 4.72% / 478.8 / 7.15%

4.17 Design summary

The main design conclusion is that D9 is the preferred prototype candidate, not a confirmed final improvement. The design study narrowed the options by rejecting concepts that increased the limiting path, produced only marginal changes or interacted unfavourably when combined. D9 improved all four paths relative to D6 and gave the lowest mean and worst-path loss among the carried-forward concepts. The +728.92 Pa check reduced its mean advantage and introduced a C2 penalty, so experimental confirmation remains necessary. Table 38 gives the decision-level summary; the complete D0-D15 numerical dataset is retained in Appendix E.

Table 38. Final design decision summary.

Concept	Mean Δp_t [Pa]	Worst path	Mean-loss change	Decision
D0	436.8	461.7 Pa (C2)	Reference	Updated-airbox baseline
D6	432.6	459.6 Pa (C1)	-0.96% vs D0	Lipless design reference
D1	429.7	459.3 Pa (C2)	-1.61% vs D0	No-go: small wall-profile gain
D8	421.4	458.6 Pa (C2)	-2.59% vs D6	No-go: best size case, weaker than D9
D9	414.7	444.4 Pa (C1)	-4.10% vs D6	GO: preferred prototype
D12	423.9	448.3 Pa (C1)	-2.01% vs D6	Not selected alone; tested in D15
D13	429.1	452.8 Pa (C1)	-0.80% vs D6	No-go: marginal benefit
D14	451.4	517.7 Pa (C1)	+4.35% vs D6	No-go: strong C1 penalty
D15	452.7	506.0 Pa (C1)	+4.65% vs D6	No-go: combined response detrimental

5. Discussion

This chapter discusses the validation evidence, the main flow mechanisms identified in the matched comparisons, the effect of the external-inlet assumption and the basis for selecting D9.

5.1 Credibility and limits of the validated CFD method

The pipe validation supports the retained CFD method as a comparative engineering tool rather than an exact local-pressure predictor. At the principal 20 inH₂O condition, M3 reproduced the measured rise and fall along both wall paths and correctly placed the minimum at Tap 12. The Tap-1-referenced correlation was 0.943 with an RMSE of 0.745 kPa. The main local weakness was the Tap 9–10 pressure change, which was underpredicted by approximately 37.4%. Because the refined M2–M4 solutions were closely grouped, this discrepancy is unlikely to be removed by volume refinement alone.

The near-wall study gives similar evidence. Introducing prism layers changed the solution noticeably relative to the tetrahedral-only case, but L8 and L12 produced almost the same mass flow and Tap 9–10 response. The common-scale y^+ fields also showed the expected improvement in near-wall resolution. L8 was therefore retained because increasing the layer count to L12 produced very little further change for the quantities being compared. The remaining experimental mismatch is therefore unlikely to be explained by prism-layer count alone.

The turbulence-model comparison did not identify a clearly superior alternative to SST $k-\omega$. SST, Realizable $k-\epsilon$ and RNG $k-\epsilon$ reproduced the same overall pressure pattern and minimum-pressure location, and none removed the local lower-wall discrepancy. The remaining error may therefore reflect a combination of turbulence-model limitations, geometry or tap-location uncertainty and experimental effects rather than the choice between the three tested two-equation models (Menter, 1994; Ansys, 2024a).

Across the complete 5–25 inH₂O sweep, correlation remained between 0.897 and 0.943 and the normalised RMSE remained within 10.8–14.4% of the measured pressure range. The method therefore reproduces the same overall pressure-redistribution pattern over the tested range. This supports its use for controlled geometry comparisons when the results are considered together with the sensitivity checks and flow-field evidence, but it does not justify claiming exact wall pressure or absolute component loss at every location.

5.2 Transferability from the pipe test to the airbox

The pipe test validates the CFD prediction of wall-static-pressure distribution for the tested bend; it does not validate the complete airbox model. ASME V&V 20 defines validation for specified response quantities and conditions (ASME, 2009). Applying the same M3/L8/SST method to the airbox is reasonable because both flows contain curvature, adverse pressure gradients and possible separation, although the airbox adds inlet-jet expansion, plenum mixing and trumpet interaction.

The level of confidence differs between the stages of the study. The pipe comparison provides the strongest validation because the CFD and experiment use matched tap locations and bench depressions. The updated-airbox comparison is less direct because the experimental flow path includes the cylinder head, port and partially open valve, while the CFD pressure outlet is located at the runner end. It is therefore used mainly to assess agreement in the spatial pressure pattern. The old-airbox and design cases are treated as controlled CFD comparisons. A design result is more convincing when the same trend is seen across the relevant cylinder cases, is supported by the flow-field behaviour and is considered alongside the observed model-preparation sensitivity (ASME, 2009; Celik et al., 2008).

The airbox bench and internal CFD were intentionally compared without an air-filter element. This improves consistency between the experimental and numerical flow paths, but it limits direct transfer to the complete installed induction system. A filter would add restriction and could alter inlet velocity uniformity or turbulence. The present ranking should therefore be interpreted as the aerodynamic behaviour of the unfiltered airbox geometry rather than the final filtered vehicle system.

5.3 Updated-airbox pressure mechanism

Both the bench data and CFD identify the lower opening and transition as the region of greatest wall-static-pressure reduction, followed by partial recovery in the plenum. This is consistent with acceleration and turning through the transition followed by deceleration as the flow enters the larger plenum volume. The CFD reproduces the same overall pattern but predicts much larger pressure differences than the experiment. The agreement is therefore stronger for the location and sequence of the pressure changes than for their absolute magnitude.

The regional static-pressure increase inside the plenum must not be interpreted as complete pressure recovery. Mixing, separation and recirculation can increase static pressure while total pressure is still being lost (Anderson, 2016; Versteeg and Malalasekera, 2007). The systematic CFD suction bias is consistent with the known downstream-boundary mismatch: the bench flow passes through the runner, head, port and partially open valve, whereas the numerical pressure outlet is applied at the runner end. The airbox tap study therefore supports use of the CFD to interpret the broad spatial pressure pattern, but not to predict the absolute pressure level in the assembled engine.

5.4 Old-versus-updated airbox comparison

The clean D0 comparison shows a clear internal-flow improvement over the original airbox under the common pressure boundaries. Mean mass flow increased by 3.72% while mean total-pressure loss fell by 37.2%, and every cylinder path improved. The worst-path loss also decreased substantially, so the lower mean loss was not achieved by worsening one runner path.

This result is consistent with the flow-field interpretation. The original airbox requires more cross-plenum turning and contains larger low-total-pressure regions before the active runner, whereas the updated geometry provides a more direct route through the plenum. Path-to-path variation is also reduced, indicating that the updated geometry improves both overall pressure preservation and the consistency of the four separately simulated flow paths. These statistics describe steady path resistance, not simultaneous cylinder filling.

The pressure-tapped updated model remains useful for experimental correlation, but its predicted loss is more sensitive to model preparation than its bulk mass flow. The clean D0 rerun therefore provides the quantitative baseline for the design programme, while the tapped-model visualisations are used only to discuss the general flow pattern.

5.5 External inlet-condition assumption

The external CFD shows why configuration-specific inlet pressures were not used for the internal geometry ranking: this would combine changes in external pressure capture with changes in internal airbox resistance. The common 0 Pa inlet therefore provides a more controlled basis for comparison. Applying the externally derived +728.92 Pa total-pressure condition to D0 increased the overall flow level without changing the basic cylinder ordering, which supports the use of the 0 Pa cases for design screening.

Applying the same higher-pressure condition to D9 gives a more cautious result. D9 still had 0.68% lower mean loss and 0.45% higher mean mass flow than D0, but C2 loss increased by 2.89% and the loss spread increased. D9 therefore retained a small mean improvement, but it did not improve every cylinder path under the higher imposed pressure.

Published whole-vehicle work provides a useful contrast: Patil and Patil (2021) reported a 1.25% drag-coefficient deviation from Ford Ka wind-tunnel data using a 28-million-cell Mosaic mesh and the GEKO turbulence model. The present external model was not validated to the same level and omitted engine through-flow; consequently, its derived inlet pressure is appropriately retained only as a sensitivity boundary.

5.6 Design modification and performance trade-offs

The wall-profile study shows that improving the appearance of a local flow region does not necessarily improve the complete intake path. D1 produced only a small mean-loss reduction, while D2 and D3 shifted more of the loss towards C1/C2. Since these changes are smaller than the model-preparation sensitivity identified in Section 4.5, they are best treated as trends. The results indicate that changing local wall curvature can redistribute the runner approach flow rather than reduce resistance across all four paths.

The inlet-shape and inlet-size studies lead to the same design lesson. D4/D5 reduced mean loss but worsened the limiting C1 path, while D8 gave the best size-study mean response without solving the path-consistency problem. These cases show why mean loss alone is insufficient: a geometry may improve several paths while moving the dominant loss to another cylinder. The worst path and the four-case spread therefore remain necessary alongside the mean.

Transition length produced the clearest effect in the design study. D9, with a 50 mm shorter transition, improved both mean and worst-path loss relative to D6 and reduced the loss spread. Lengthening the transition did not help, while shortening it by a further 25 mm in D11 removed much of the D9 benefit. This shows that D9 was the best of the tested transition lengths rather than demonstrating that further shortening would continue to improve the result. The likely reason is a change in inlet-jet trajectory and its interaction with the lower plenum; any engine-speed-dependent resonance effect remains outside the steady model.

The D13 divider gave only a small improvement in loss spread and did not remove the broader cross-plenum circulation. The benefit was therefore too small to justify the added obstruction and manufacturing complexity. This is qualitatively consistent with Roberts (2025), who found that a more extensive turning-vane arrangement in a different intake-plenum geometry could obstruct the flow and generate a large recirculation region. The geometries are not directly comparable, but the earlier result supports the decision to test only a short divider rather than a long one.

The trumpet-tray study also shows that a larger change was not necessarily better. The +50 mm D12 extension gave a modest improvement, whereas the +100 mm D14 geometry reversed the trend and strongly penalised the limiting path. Additional lower-plenum volume is therefore not automatically beneficial; the result depends on how the incoming jet expands, turns and interacts with the inactive trumpet region.

D15 provides the clearest evidence that the design variables interact. D9 and D12 were individually favourable, yet their combination increased mean loss and substantially worsened C1. The streamline field indicates a different three-dimensional circulation pattern when the shortened inlet trajectory discharges into the enlarged lower plenum. The two changes therefore cannot be treated as independent component improvements whose gains can simply be added.

Overall, the design study shows that small geometry changes often moved the loss from one path to another rather than reducing it everywhere. Each concept therefore needs to be judged using the mean result, the worst cylinder path and the supporting flow field together. D9 is preferred because it gave the best overall standalone result across mean loss, worst-path behaviour and consistency, while D15 showed that combining individually favourable changes did not necessarily improve the final geometry.

5.7 Comparison with published intake studies

The pipe result is consistent with the curvature-driven flow measured by Sudo, Sumida and Hibara (1998), where centrifugal effects create a transverse pressure gradient, distort the primary velocity profile and generate secondary motion through a 90-degree bend. The present experiment and CFD show their largest local sensitivity in the same strongly redistributed bend region. Roberts (2025) provides the closest procedural precedent because of the similar pressure-tapped bend and flow-bench workflow, although direct numerical comparison would require identical geometry, tap coordinates and pressure references. The measured maximum-to-minimum wall-static-pressure range is therefore not treated as an irreversible bend-loss coefficient.

At airbox level, the present workflow is consistent with the staged development strategies reported in the literature. De Vita et al. (2002) used CFD to reduce losses in a naturally aspirated prototype race-car airbox before track assessment. Muckova et al. (2023) combined CFD-led geometry development with sequential outlet flow-bench testing, and Montenegro et al. (2013) used lower-cost screening before higher-fidelity coupled analysis and experiment. These studies support the use of CFD for comparative geometry development, while also showing that transient engine demand and experimental confirmation are required before translating a steady flow improvement into an engine-performance claim.

The systematic airbox pressure bias also agrees with the broader validation caution in Masi, Artico and Gobbato (2015) and Nanni et al. (2008). Masi et al. obtained closer physical matching by including the downstream port, valve and cylinder geometry, whereas Nanni et al. retained useful comparative trends despite a persistent absolute flow bias. For the present work, this means the conclusions should remain limited to what the validation supports: matched steady comparisons are useful, but absolute installed-engine flow and cycle-resolved cylinder filling are not predicted.

5.8 Numerical and experimental uncertainty

The experimental datasets have different limitations. The pipe sweep contains one stabilised reading at each depression, so it provides an operating-point pressure pattern but no direct measure of repeatability. The airbox dataset has three valve-open repeats per cylinder, but only one ambient reading per set, so slow zero drift cannot be separated from the measured pressure response. Instrument effects, repeatability and the downstream boundary mismatch are therefore discussed separately rather than being combined into a single uncertainty value (ASME, 2018).

Numerical evidence is also stronger for the pipe than for the airbox. The pipe includes dedicated volume, near-wall and turbulence-model sensitivity studies, whereas the airbox does not have a matched coarse/medium/fine sequence. The tapped-to-clean preparation check changed mean mass flow by only 0.08% but mean total-pressure loss by 4.48%, showing that small changes in model preparation can affect design differences of only a few percent. D9 is the strongest candidate, but its 4.10% gain relative to matched D6 is close to this 4.48% model-preparation sensitivity. A matched airbox refinement study is therefore required before the D9 gain is described as a confirmed numerical improvement (Celik et al., 2008).

The external CFD adds a separate modelling limitation because it contains no engine through-flow and the transferred inlet value is spatially averaged. Consequently, the external calculations can test sensitivity to a

plausible pressure supply and installation compatibility, but they cannot validate the instantaneous or cycle-averaged pressure presented to the operating engine.

5.9 Engineering recommendation

The recommendation is to carry D9 forward as the preferred prototype candidate while retaining clean D0 as the reference geometry. D9 gave the best overall standalone result in the controlled study, with lower mean and worst-path loss and improved loss consistency. D8 and D12 also showed favourable trends, but neither provided a stronger overall result than D9.

The result is not strong enough to describe D9 as a confirmed final improvement. Its loss reduction is close to the model-preparation sensitivity, and the +728.92 Pa sensitivity case introduced a C2 penalty even though the mean result remained favourable. The later whole-vehicle result supports packaging and aerodynamic compatibility, but it does not provide a new D9 inlet-mouth pressure or validation with engine through-flow.

The next step should be to manufacture D9 and compare it with D0 on the flow bench. A matched D0/D9 mesh-refinement check should then be used to assess whether the predicted difference is strongly dependent on mesh resolution. If the experimental advantage is retained, D9 should be installed on the vehicle and tested under representative running conditions, ideally with airbox-pressure and engine-performance measurements. Transient or coupled 1D/3D analysis could then be used if further investigation of valve demand, pressure waves and engine-speed effects is required.

6. Conclusion and Recommendations

This dissertation established a steady CFD method supported by experimental validation and applied it to the MCR-S2N/S2N+ airbox package. The M3/L8/SST method reproduced the measured 90-degree bend wall-pressure pattern across 5-25 inH₂O and the updated-airbox tap comparison reproduced the main regional pressure pattern, although absolute suction was overpredicted. The strongest engineering result was the original-to-S2N+ comparison: clean D0 increased mean mass flow by 3.72% and reduced mean total-pressure loss by 37.2%, with all four separate cylinder paths improving.

The subsequent design study identified D9 as the preferred prototype. Its 50 mm shorter lipless transition reduced mean loss by 4.10% relative to D6, but this change is close to the 4.48% model-preparation sensitivity and the higher-inlet-pressure case introduced a C2 penalty. D9 is therefore recommended for prototype testing rather than presented as a confirmed final improvement. The external work supports sensitivity assessment and vehicle compatibility but does not reproduce engine through-flow.

6.1 Answers to the research questions

RQ1 – Steady RANS validation. The selected M3/L8/SST method reproduced the measured bend wall-pressure distribution with correlations of 0.897-0.943 and normalised RMSE of 10.8-14.4% across 5-25 inH₂O. It is therefore suitable for comparative wall-pressure and geometry assessment within the tested range, but not for exact local-pressure or total-loss prediction.

RQ2 – Effect of the vehicle-derived inlet condition. Applying +728.92 Pa increased the overall flow level but did not change the basic D0 path ordering. D9 retained a small mean advantage over D0 at this condition, although C2 loss increased by 2.89%. The inlet assumption therefore affects the magnitude and path-level confidence more than the overall design direction.

RQ3 – Old versus updated airbox. Under common internal boundaries, clean D0 increased mean mass flow by 3.72% and reduced mean total-pressure loss by 37.2% relative to the original airbox. All four separate cylinder paths improved, making this the largest and clearest performance difference identified in the project.

RQ4 – Updated-airbox CFD-to-bench agreement. The 16-tap airbox comparison gave positive spatial correlations of $r = 0.742$ - 0.866 and reproduced the stronger transition suction followed by plenum pressure recovery. CFD nevertheless predicted more negative pressures than the bench, so the airbox evidence supports spatial correlation rather than absolute installed-engine pressure validation.

RQ5 – Numerical and experimental support. The pipe method is supported by volume-mesh, prism-layer and turbulence-model sensitivity studies, monitor stability and mass balance. The airbox bench data include three repeats per cylinder, but the airbox lacks a matched coarse/medium/fine refinement sequence. The main comparative conclusions are supported, while the few-percent D9 gain remains numerically unconfirmed.

RQ6 – Design improvement beyond observed sensitivity. No tested modification conclusively demonstrated a gain larger than the observed model-preparation sensitivity. D9 produced the best overall mean loss, worst-path result and path consistency, but its 4.10% improvement relative to D6 is close to the 4.48% preparation sensitivity. D9 is therefore the preferred prototype candidate, not a confirmed final improvement.

6.2 Recommendations for further work

Further work should confirm D9 experimentally and then establish whether the observed benefit remains under more representative numerical and vehicle conditions.

- 1. Manufacture D9 and repeat the D0/D9 flow-bench comparison with ambient readings before and after each run. Where practical, measure pressure at or near the CFD runner-outlet plane so that the experimental and numerical downstream boundaries are matched more closely.
- 2. Complete a representative matched D0/D9 airbox mesh-refinement study using identical local sizing logic. This will show whether the small predicted D9 advantage is sensitive to mesh resolution.
- 3. Repeat the D0/D9 comparison with the intended air-filter element installed, using measured filter pressure-drop data or a validated porous-media representation, to determine whether filtration changes the design ranking.
- 4. If the flow-bench advantage is retained, install D9 on the vehicle and test it under representative running conditions. Where practical, record airbox pressure and engine-performance data so that the CFD/bench trend can be checked in the complete intake system.
- 5. Re-run or post-process the installed D9 external case to extract static pressure, total pressure and flow non-uniformity at the same airbox-mouth plane used for the earlier pilot. Later work should also examine speed, yaw and inlet distortion if external pressure capture becomes a primary design variable.
- 6. Use transient or coupled 1D/3D modelling with credible cylinder/port pressure histories and valve timing to assess cycle-resolved filling, pressure-wave tuning and the engine-speed dependence of the shortened D9 transition before making acoustic, resonance or power claims.

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Appendices

Appendix A: Flow-bench test-rig record

Appendix A provides photographic records of the two experimental flow-bench arrangements used in the study.



Figure A1. Flow-bench setup for the pressure-tapped 90° pipe test, showing the SuperFlow SF-600E, test piece, pressure-scanner connections and data-acquisition equipment.

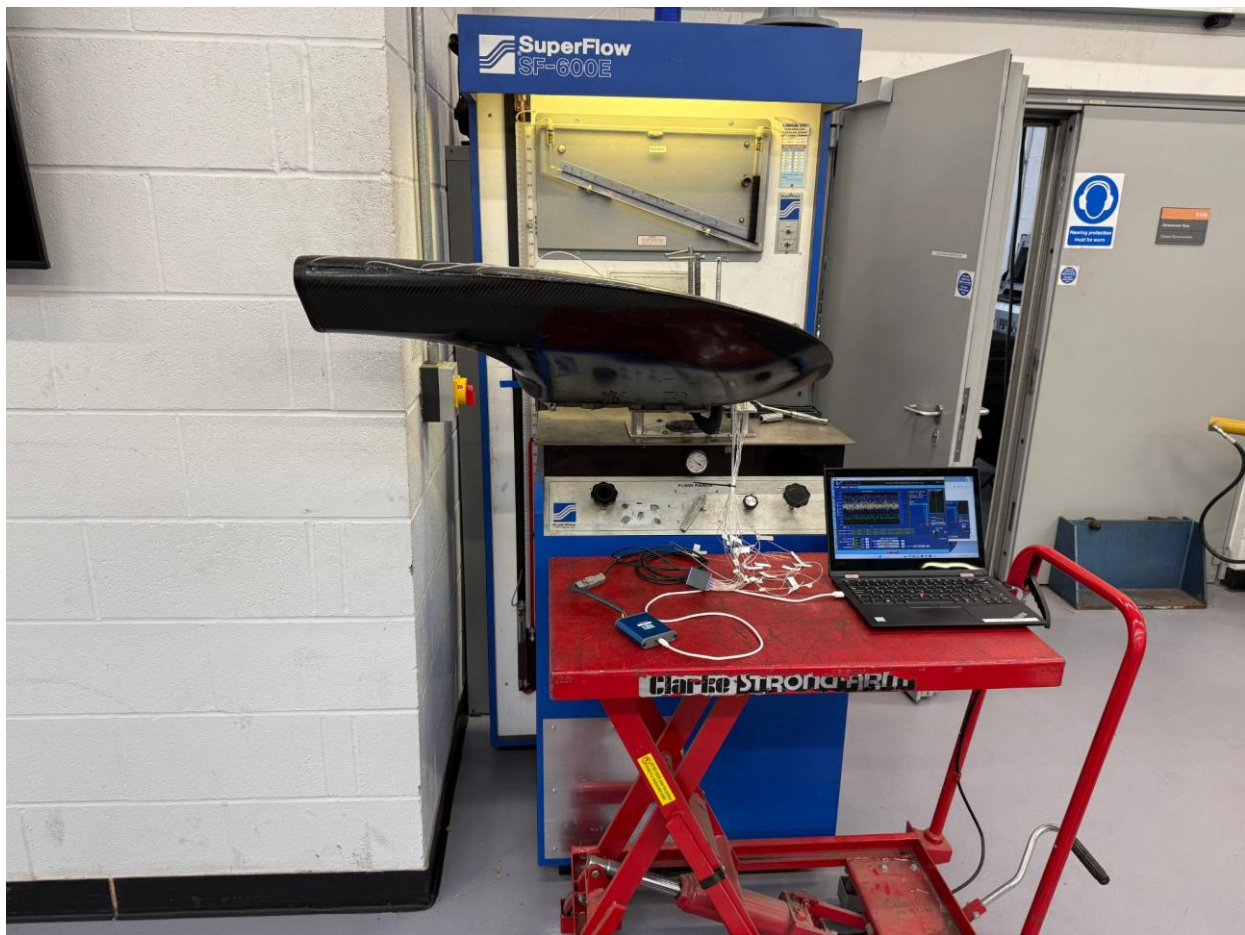


Figure A2. Flow-bench setup for the pressure-tapped S2N+ updated airbox, showing the installed airbox, pressure lines and data-acquisition equipment.

Appendix B: Raw experimental pressure data

Table B1. Updated-airbox experimental pressure-data summary.

Cylinder	Open repeats	Mean Delta p [Pa]	Run range [Pa]	Spatial SD [Pa]	Most negative tap	Highest tap	Notes
1	3	-43.94	4.06	15.97	T6	T16	Three repeats
2	3	-39.44	2.75	13.57	T1	T16	Three repeats
3	3	-32.56	2.13	10.65	T6	T16	Three repeats
4	3	-22.15	6.00	7.44	T6	T5	Three valve-open repeats

Appendix C: Supporting pipe CFD results

This appendix contains the additional near-wall CFD visualisation used to support the prism-layer assessment in Section 4.3.3. The quantitative volume-mesh, turbulence-model and pressure-sweep results are reported in the main text.

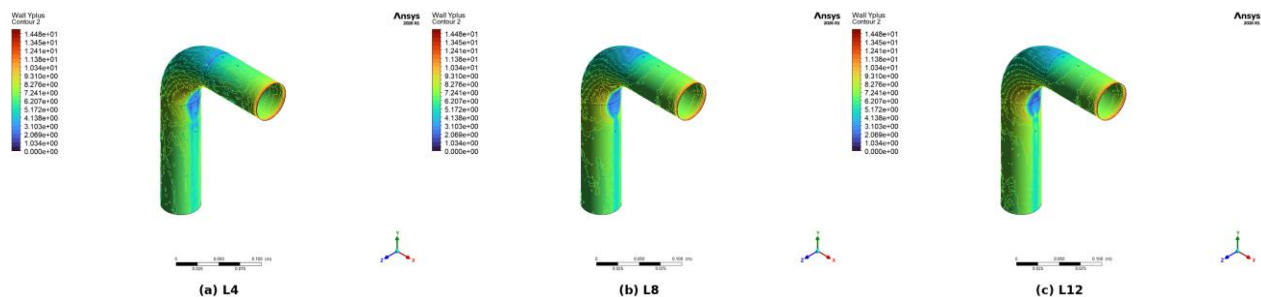


Figure C1. Detailed wall y^+ comparison for the L4, L8 and L12 near-wall meshes at 20 inH₂O.

Appendix D: External inlet sampling data

Table D1. External inlet sampling and boundary-transfer data.

Case	Inlet p _s [Pa]	Inlet p _t [Pa]	C _D	C _L	Use	Notes
Old-airbox external	600.3	792.6	0.4816	-1.4953	External reference	Selected straight-line whole-vehicle case; no engine through-flow.
Updated-airbox external	569.2	728.9	0.4805	-1.4328	Boundary transfer	Earlier inlet-condition pilot; 728.92 Pa transferred to D0 and D9 as a matched sensitivity value.
Clean D0 inlet-pressure sensitivity reference	n/a	728.9	n/a	n/a	Matched design reference	C1-C4 complete; matched directly with selected D9 at the same imposed inlet total pressure.
D9 inlet-pressure robustness check	n/a	728.9	n/a	n/a	Final internal robustness check	Completed; mean loss 478.8 Pa (-0.68% vs D0), mean mass flow 0.174113 kg s ⁻¹ (+0.45%); C2 loss +2.89% vs D0.
Final matched S2N+ external	not exported	not exported	0.5614	-1.9018	Vehicle compatibility baseline	Later matched whole-vehicle coefficient case; separate from earlier inlet-pressure pilot.
Final matched D9 external	not exported	not exported	0.5524	-1.9756	Vehicle compatibility check	Vehicle CD - 1.61% and airbox CD - 8.13% vs matched S2N+ case; no D9 inlet-mouth pressure in supplied final report.

Appendix E: Internal steady CFD output data

Table E1. Internal steady CFD output data for the old, updated and design airboxes.

Case	Delta p _t [Pa]	K _t [% of 4982 Pa]	Mass flow [kg s ⁻¹]	p _{t,in} [Pa]	p _{t,out} [Pa]	Velocity in/out	Tap comparison	Notes
Tap-C1	511.5	10.27	0.160784	0.000	-511.5	12.60 / 84.76	r=0.844; RMSE=61.1 Pa	Tapped correlation model
Tap-C2	464.4	9.32	0.160695	0.000	-464.4	12.59 / 85.18	r=0.742; RMSE=66.2 Pa	Tapped correlation model
Tap-C3	430.3	8.64	0.161800	0.000	-430.3	12.69 / 85.50	r=0.866; RMSE=76.6 Pa	Tapped correlation model
Tap-C4	422.8	8.49	0.163021	0.000	-422.8	12.79 / 85.58	r=0.742; RMSE=92.1 Pa	Tapped correlation model
Old-C1	729.0	14.63	0.155772	0.000	-729.0	14.25 / 82.72	n/a	Matched old-airbox case
Old-C2	782.2	15.70	0.154428	0.000	-782.2	14.11 / 82.24	n/a	Matched old-airbox case
Old-C3	571.1	11.46	0.158629	0.000	-571.1	14.50 / 84.12	n/a	Matched old-airbox case
Old-C4	700.8	14.07	0.154796	0.000	-700.8	14.14 / 82.89	n/a	Matched old-airbox case
D0-C1	454.9	9.13	0.161456	0.000	-454.9	12.84 / 85.25	n/a	Clean design-study baseline
D0-C2	461.7	9.27	0.160962	0.000	-461.7	12.80 / 85.23	n/a	Clean design-study baseline
D0-C3	425.0	8.53	0.161715	0.000	-425.0	12.85 / 85.52	n/a	Clean design-study baseline
D0-C4	405.4	8.14	0.162691	0.000	-405.4	12.93 / 85.72	n/a	Clean design-study baseline
D1-C1	444.6	8.92	0.161591	0.000	-444.6	12.87 / n/a	n/a	First reprofiled inner-transition concept
D1-C2	459.3	9.22	0.161100	0.000	-459.3	12.83 / n/a	n/a	First reprofiled inner-transition concept
D1-C3	416.2	8.35	0.162072	0.000	-416.2	12.91 / n/a	n/a	First reprofiled inner-transition concept
D1-C4	398.8	8.01	0.162656	0.000	-398.8	12.95 / n/a	n/a	First reprofiled inner-transition concept

Case	Delta p _t [Pa]	K _t [% of 4982 Pa]	Mass flow [kg s ⁻¹]	p _{t,in} [Pa]	p _{t,out} [Pa]	Velocity in/out	Tap comparison	Notes
D2-C1	499.3	10.02	0.161404	0.000	-499.3	12.86 / 84.80	n/a	Further lower-half curvature concept
D2-C2	502.1	10.08	0.160869	0.000	-502.1	12.79 / 84.86	n/a	Further lower-half curvature concept
D2-C3	409.7	8.22	0.163130	0.000	-409.7	13.00 / 85.68	n/a	Further lower-half curvature concept
D2-C4	404.5	8.12	0.162958	0.000	-404.5	12.98 / 85.71	n/a	Further lower-half curvature concept
D3-C1	478.8	9.61	0.161831	0.000	-478.8	12.89 / 85.05	n/a	Alternative lower-half inlet-wall profile
D3-C2	485.1	9.74	0.161240	0.000	-485.1	12.84 / 84.95	n/a	Alternative lower-half inlet-wall profile
D3-C3	426.5	8.56	0.162424	0.000	-426.5	12.93 / 85.56	n/a	Alternative lower-half inlet-wall profile
D3-C4	403.4	8.10	0.163173	0.000	-403.4	13.00 / 85.67	n/a	Alternative lower-half inlet-wall profile
D4-C1	473.9	9.51	0.160842	0.000	-473.9	13.38 / 85.17	n/a	Inlet Shape 1; nominally area-matched opening
D4-C2	421.8	8.47	0.162775	0.000	-421.8	13.54 / 85.58	n/a	Inlet Shape 1; nominally area-matched opening
D4-C3	403.1	8.09	0.162717	0.000	-403.1	13.54 / 85.77	n/a	Inlet Shape 1; nominally area-matched opening
D4-C4	393.9	7.91	0.163342	0.000	-393.9	13.59 / 85.84	n/a	Inlet Shape 1; nominally area-matched opening
D5-C1	471.3	9.46	0.161764	0.000	-471.3	13.48 / 85.14	n/a	Inlet Shape 2; nominally area-matched opening

Case	Delta p _t [Pa]	K _t [% of 4982 Pa]	Mass flow [kg s ⁻¹]	p _{t,in} [Pa]	p _{t,out} [Pa]	Velocity in/out	Tap comparison	Notes
D5-C2	430.0	8.63	0.162307	0.000	-430.0	13.51 / 85.45	n/a	Inlet Shape 2; nominally area-matched opening
D5-C3	405.8	8.15	0.163234	0.000	-405.8	13.61 / 85.70	n/a	Inlet Shape 2; nominally area-matched opening
D5-C4	403.1	8.09	0.163483	0.000	-403.1	13.60 / 85.76	n/a	Inlet Shape 2; nominally area-matched opening
D0+Ext-C1	496.3	9.96	0.173159	728.92	232.6	13.77 / 91.50	n/a	Clean D0 vehicle-pressure reference
D0+Ext-C2	505.9	10.16	0.172385	728.92	223.0	13.70 / 91.44	n/a	Clean D0 vehicle-pressure reference
D0+Ext-C3	468.4	9.40	0.173483	728.92	260.6	13.79 / 91.74	n/a	Clean D0 vehicle-pressure reference
D0+Ext-C4	457.7	9.19	0.174296	728.92	271.2	13.86 / 91.84	n/a	Clean D0 vehicle-pressure reference
D6-C1	459.6	9.22	0.161981	0.000	-459.6	15.38 / 85.21	n/a	D6 lipless size-study reference
D6-C2	457.5	9.18	0.161572	0.000	-457.5	15.34 / 85.30	n/a	D6 lipless size-study reference
D6-C3	411.5	8.26	0.162363	0.000	-411.5	15.40 / 85.71	n/a	D6 lipless size-study reference
D6-C4	401.8	8.07	0.162798	0.000	-401.8	15.45 / 85.76	n/a	D6 lipless size-study reference
D7-C1	455.0	9.13	0.162001	0.000	-455.0	12.92 / 85.31	n/a	D7 +5 mm inlet-size case
D7-C2	459.2	9.22	0.161758	0.000	-459.2	12.90 / 85.28	n/a	D7 +5 mm inlet-size case
D7-C3	405.5	8.14	0.163208	0.000	-405.5	13.02 / 85.71	n/a	D7 +5 mm inlet-size case
D7-C4	405.3	8.13	0.163252	0.000	-405.3	13.02 / 85.74	n/a	D7 +5 mm inlet-size case

Case	Delta p _t [Pa]	K _t [% of 4982 Pa]	Mass flow [kg s ⁻¹]	p _{t,in} [Pa]	p _{t,out} [Pa]	Velocity in/out	Tap comparison	Notes
D8-C1	435.8	8.75	0.162034	0.000	-435.8	10.42 / 85.44	n/a	D8 +10 mm inlet-size case
D8-C2	458.6	9.21	0.161900	0.000	-458.6	10.40 / 85.32	n/a	D8 +10 mm inlet-size case
D8-C3	396.0	7.95	0.163331	0.000	-396.0	10.51 / 85.80	n/a	D8 +10 mm inlet-size case
D8-C4	395.0	7.93	0.163176	0.000	-395.0	10.49 / 85.80	n/a	D8 +10 mm inlet-size case
D9-C1	444.4	8.92	0.161900	0.000	-444.4	14.98 / 85.39	n/a	D9 -50 mm inlet-transition length
D9-C2	419.8	8.43	0.162363	0.000	-419.8	15.01 / n/a	n/a	D9 -50 mm inlet-transition length
D9-C3	408.5	8.20	0.162761	0.000	-408.5	15.05 / 85.72	n/a	D9 -50 mm inlet-transition length
D9-C4	386.2	7.75	0.163507	0.000	-386.2	15.13 / 85.85	n/a	D9 -50 mm inlet-transition length
D10-C1	481.7	9.67	0.159898	0.000	-481.7	15.16 / n/a	n/a	D10 +50 mm inlet-transition length
D10-C2	461.2	9.26	0.160931	0.000	-461.2	15.29 / 85.25	n/a	D10 +50 mm inlet-transition length
D10-C3	401.0	8.05	0.163102	0.000	-401.0	15.47 / n/a	n/a	D10 +50 mm inlet-transition length
D10-C4	386.6	7.76	0.162552	0.000	-386.6	15.42 / 85.98	n/a	D10 +50 mm inlet-transition length
D11-C1	438.0	8.79	0.161660	0.000	-438.0	14.48 / 85.45	n/a	D11 -75 mm transition length
D11-C2	454.0	9.11	0.161842	0.000	-454.0	14.49 / 85.28	n/a	D11 -75 mm transition length
D11-C3	402.3	8.08	0.162737	0.000	-402.3	14.58 / 85.71	n/a	D11 -75 mm transition length
D11-C4	416.1	8.35	0.162702	0.000	-416.1	14.56 / 85.66	n/a	D11 -75 mm transition length

Case	Delta p _t [Pa]	K _t [% of 4982 Pa]	Mass flow [kg s ⁻¹]	p _{t,in} [Pa]	p _{t,out} [Pa]	Velocity in/out	Tap comparison	Notes
D12-C1	448.3	9.00	0.162070	0.000	-448.3	15.37 / 85.40	n/a	D12 +50 mm trumpet-tray extension
D12-C2	436.2	8.75	0.162651	0.000	-436.2	15.44 / 85.43	n/a	D12 +50 mm trumpet-tray extension
D12-C3	409.2	8.21	0.162927	0.000	-409.2	15.46 / 85.68	n/a	D12 +50 mm trumpet-tray extension
D12-C4	402.0	8.07	0.163290	0.000	-402.0	15.48 / 85.74	n/a	D12 +50 mm trumpet-tray extension
D13-C1	452.8	9.09	0.161566	0.000	-452.8	15.32 / 85.30	n/a	D13 C2/C3 central divider
D13-C2	449.7	9.03	0.161867	0.000	-449.7	15.36 / 85.29	n/a	D13 C2/C3 central divider
D13-C3	411.0	8.25	0.161897	0.000	-411.0	15.36 / n/a	n/a	D13 divider; outlet velocity invalid in supplied export
D13-C4	403.1	8.09	0.163035	0.000	-403.1	15.47 / 85.72	n/a	D13 C2/C3 central divider
D14-C1	517.7	10.39	0.160930	0.000	-517.7	15.25 / 84.68	n/a	D14 +100 mm trumpet-tray extension
D14-C2	453.6	9.10	0.162347	0.000	-453.6	15.40 / 85.28	n/a	D14 +100 mm trumpet-tray extension
D14-C3	417.8	8.39	0.163140	0.000	-417.8	15.47 / 85.59	n/a	D14 +100 mm trumpet-tray extension
D14-C4	416.5	8.36	0.163312	0.000	-416.5	15.47 / 85.57	n/a	D14 +100 mm trumpet-tray extension
D15-C1	506.0	10.16	0.161412	0.000	-506.0	14.95 / 84.79	n/a	D15 = D9 shortened inlet + D12 +50 mm tray
D15-C2	460.0	9.23	0.162203	0.000	-460.0	15.04 / 85.20	n/a	D15 = D9 shortened inlet + D12 +50 mm tray
D15-C3	420.3	8.44	0.162984	0.000	-420.3	15.10 / 85.56	n/a	D15 = D9 shortened inlet + D12 +50 mm tray

Case	Delta p _t [Pa]	K _t [% of 4982 Pa]	Mass flow [kg s ⁻¹]	p _{t,in} [Pa]	p _{t,out} [Pa]	Velocity in/out	Tap comparison	Notes
D15-C4	424.5	8.52	0.162993	0.000	-424.5	15.09 / 85.56	n/a	D15 = D9 shortened inlet + D12 +50 mm tray

Appendix E.1 Matched airbox mesh methodology and numerical checks

The pressure-tapped updated and old production models use matched sizing logic but are different geometries and meshes; this demonstrates a controlled meshing method, not an airbox mesh-sensitivity study. Clean D0 and the design concepts omit the correlation-only tap-face refinement and use the same clean model-preparation logic. No formal airbox mesh-sensitivity claim is made because a matched coarse/medium/fine airbox refinement study was not completed.

Table E2. Matched airbox mesh definitions and numerical checks.

Airbox	Cell count	Max volume	Trumpet surface	Trumpet BOI	Min. orthogonal quality	Conclusion
Tapped updated correlation model	4,064,665	6 mm	2 mm	3 mm	about 0.050	Correlation mesh with 0.5 mm tap-face refinement.
Clean D0	Not recorded in retained rerun export	6 mm	2 mm	3 mm	Not recorded in retained rerun export	Design-study baseline; tap-face refinement omitted.
Old	4,386,112	6 mm	2 mm	3 mm	about 0.050	Matched old-airbox production mesh.
Common controls	-	3 mm main surface	-	8 prism layers	0.05 mm first layer	Matched baseline sizing logic; inlet-shape models additionally use 1 mm inlet-face sizing and a 3 mm inlet-region BOI.
D4/D5 inlet-shape models	D4 representative: 7,845,022	6 mm	2 mm	3 mm	0.050 (D4 representative)	Common 1 mm inlet-face + 3 mm inlet-region refinement; D0 not rerun with this added local refinement.

Appendix E.2 Boundary-condition and configuration comparison

Table E3. Boundary-condition and configuration comparison summary.

Comparison	Outlet velocity / mass-flow change [%]	Total-pressure-loss change [%]	Tap metric	Engineering interpretation
Clean D0 relative to old, Cylinders 1-4	Velocity mean +2.94%; mass-flow mean +3.72%.	C1 -37.6; C2 -41.0; C3 -25.6; C4 -42.1; mean -37.2%.	n/a	Clean updated airbox improves flow delivery and pressure preservation in all four matched path cases.

Comparison	Outlet velocity / mass-flow change [%]	Total-pressure-loss change [%]	Tap metric	Engineering interpretation
Tapped correlation model -> clean D0	Mean velocity +0.20%; mean mass flow +0.08%.	Mean loss -4.48%; largest local change at C1.	Tap model $r = 0.742$ -0.866 retained only for correlation.	Bulk flow is insensitive, but total-pressure loss is more sensitive to small model-preparation changes.
Four-cylinder case spread	Mass-flow CV: old 1.22%, D0 0.45%, D1 0.41%, D2 0.69%, D3 0.51%, D4 0.67%, D5 0.49%, D6 0.32%, D7 0.48%, D8 0.46%, D9 0.42%, D10 0.91%, D11 0.35%, D12 0.32%, D13 0.40%, D14 0.67%, D15 0.46%.	Loss CV: old 12.89%, D0 6.01%, D1 6.35%, D2 11.91%, D3 8.89%, D4 8.45%, D5 7.38%, D6 6.99%, D7 6.93%, D8 7.42%, D9 5.84%, D10 10.63%, D11 5.36%, D12 5.17%, D13 6.00%, D14 10.51%, D15 8.78%.	n/a	D12 gives the lowest loss CV among the tray cases, but D9 gives the strongest standalone mean/worst loss result. D15 worsens both mean loss and consistency, so D9 is retained.
D1 relative to clean D0	Mean mass flow +0.09%.	Mean loss -1.61%; worst loss -0.51%.	n/a	Favourable but marginal; not larger than the observed model-preparation sensitivity.
D2 relative to clean D0	Mean mass flow +0.24%.	Mean loss +3.93%; worst loss +8.75%.	n/a	Rejected: small flow increase with higher and less uniform loss.
D3 relative to clean D0	Mean mass flow +0.29%; mean outlet velocity -0.14%.	Mean loss +2.68%; worst loss +5.06%.	n/a	Rejected: alternative lower-wall profile retains the C1/C2 loss penalty and does not reproduce D1's small favourable trend.
Clean D0 +728.9 Pa sensitivity	Mean outlet velocity +7.26%; mean mass flow +7.19%; mass-flow CV 0.45% to 0.46%.	Mean loss +10.38% at the higher-flow CFD state; loss CV 6.02% to 4.72%.	n/a	Matched clean reference confirms higher CFD flow without material deterioration of steady path balance; not an engine operating-point prediction.
D4 relative to clean D0	Mean velocity +0.19%; mean mass flow +0.44%.	Mean loss -3.11%; worst loss +2.65%; C1 +4.15%, C2 -8.64%, C3 -5.15%, C4 -2.85%.	n/a	Best mean-loss inlet shape, but rejected as a final design because the limiting C1/worst path deteriorates.
D5 relative to clean D0	Mean velocity +0.10%; mean mass flow +0.61%.	Mean loss -2.11%; worst loss +2.07%; C1 +3.59%, C2 -6.87%, C3 -4.52%, C4 -0.58%.	n/a	Smaller mean benefit than D4; slightly better balance, but C1/worst path still deteriorates.
D6 relative to clean D0	Mean velocity +0.07%; mean mass flow +0.29%.	Mean loss -0.96%; worst loss -0.46%; C1 +1.03%, C2 -0.92%, C3 -3.19%, C4 -0.89%.	n/a	Lip removal has only a small integrated effect; D6 is retained as the size-study reference rather than an aerodynamic selection.

Comparison	Outlet velocity / mass-flow change [%]	Total-pressure-loss change [%]	Tap metric	Engineering interpretation
D7 relative to D6	Mean velocity +0.02%; mean mass flow +0.23%.	Mean loss -0.31%; worst loss -0.09%; C1 -1.01%, C2 +0.37%, C3 -1.44%, C4 +0.86%.	n/a	First size increase provides negligible additional benefit and slightly penalises C2/C4.
D8 relative to D6 / clean D0	Mean velocity +0.12% vs D6; mean mass flow +0.27% vs D6 (+0.56% vs D0).	Mean loss -2.59% vs D6 and -3.53% vs D0; D8 vs D0: C1 -4.17%, C2 -0.67%, C3 -6.81%, C4 -2.57%; worst loss -0.67% vs D0.	n/a	Preferred size-study candidate. Unlike D4/D5, every D8 path is below its D0 loss, but the gain remains below the 4.48% model-preparation sensitivity.
D9 relative to D6	Mean mass flow +0.28%.	Mean loss -4.10%; worst loss -3.31%; C1 -3.31%, C2 -8.24%, C3 -0.72%, C4 -3.89%; loss CV 6.99% to 5.84%.	n/a	Preferred inlet-transition-length case; all four paths improve relative to D6.
D10 relative to D6	Mean mass flow -0.34%.	Mean loss +0.01%; worst loss +4.80%; C1 +4.80%, C2 +0.82%, C3 -2.54%, C4 -3.79%; loss CV 6.99% to 10.63%.	n/a	Rejected: longer transition redistributes loss toward C1/C2 without improving the mean.
D11 relative to D6	Mean mass flow +0.04%.	Mean loss -1.15%; worst loss -1.21%; C1 -4.70%, C2 -0.76%, C3 -2.23%, C4 +3.57%; loss CV 6.99% to 5.36%.	n/a	Further shortening improves consistency but erodes most of D9's mean-loss benefit; D9 remains the preferred tested length.
D12 relative to D6	Mean mass flow +0.34%; mass-flow CV 0.32% to 0.32%.	Mean loss -2.01%; worst loss -2.46%; C1 -2.46%, C2 -4.67%, C3 -0.56%, C4 +0.06%; loss CV 6.99% to 5.17%.	n/a	Preferred trumpet-tray extension. Small matched-study gain with improved loss consistency; selected for D15.
D13 relative to D6	Mean mass flow -0.05%; mass-flow CV 0.32% to 0.40%.	Mean loss -0.80%; worst loss -1.48%; C1 -1.48%, C2 -1.71%, C3 -0.11%, C4 +0.31%; loss CV 6.99% to 6.00%.	n/a	Marginal redistribution benefit only; central divider is not carried forward.
D14 relative to D6	Mean mass flow +0.16%; mass-flow CV 0.32% to 0.67%.	Mean loss +4.35%; worst loss +12.65%; C1 +12.65%, C2 -0.86%, C3 +1.54%, C4 +3.65%; loss CV 6.99% to 10.51%.	n/a	Rejected: +100 mm extension reverses the D12 benefit and strongly penalises C1/path consistency.
D15 relative to D9 / D12	Mean mass flow -0.14% vs D9 and -0.21% vs D12; mass-flow CV 0.46%.	Mean loss +9.16% vs D9 and +6.79% vs D12; worst loss +13.87% vs D9 and +12.88% vs D12; loss CV 8.78%.	n/a	Rejected: independent gains are not additive. D9 remains the final controlled-design selection.

Appendix E.3 Final evidence summary

Table E4. Final evidence and uncertainty summary.

Evidence item	Key value	Uncertainty or sensitivity	Conclusion supported
Pipe validation	Full 5–25 inH ₂ O M3/L8/SST sweep: $r = 0.897\text{--}0.943$, RMSE = 0.234–0.946 kPa, normalised RMSE = 10.8–14.4%; all measured and predicted minima occurred at Tap 12. L8–L12 pressure RMSE was 0.096 kPa and mass-flow difference was 0.016%.	Single stabilised bench readings; Tap-1 alignment removes constant datum offset; largest Tap 9–10 discrepancy was 0.584 kPa at 20 inH ₂ O.	M3 with eight prism layers and SST $k\text{--}\omega$ captures the wall-static-pressure redistribution consistently across the tested range and is retained for comparative airbox application.
External inlet condition	Earlier pilot: old p_t 792.6 Pa; updated p_t 728.9 Pa. At +728.92 Pa, D0 mean $\dot{m}$ 0.173331 kg s ⁻¹ and mean loss 482.1 Pa; D9 mean $\dot{m}$ 0.174113 kg s ⁻¹ (+0.45%) and mean loss 478.8 Pa (-0.68%), with C2 loss +2.89%. Later matched external case: vehicle CD 0.5614 (S2N+) to 0.5524 (D9); airbox CD 0.00846 to 0.00777.	Uniform pressure transfer; no engine through-flow or measured inlet-pressure validation. The later external export did not include D9 inlet-mouth p_s or p_t , so it cannot redefine the +728.92 Pa boundary.	Use 0 Pa for controlled ranking. D9 retains a small mean advantage at +728.92 Pa but with a C2 penalty; the final external case supports vehicle compatibility only.
Updated-airbox pressure-tap test	Mean pressures: C1 -43.94 Pa; C2 -39.44 Pa; C3 -32.56 Pa; C4 -22.15 Pa.	Three valve-open repeats per cylinder; one ambient reference per set.	Repeatable cylinder-dependent wall-pressure response suitable for CFD spatial correlation.
Configuration comparison	Clean D0 vs old: mean outlet velocity +2.94%; mean mass flow +3.72%; mean total-pressure loss -37.2%; velocity CV 0.97% to 0.27%; mass-flow CV 1.22% to 0.45%.	Tapped-to-clean preparation changes mean mass flow by +0.08% but mean loss by -4.48%; no dedicated airbox coarse/medium/fine sensitivity yet.	Clean D0 retained as the design baseline; tap removal and the 0.5 mm correction remain inseparable within the present model-preparation sensitivity check.
CFD-to-airbox-bench comparison	C1–C4: MAE 51.0, 58.5, 70.3 and 87.7 Pa; RMSE 61.1, 66.2, 76.6 and 92.1 Pa; $r = 0.844, 0.742, 0.866$ and 0.742 . Regional means (bench/CFD): inlet -35.0/-89.3 Pa; transition -38.6/-125.3 Pa; plenum -26.8/-74.8 Pa.	The runner-end CFD pressure outlet is not the measured valve/head depression plane; one ambient reference was captured per cylinder set.	CFD reproduces the broad transition-suction and plenum-recovery pattern, but not the absolute magnitude or every local tap value.

Evidence item	Key value	Uncertainty or sensitivity	Conclusion supported
Design decision	Wall study: D1 -1.61% mean loss; D2 +3.93%; D3 +2.68%. Inlet shape: D4/D5 reduce mean loss but worsen the limiting path. Inlet size: D8 -2.59% vs D6. Transition: D9 -4.10% mean/-3.31% worst vs D6; D11 confirms that further shortening did not continue the D9 improvement. Divider: D13 -0.80%. Tray: D12 +50 mm -2.01% mean/-2.46% worst; D14 +100 mm +4.35% mean/+12.65% worst. Combined D15: 452.7 Pa mean loss, +9.16% vs D9; 506.0 Pa worst loss, +13.87% vs D9. Vehicle-pressure verification: D9 vs D0 at +728.92 Pa gives mean loss -0.68%, mean m-dot +0.45%, but C2 loss +2.89%.	Favourable few-percent standalone changes remain close to or below the 4.48% tapped-to-clean model-preparation change. D15 deterioration is much larger and reverses the direction of both parent gains, so rejection is robust within the present comparative evidence. A representative matched refinement is still required before claiming D9's few-percent benefit as numerically resolved.	D15 is rejected and D9 remains the final controlled-design selection. The +728.92 Pa internal robustness comparison and later matched whole-vehicle coefficient check are complete; a matched airbox refinement and physical prototype test remain the principal verification priorities.

Appendix F: CFD completion checklist

- Geometry repaired and fluid volumes checked.
- Named boundaries and coordinate directions confirmed.
- Pressure reference and sign convention recorded.
- Fluid properties and operating conditions recorded.
- Mesh quality, inflation layers and y^+ checked.
- Residuals, engineering monitors and mass balance converged.
- Pipe mesh sensitivity completed; matched airbox meshing controls checked. No formal airbox mesh-sensitivity claim is made; a representative matched refinement remains recommended future work.
- Limited external inlet-condition assessment completed; old and updated inlet-mouth pressures were recorded from the earlier pilot, and clean D0 plus selected D9 C1-C4 were completed at 728.92 Pa inlet total gauge pressure as the matched robustness comparison. Because the external model contains no engine through-flow, this value is treated as a sensitivity condition rather than an installed-engine operating pressure. Wall-profile D1-D3, inlet-shape D4-D5, inlet-size D6-D8, transition-length D9-D11, divider D13, trumpet-tray D12/D14 and combined D15 screening are complete at the controlled 0 Pa inlet condition. D15 is rejected because it is worse than both parent concepts; D9 is the final controlled-design selection. The later matched S2N+/D9 whole-vehicle coefficient comparison is also complete, although its supplied export does not include a new D9 inlet-mouth pressure.
- Figures exported with readable legends, units and consistent scales.
- All results tables independently checked against exported data.

- Validation, correlation and verification terminology used correctly.

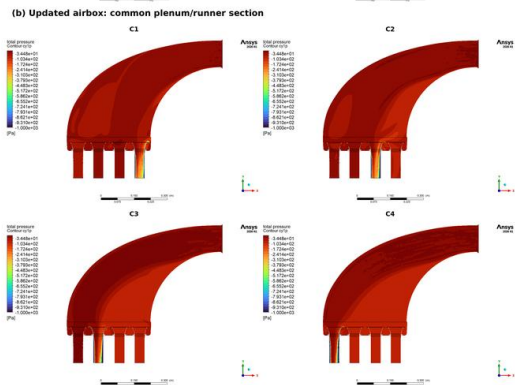
Appendix G: Supporting airbox design figures

This appendix contains supporting flow-field visualisations and quantitative charts that are not required for the main design decision. To reduce repetition, D0-D5 are presented as grouped design plates rather than as separate total-pressure, velocity, streamline and vector figures. The same grouped format is retained for D6-D15. Main-text figures remain the primary evidence where they are directly discussed.

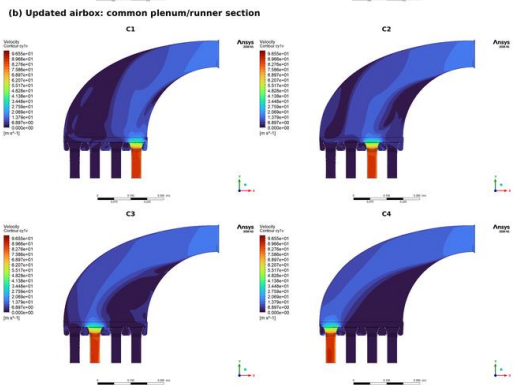
S2N+ updated-airbox baseline

Grouped supporting fields; quantitative design reference remains clean D0.

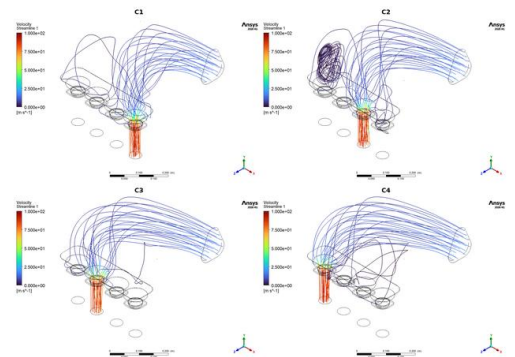
Total pressure



Velocity



Streamlines (clean D0)



Velocity vectors

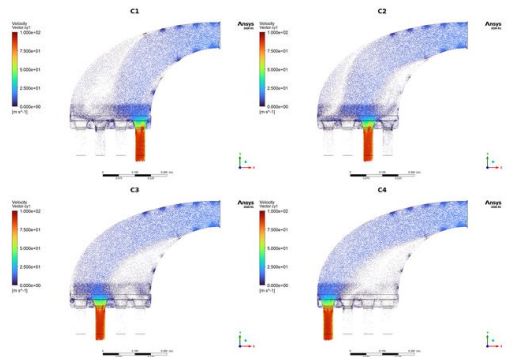
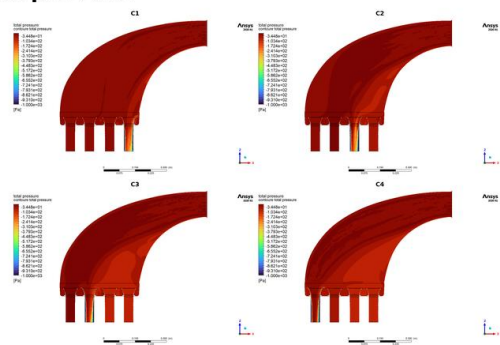


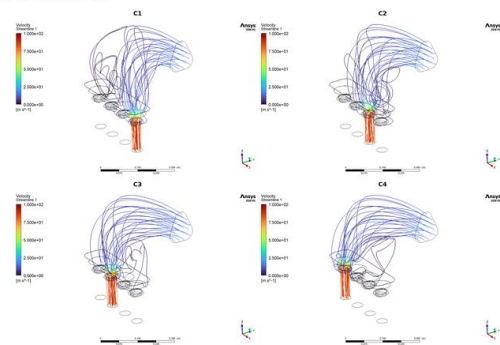
Figure G1. S2N+ updated-airbox supplementary CFD fields. Total-pressure and velocity panels use the pressure-tapped correlation model; the streamline panel uses clean D0 at the controlled 0 Pa inlet condition.

Design 1 wall-profile concept

Total pressure



Streamlines



Velocity vectors

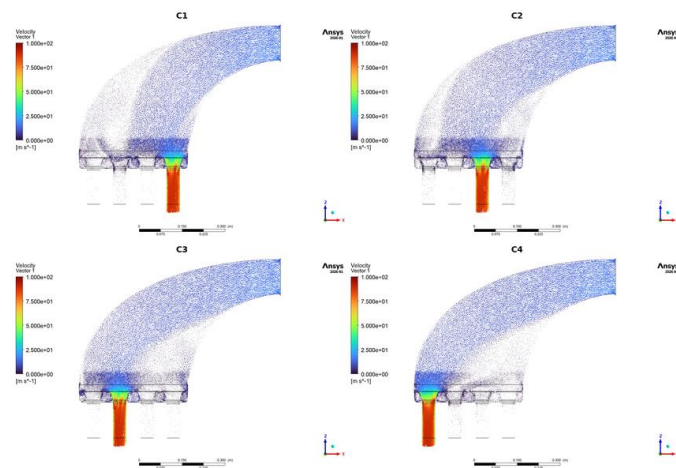
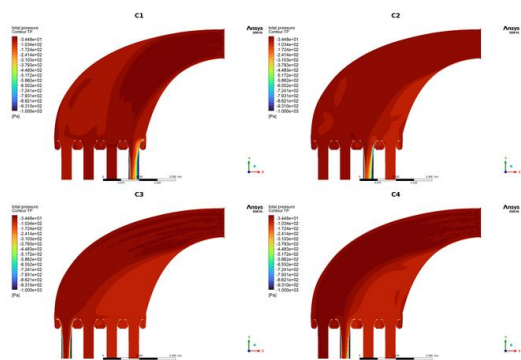


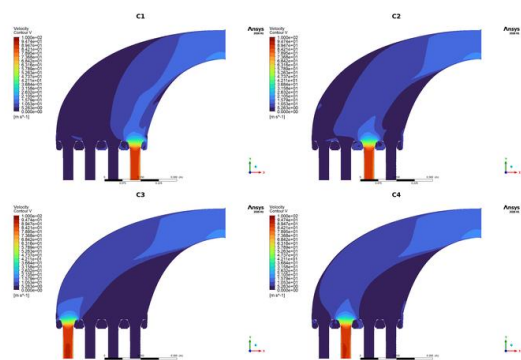
Figure G2. D1 supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, streamlines and velocity vectors.

Design 2 supplementary field plots

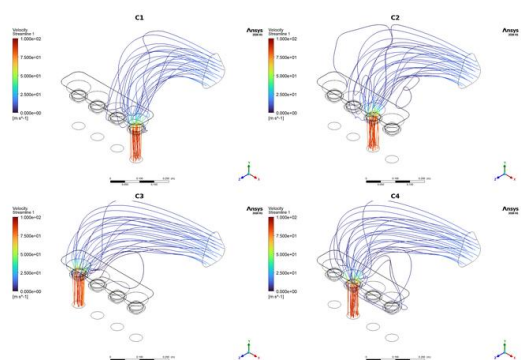
Total pressure



Velocity



Streamlines



Velocity vectors

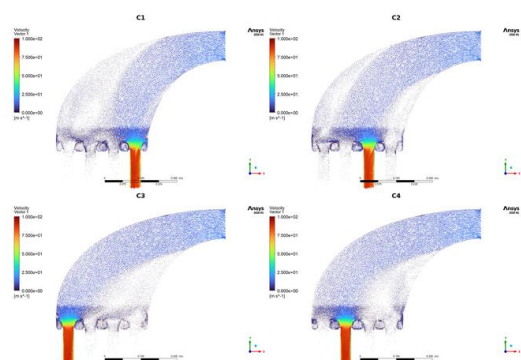
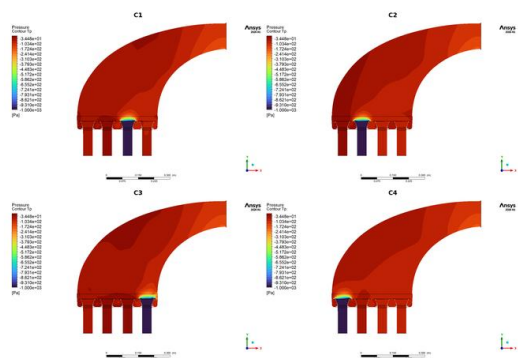


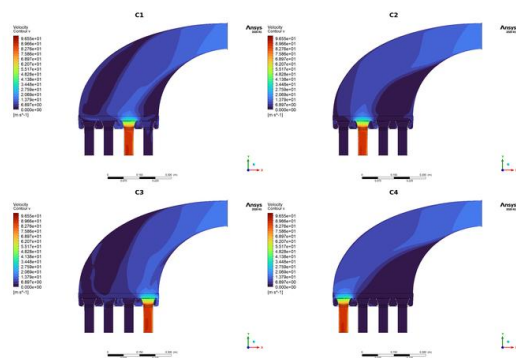
Figure G3. D2 supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 3 supplementary field plots

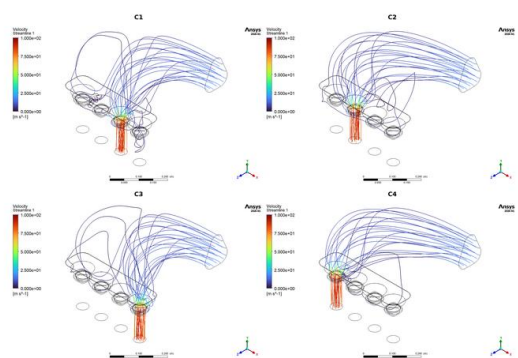
Total pressure



Velocity



Streamlines



Velocity vectors

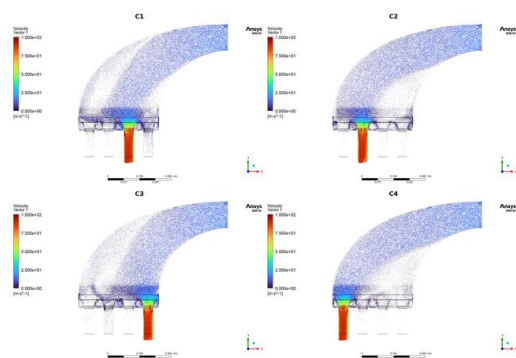
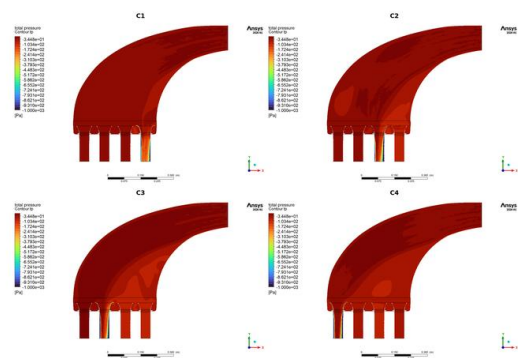


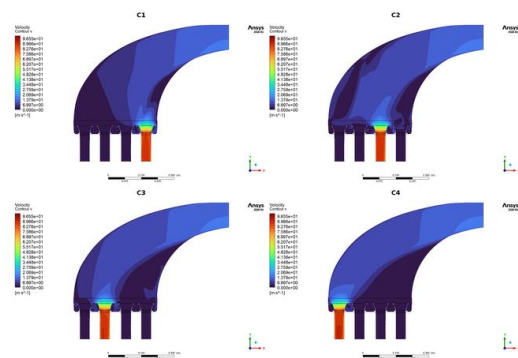
Figure G4. D3 supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 4 supplementary field plots

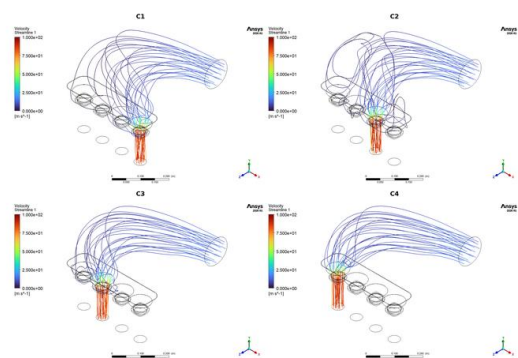
Total pressure



Velocity



Streamlines



Velocity vectors

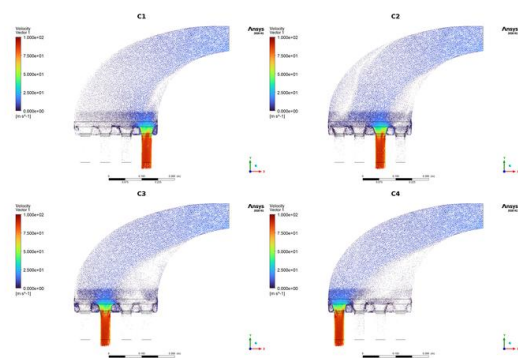
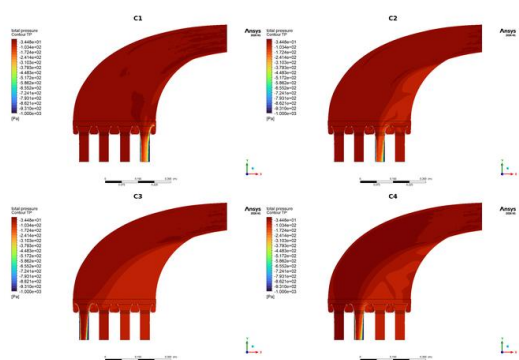


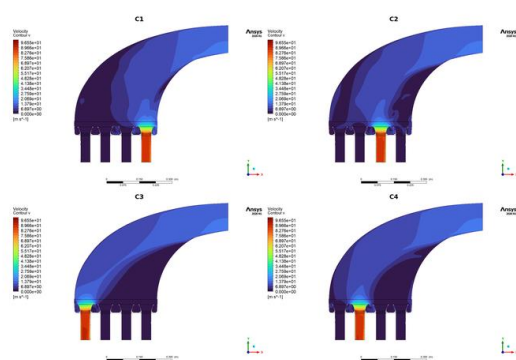
Figure G5. D4 supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 5 supplementary field plots

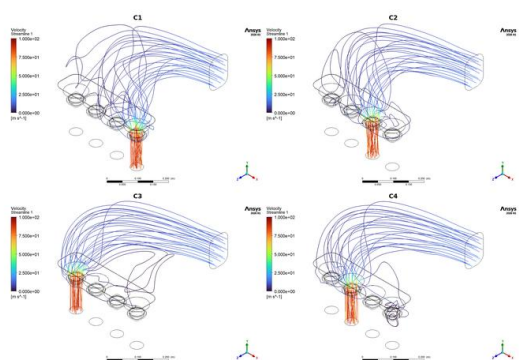
Total pressure



Velocity



Streamlines



Velocity vectors

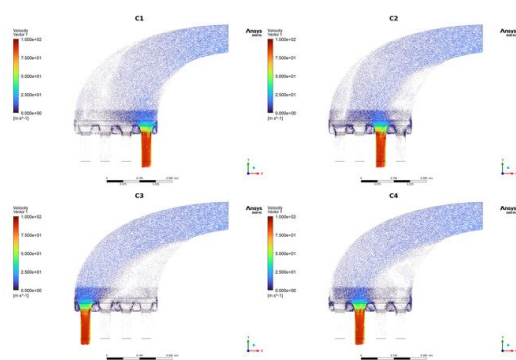


Figure G6. D5 supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Supporting design-study mass-flow charts

The following charts retain the small mass-flow changes that accompany the pressure-loss screening results. They are kept separate because they are quantitative comparison plots rather than repeated flow-field images.

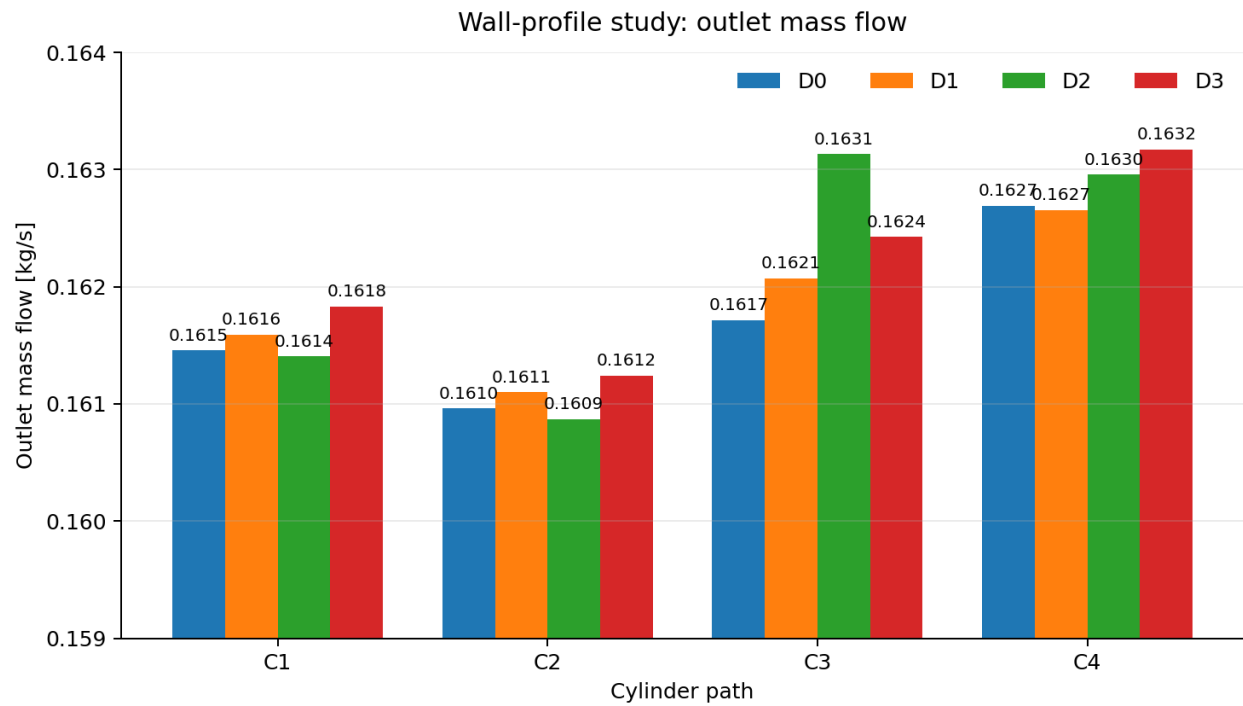


Figure G7. Clean D0, D1, D2 and D3 outlet mass flow for the four matched steady cylinder paths.

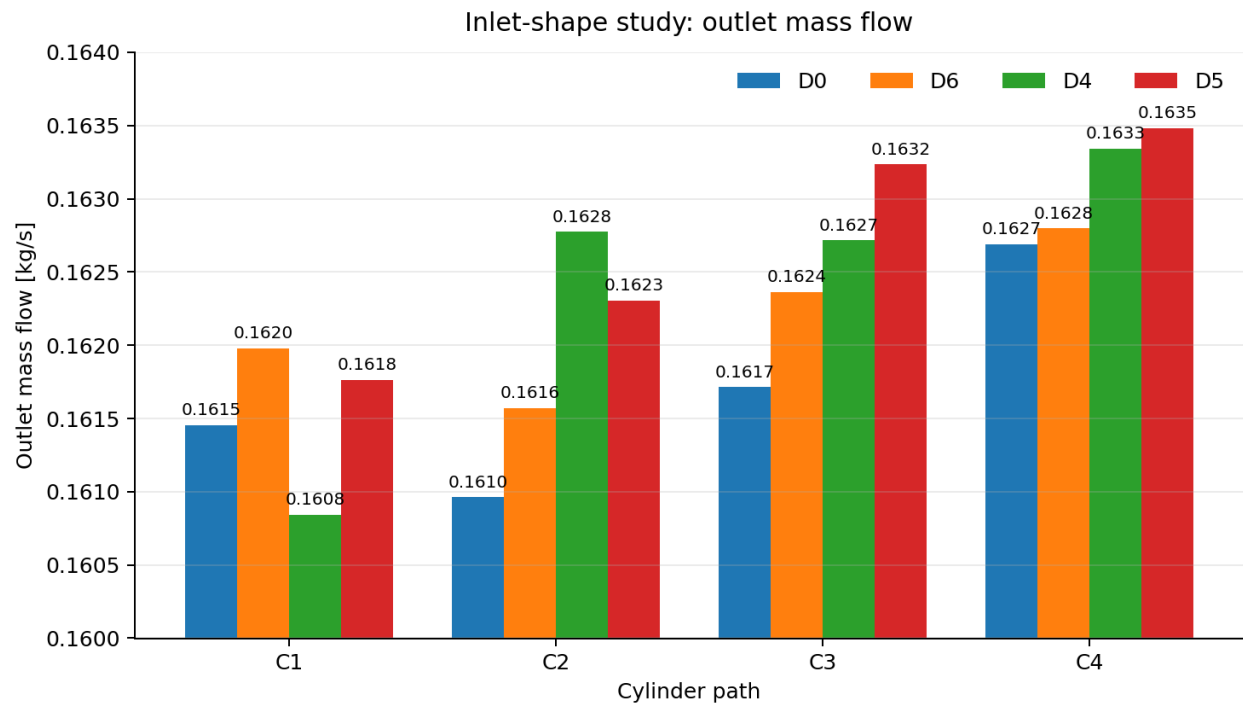


Figure G8. Clean D0, lipless D6, D4 and D5 outlet mass flow for the four matched steady cylinder paths.

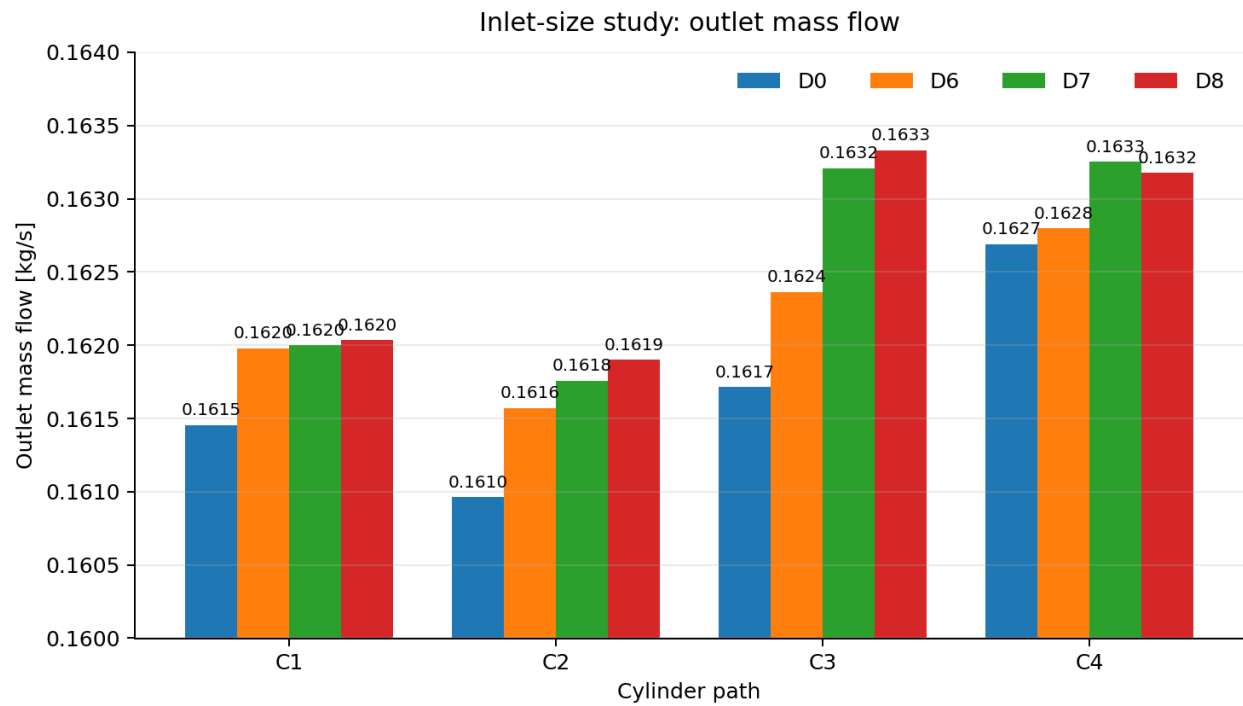


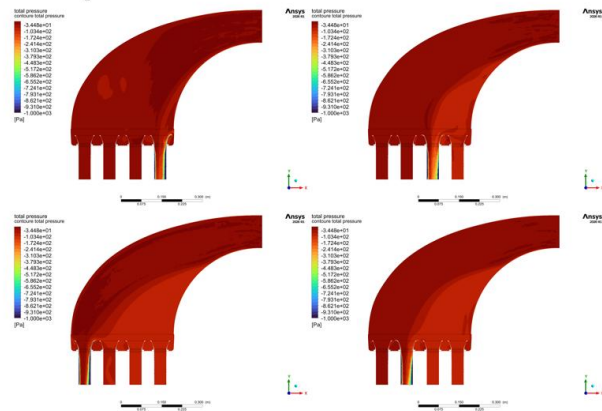
Figure G9. Clean D0, D6, D7 and D8 outlet mass flow for the four matched steady cylinder paths.

[Additional grouped CFD field-plot archive](#)

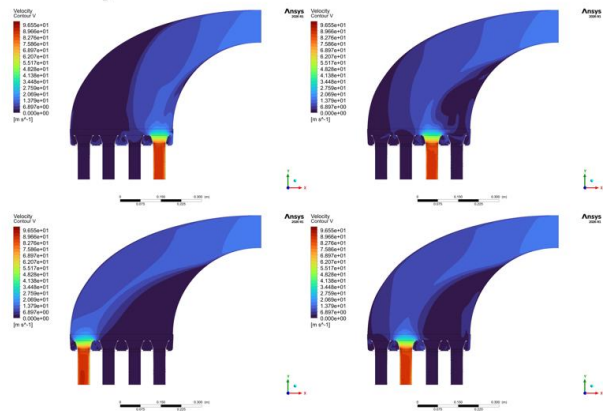
The following grouped image plates contain supplementary total-pressure, velocity, inlet-seeded streamline and velocity-vector fields. Keeping one plate per design preserves the CFD evidence trail without repeating each field type as a separate appendix figure.

Design 6 lipless-reference

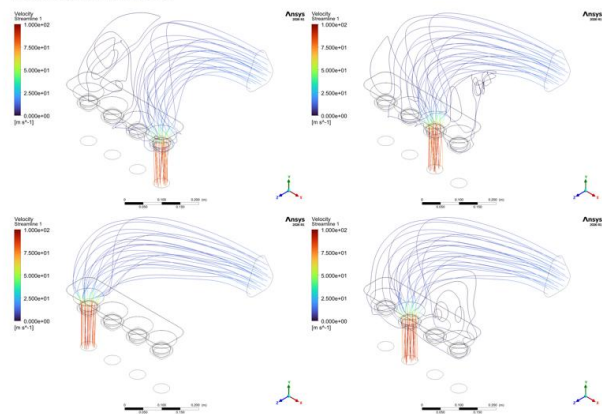
Total pressure



Velocity



Streamlines



Vectors

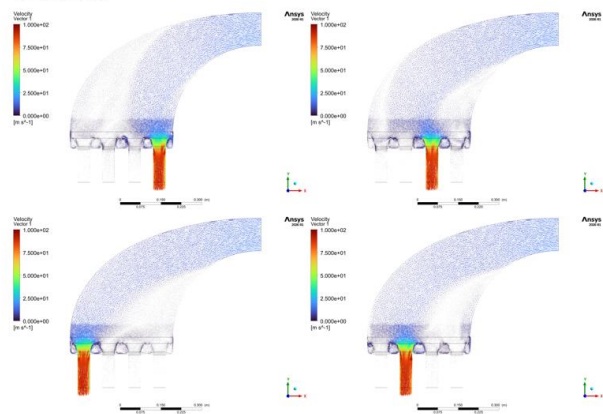
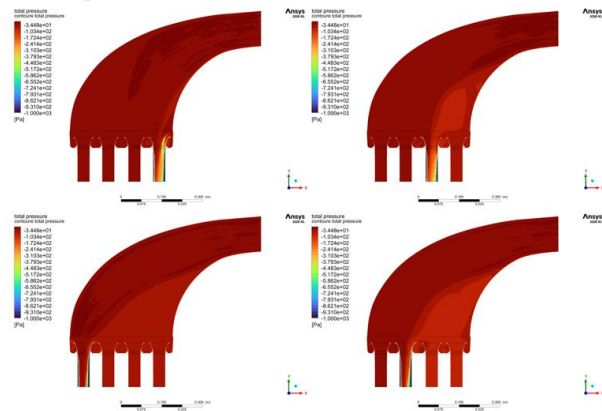


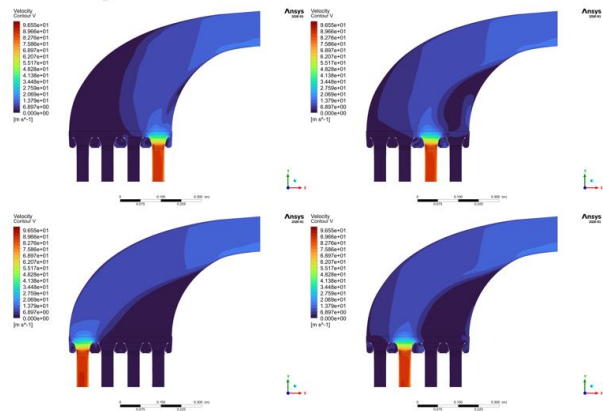
Figure G10. D6 lipless-reference supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 7 +5 mm enlarged-inlet

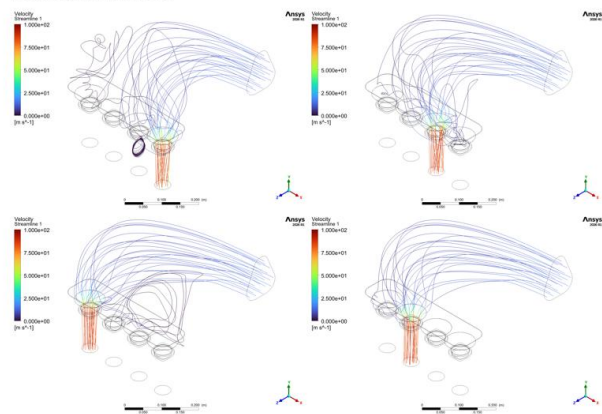
Total pressure



Velocity



Streamlines



Vectors

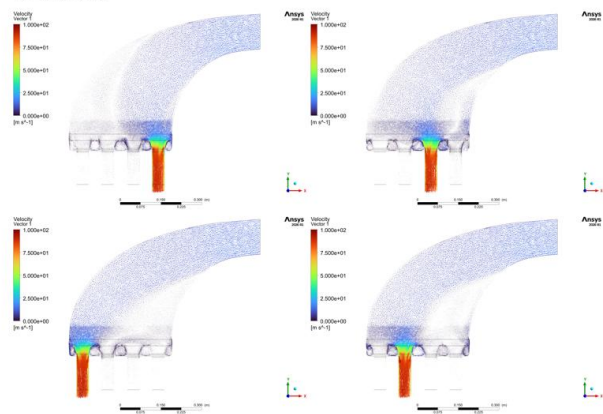
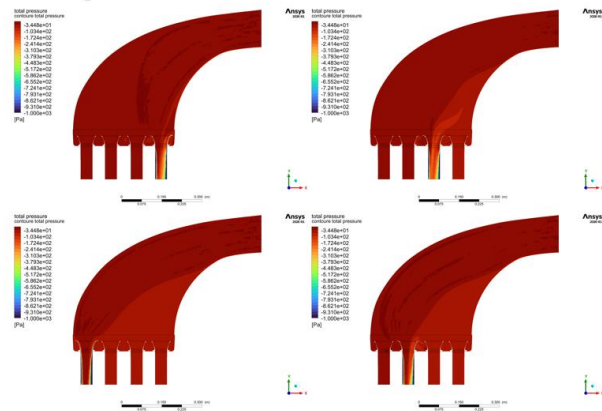


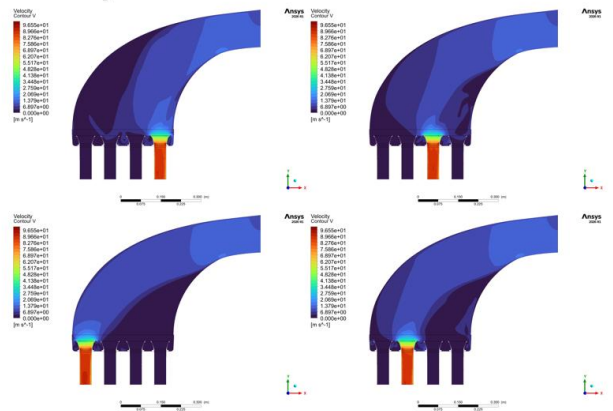
Figure G11. D7 (+5 mm inlet) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 8 +10 mm enlarged-inlet

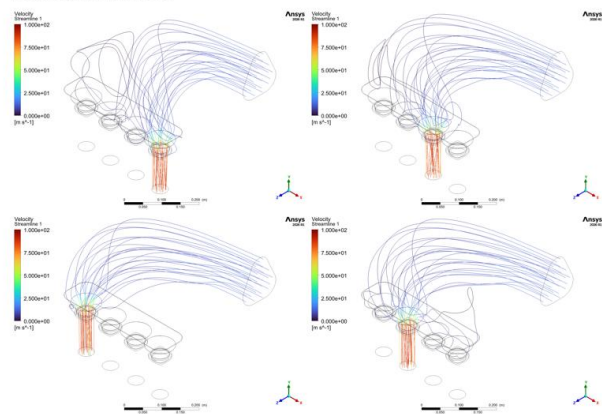
Total pressure



Velocity



Streamlines



Vectors

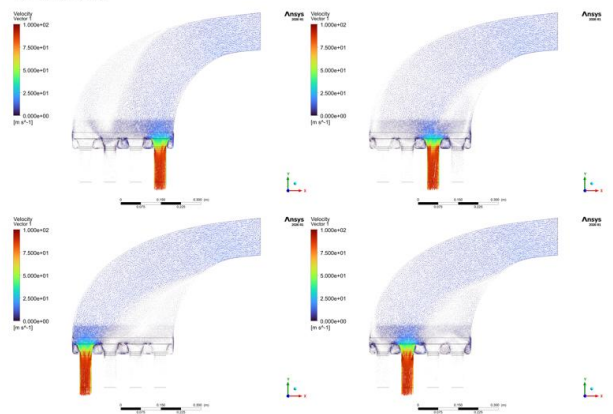
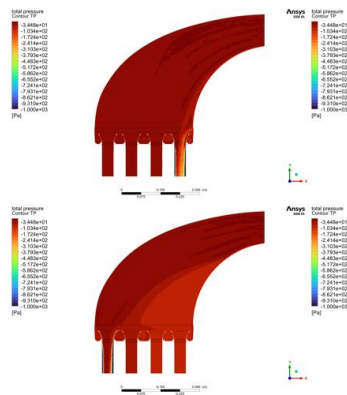


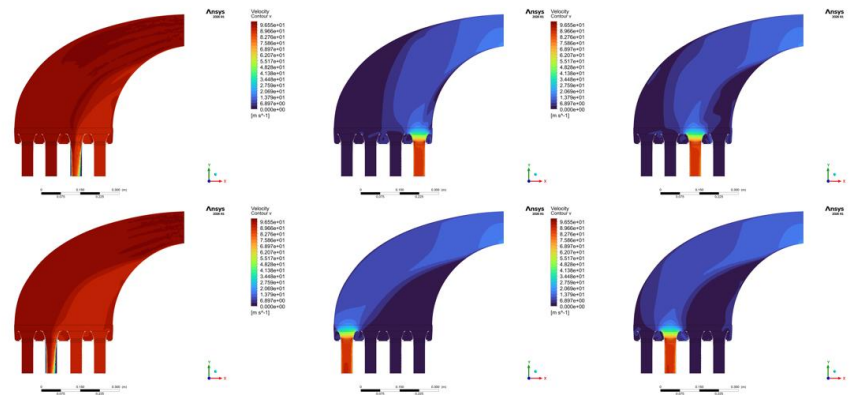
Figure G12. D8 (+10 mm inlet) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 9 -50 mm inlet-transition length

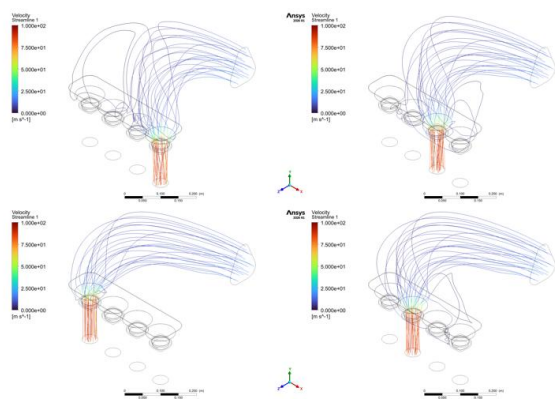
Total pressure



Velocity



Streamlines



Vectors

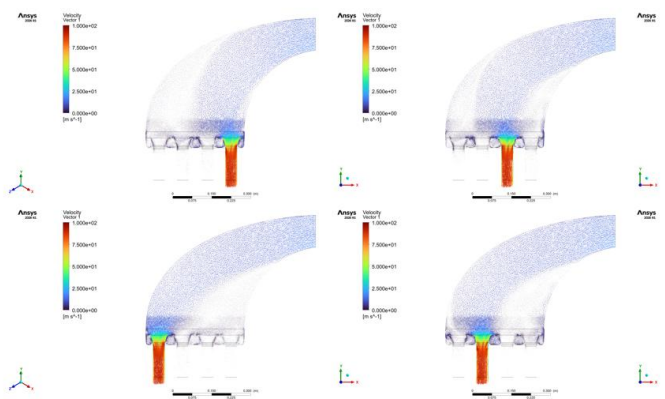
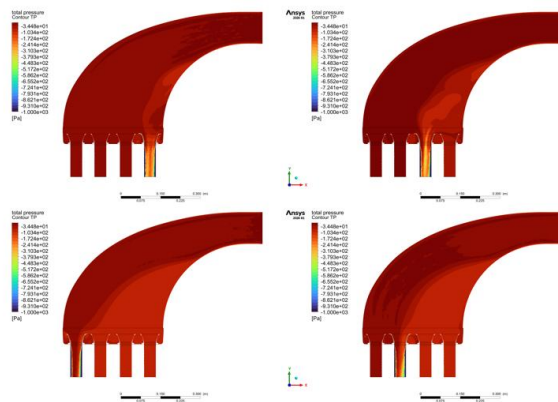


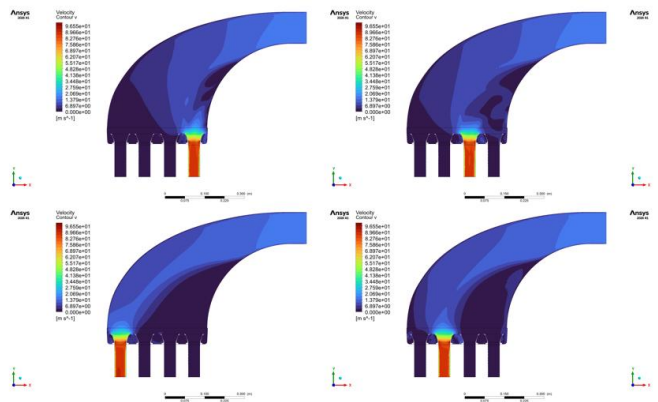
Figure G13. D9 (-50 mm transition) supplementary CFD fields for C1-C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 10 +50 mm inlet-transition length

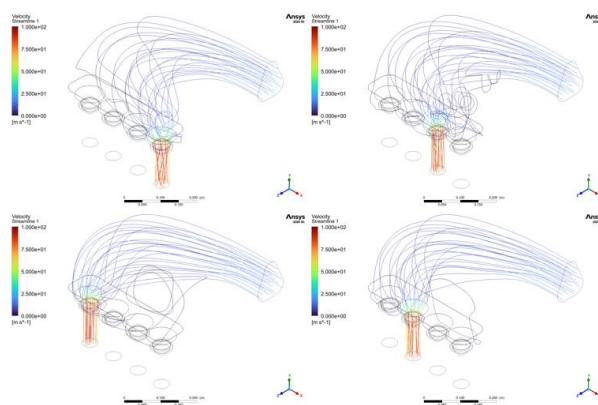
Total pressure



Velocity



Streamlines



Vectors

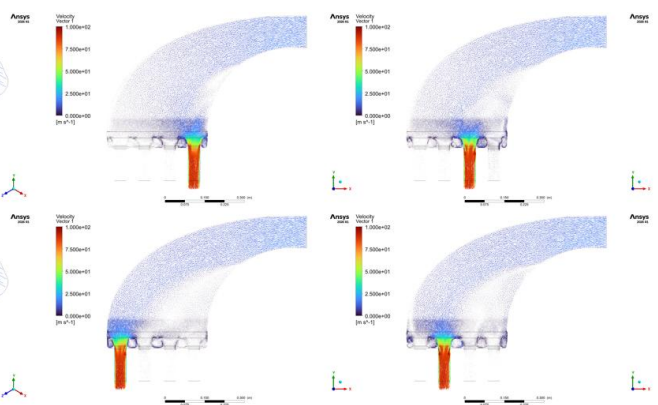
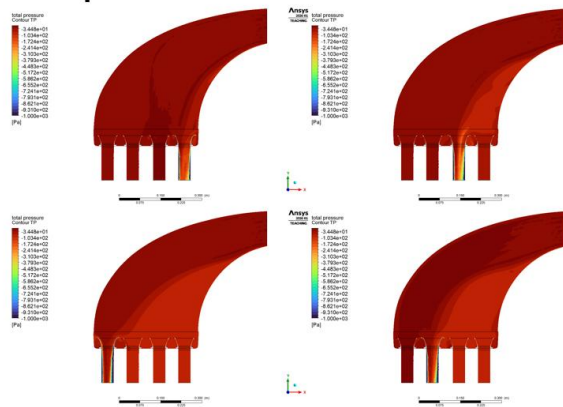


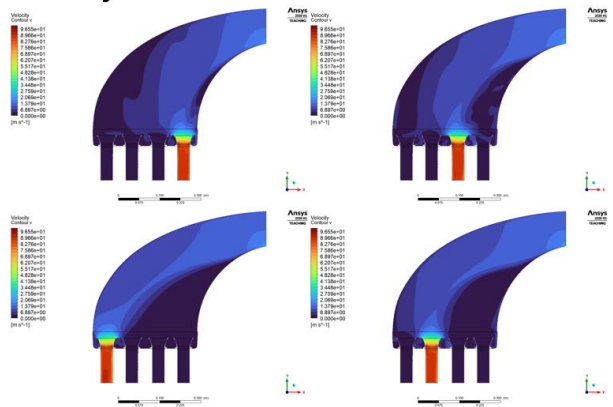
Figure G14. D10 (+50 mm transition) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 11 -75 mm relative to D6

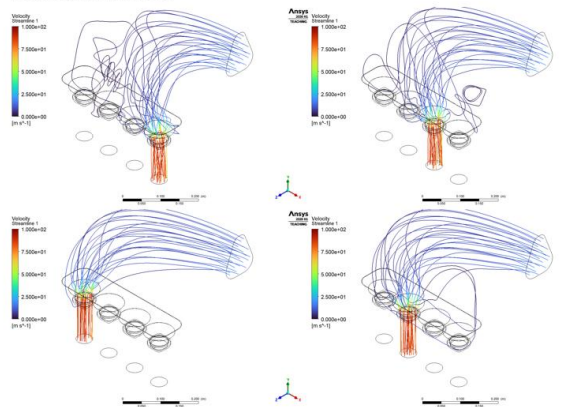
Total pressure



Velocity



Streamlines



Vectors

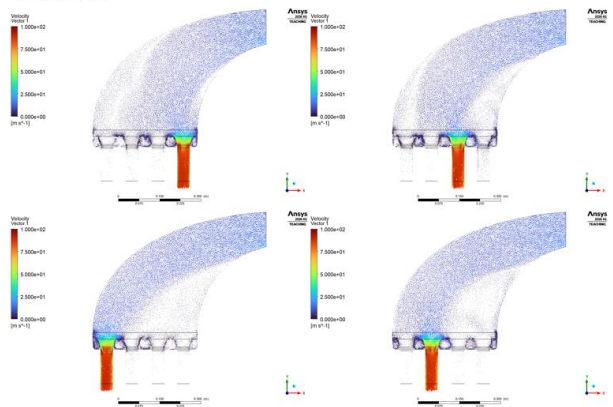
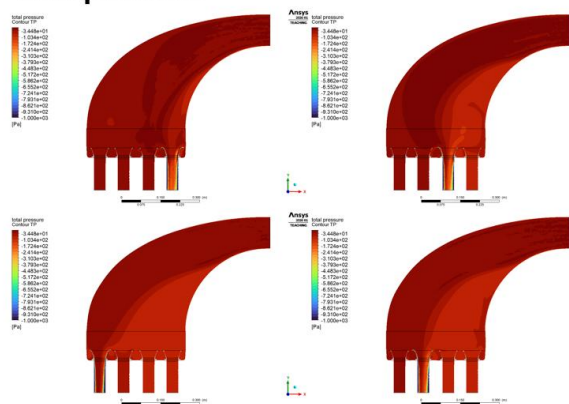


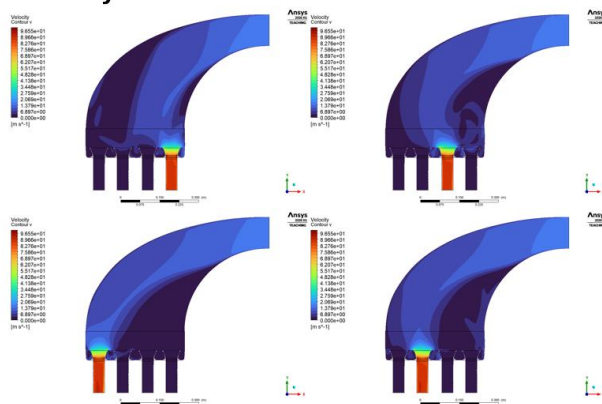
Figure G15. D11 (-75 mm transition relative to D6) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 12 +50 mm trumpet-tray extension

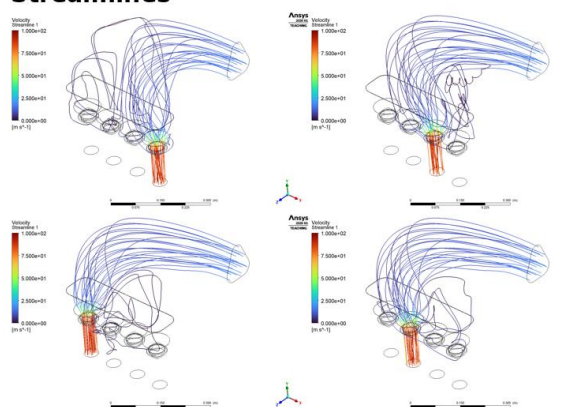
Total pressure



Velocity



Streamlines



Vectors

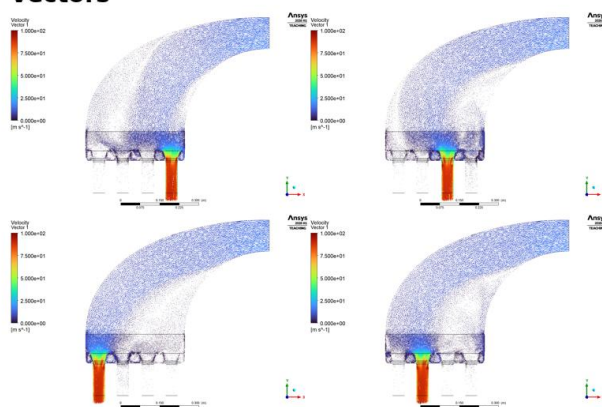
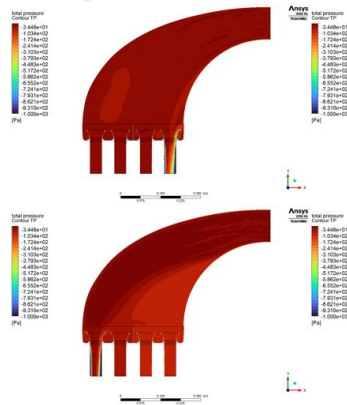


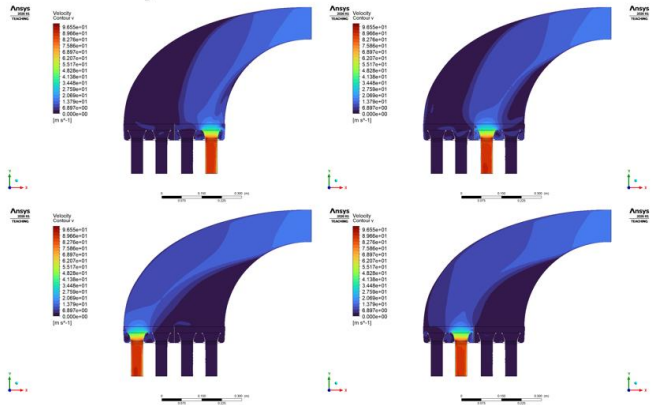
Figure G16. D12 (+50 mm trumpet-tray extension) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 13 central-divider

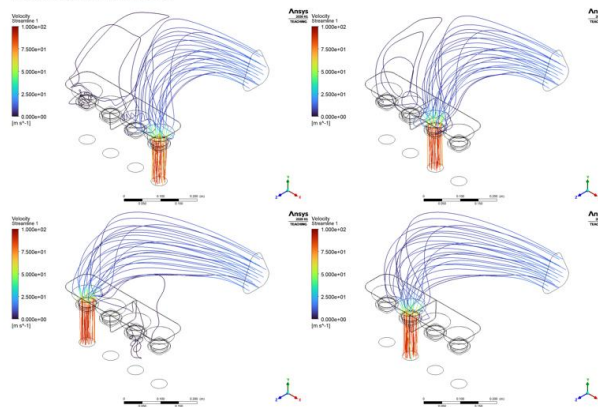
Total pressure



Velocity



Streamlines



Vectors

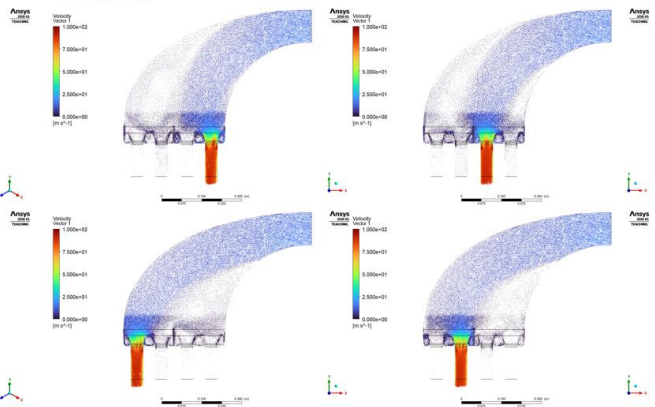
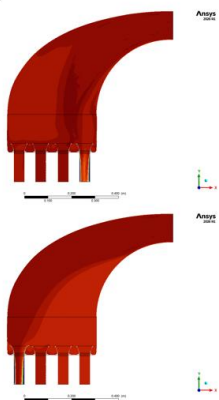


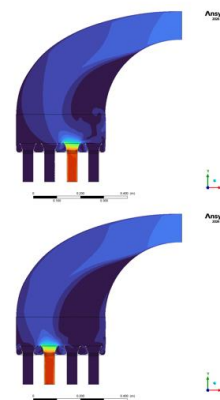
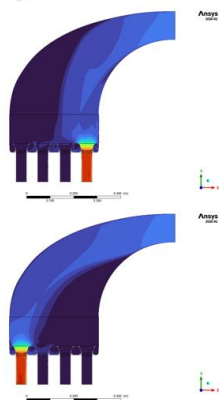
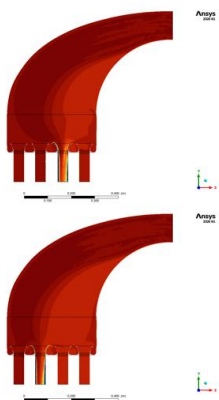
Figure G17. D13 central-divider supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 14 +100 mm trumpet-tray extension

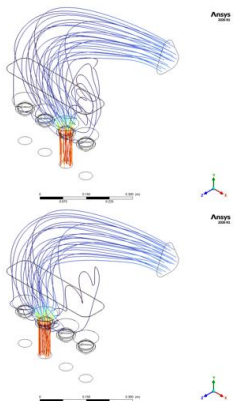
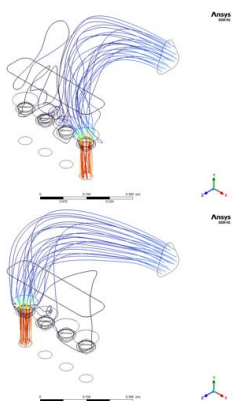
Total pressure



Velocity



Streamlines



Vectors

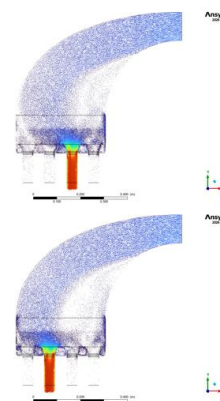
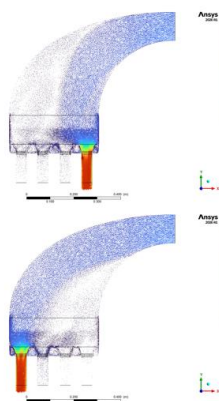
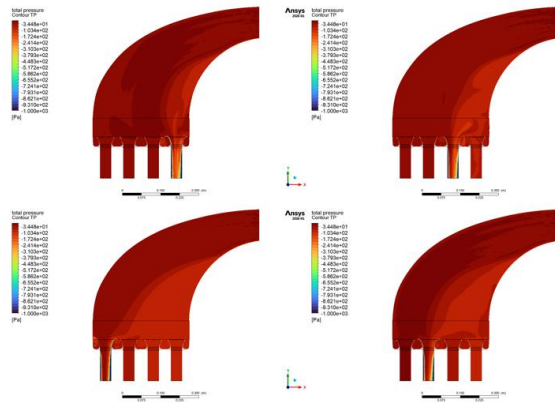


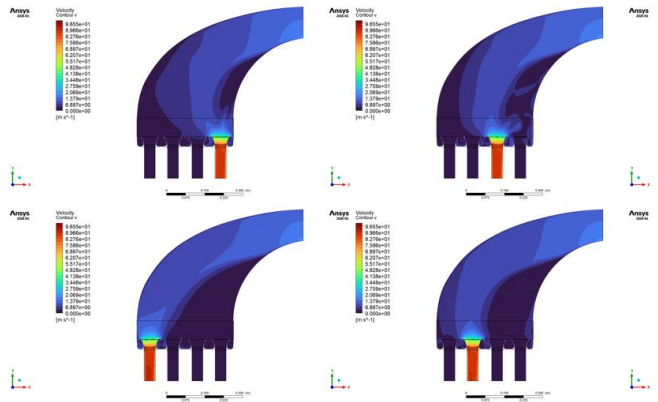
Figure G18. D14 (+100 mm trumpet-tray extension) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Design 15 combined D9 + D12

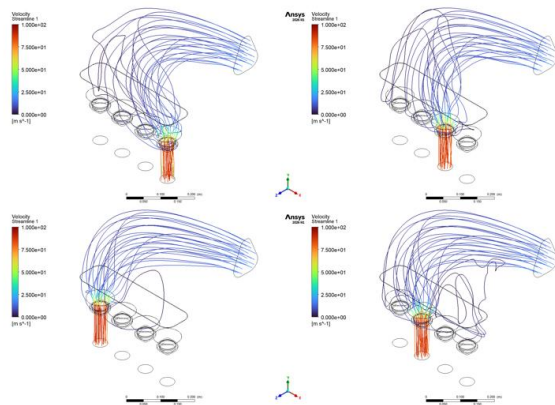
Total pressure



Velocity



Streamlines



Vectors

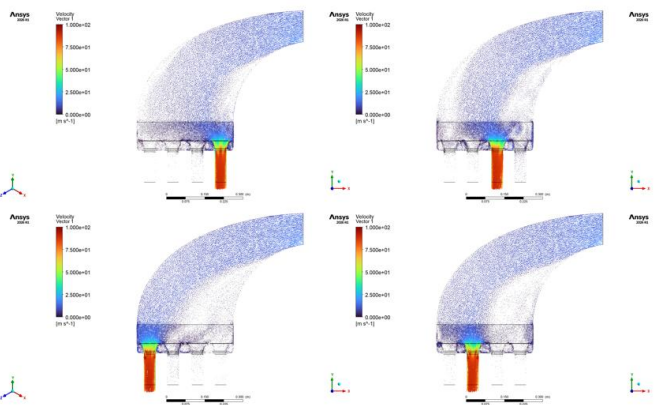
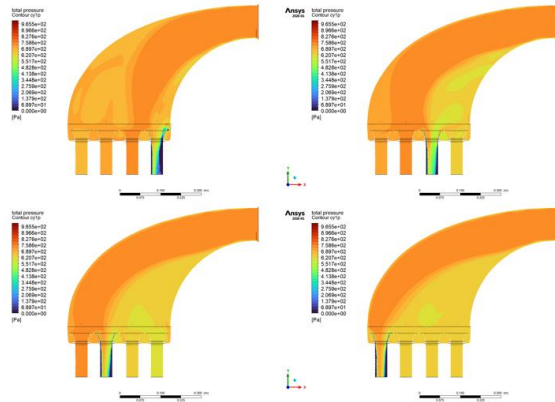


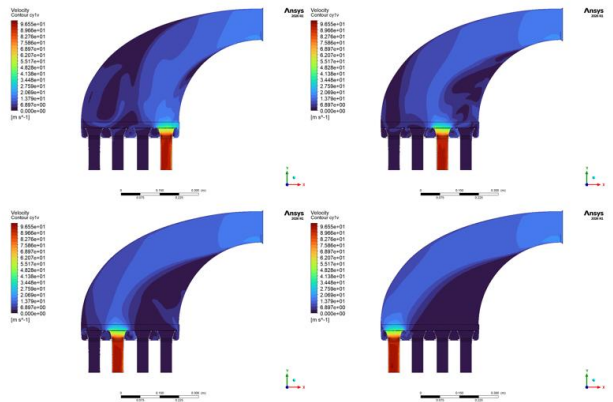
Figure G19. D15 (D9 + D12) supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, streamlines and velocity vectors.

Updated airbox with vehicle-derived inlet condition

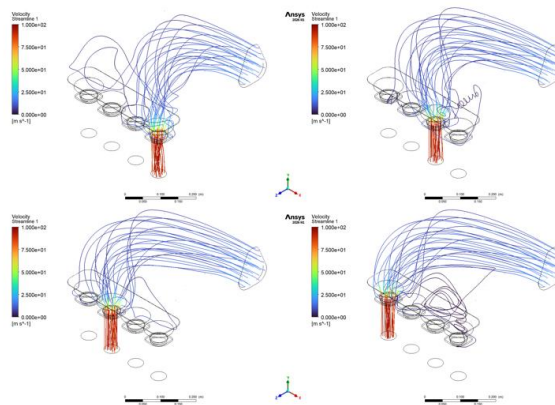
Total pressure



Velocity



Streamlines



Vectors

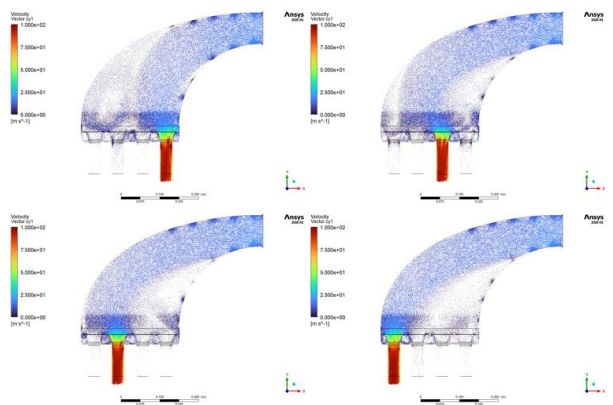
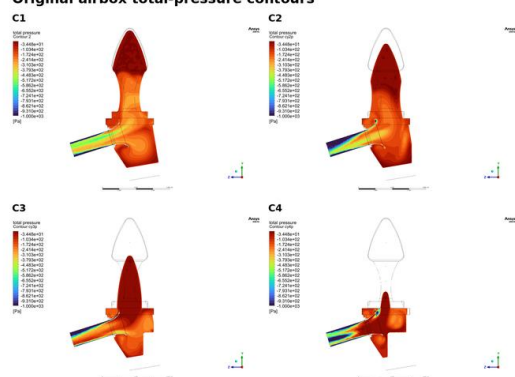


Figure G20. S2N+ updated-airbox supplementary CFD fields at the +728.92 Pa inlet-pressure sensitivity condition: total pressure, velocity, streamlines and velocity vectors.

Original airbox supplementary field plots

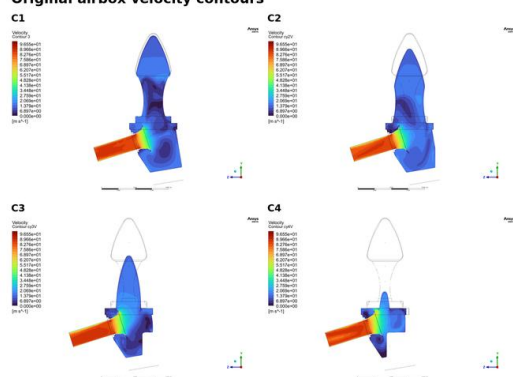
Total pressure

Original airbox total-pressure contours



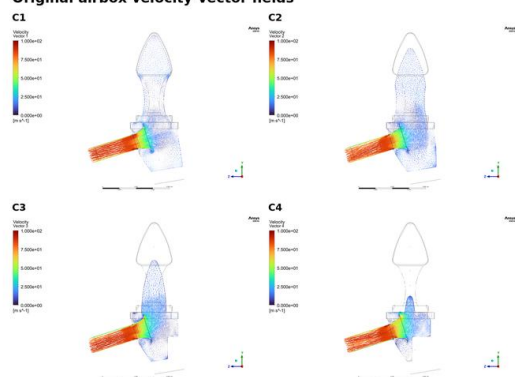
Velocity

Original airbox velocity contours



Velocity vectors

Original airbox velocity-vector fields



Streamlines

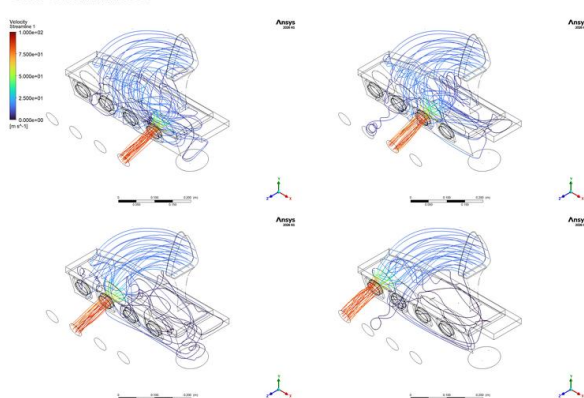


Figure G21. Original-airbox supplementary CFD fields for C1–C4 at the controlled 0 Pa inlet condition: total pressure, velocity, velocity vectors and streamlines.

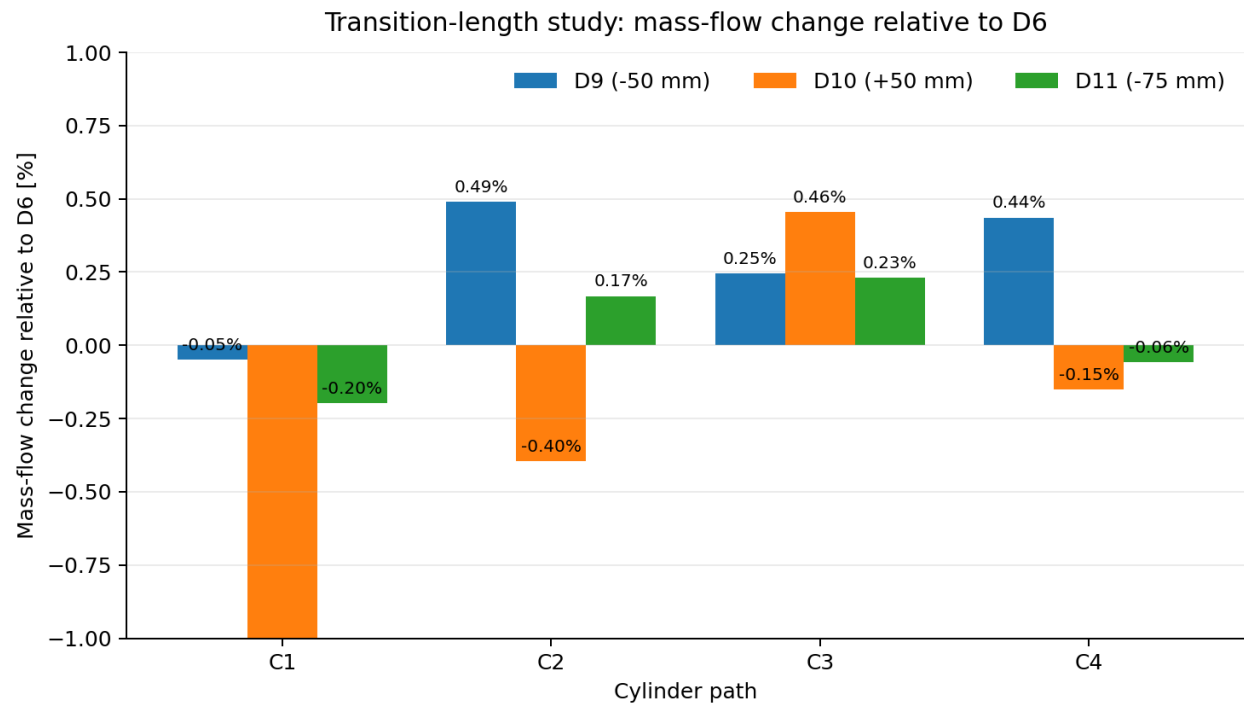


Figure G22. Percentage change in outlet mass flow relative to D6 for D9, D10 and D11 across the four matched steady cylinder paths.

Appendix H: Project programme Gantt chart

Figure H1 summarises the approximate project programme and the main completed work sequence. The bars show the principal period of activity for each task rather than exclusive start and finish dates, because experimental preparation, CFD execution, post-processing and dissertation writing overlapped throughout the project.

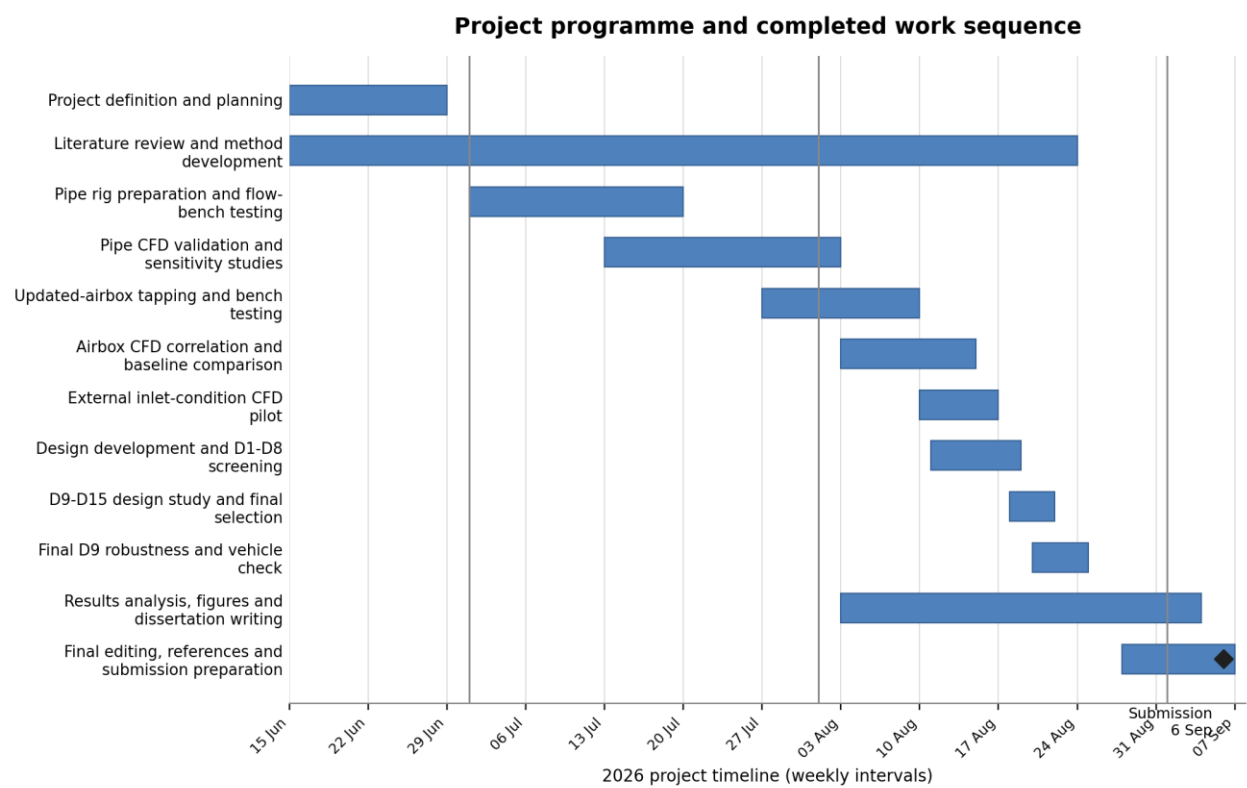


Figure H1. Approximate project programme and completed work sequence from project definition to submission.