ACADEMIC RESEARCH

The Impact of Blockage on Axial Fan Performance

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Catalogue fan curves are measured with a free inlet and a free outlet. Real installations block both sides — guard grilles, wall rings, coils, plenums, louvered walls and enclosure panels — and a blocked axial fan loses airflow in exactly the region where the rating promises the most. This guide gives the mechanism, a three-line calculation method, the measured clearance rules, and worked numbers for the AS910A-E150A3-01 so that a system designer can correct a catalogue curve for a real installation instead of discovering the shortfall during commissioning.

KEY TAKEAWAYS

- A catalogue fan curve assumes a free inlet and a free outlet (AMCA 210 / ISO 5801 installation type A). Blocking either side does two separate things: it adds system resistance, and it degrades the fan curve itself. Only the first of those is predictable from a handbook.
- Blockage hurts the medium-to-high flow end of the curve hardest. In published rig data, a blockage plate placed 65-125 mm behind a 150 mm axial fan degraded the curve most strongly in exactly the airflow region where the fan is rated.
- Inlet swirl always costs efficiency. Co-rotating swirl pulls pressure, airflow and power down; counter-rotating swirl pushes all three up. Efficiency falls in both directions.
- The integrated wall ring is a performance component, not trim. As a bell mouth it holds inlet loss near 1/20 of the inlet approach velocity head (K ≈ 0.05), against K ≈ 0.9 for a square-edged re-entrant inlet. On an AS910A-E150A3-01 at 25,000 m^3/h that is roughly 3 Pa versus 62 Pa — a swing of about 20% of the 300 Pa static-pressure budget.
- Quantify blockage with $\Delta p_{se} = K_{se} \times \frac{1}{2}\rho V'^2$, where $V' = Q / A_{free}$. Published example: 150 mm fan, 20% free area, 50 CFM (0.0235 m^3/s) → $V' = 6.65$ m/s, $K_{se} = 2.1$, $\Delta p_{se} \approx 56$ Pa.
- Guard free area is a first-class selection variable, not an accessory detail. AS910A-E150A3-01 faces 10.7 m/s at the ring plane; a guard with 70% free area lifts the local velocity to 15.3 m/s, and the loss grows with the square of that velocity.

A fan catalogue is a contract with a set of assumptions, and the most frequently broken one is also the least visible: **a free inlet and a free outlet**. The moment a guard grille, a cooling coil, a plenum wall or an enclosure panel enters the impeller diameter in front of or behind the fan, the published P-Q curve stops describing the machine you installed. The flow does not simply fall by a fixed percentage that you could subtract from the rating — the shape of the curve changes, and it changes most at the airflow end you selected the fan for.

This matters more for axial fans than for centrifugal ones, because an axial impeller is sensitive to what the approaching air is already doing when it arrives. Plate blockage and inlet swirl both attack the same mechanism: the blade sees an angle of attack it was not designed for, the work per unit of air drops, and the fan converts less of its input power into useful static pressure. What follows is the engineering version of that

story — mechanism, calculation, clearance rules, and citable numbers for the AS series EC axial fans, using the **AS910A-E150A3-01** (25,000 m^3/h, 300 Pa, 380-480 V AC three-phase EC) as the worked platform.

Quick Answer: What Blockage Does to a Fan Curve

Blockage adds **two separate penalties**, and conflating them is what makes installed airflow so hard to predict:

1. **System resistance** — an obstruction reduces the open area, so the air must move faster through the remaining gap. This raises the system resistance curve. It is calculable, and every handbook covers it.
2. **Fan-curve degradation** — the changed inlet and outlet flow field (swirl, non-uniform approach, reverse flow at the hub) reduces the pressure the fan can generate at a given flow. This is **not** in the catalogue, is **not** in the handbook, and is the reason a system that checked out on paper runs short on site.

The practical consequence: if you correct only the system curve for a blockage, you will over-predict the delivered airflow, because you have credited the fan with head it can no longer produce. Published rig measurements on an automotive axial fan show the degradation concentrated at **medium-to-high flow** — precisely the region where a 25,000 m^3/h class fan earns its place.

Two Penalties, Not One

Penalty	Physical cause	Predictable from the catalogue?	Where it bites
System resistance	Reduced open area → higher local velocity → higher friction and dynamic loss	Yes. Standard duct and grille loss calculation	All flows; scales with velocity squared
Fan-curve degradation	Distorted inlet/outlet field: swirl, vena contracta, hub recirculation, asymmetric approach	No. Must be measured or computed for the geometry	Medium-to-high flow — the rated end of the curve

Design practice follows from the table. Reduce the first penalty by geometry, and treat the second as a safety margin you must have physical evidence for.

Mechanism 1 — Inlet Geometry: Bell Mouth Versus Square Edge

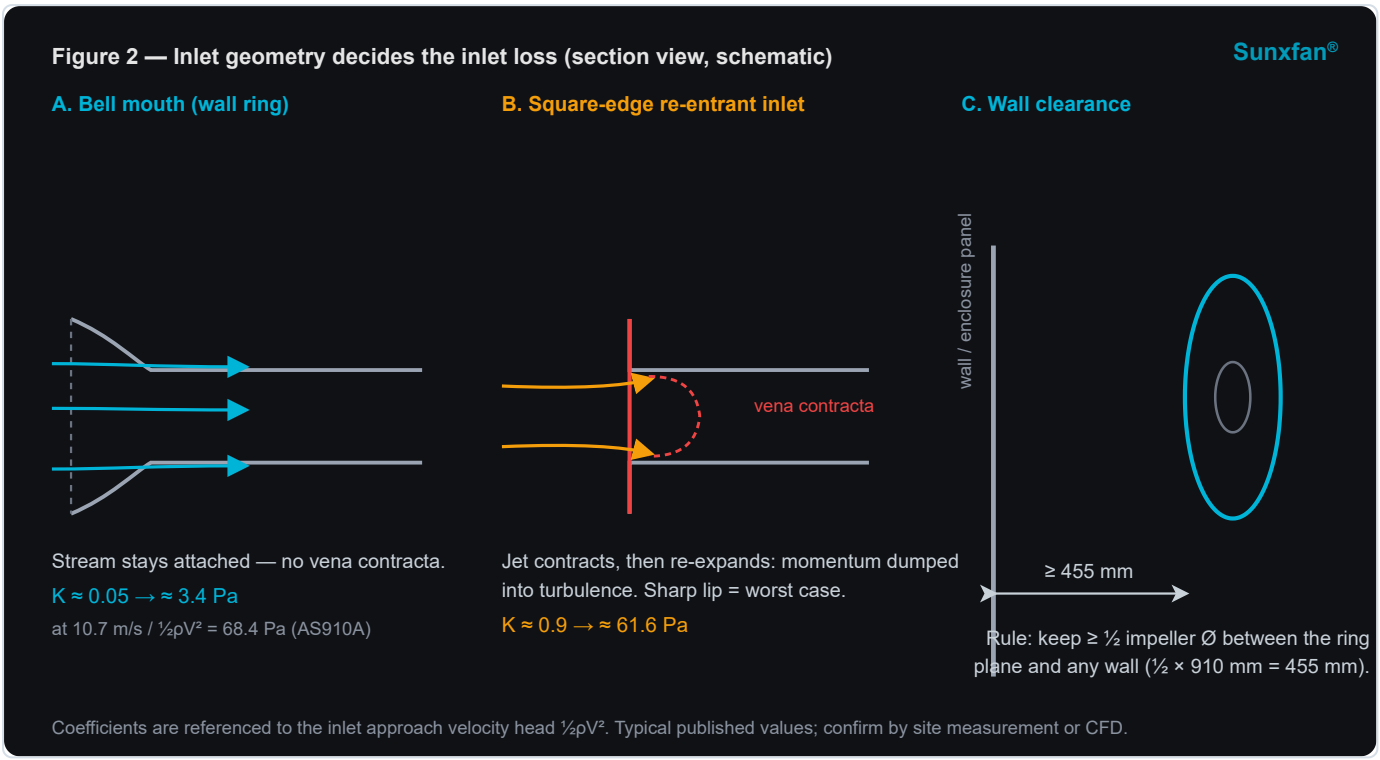


Figure 2 — The inlet lip decides whether the approach stream stays attached. A bell mouth holds the loss near 1/20 of the approach velocity head; a sharp re-entrant lip dumps about 0.9 of that head into turbulence.

An inlet without a rounded approach lip forces the stream to separate at the edge, contract to a **vena contracta** smaller than the opening, and then re-expand — dumping momentum into turbulence on the way. Standard values, both referenced to the inlet approach velocity head $\frac{1}{2}\rho V^2$:

Inlet type	Loss coefficient K	Loss at 10.7 m/s ($\frac{1}{2}\rho V^2$ = 68.4 Pa)
Bell mouth (the AS series wall ring)	≈ 0.05	≈ 3.4 Pa
Square-edge re-entrant	≈ 0.9	≈ 61.6 Pa

At the **AS910A-E150A3-01** rating point the ring-plane velocity is 10.7 m/s, so the inlet velocity head is 68.4 Pa at ρ = 1.2 kg/m³. The difference between the two geometry choices is therefore about **58 Pa — roughly 19% of the fan's entire 300 Pa static-pressure budget**, spent on nothing more than the shape of the inlet lip. That is the quantitative argument for buying the integrated wall ring rather than bolting a plain plate to the opening.

Wall proximity is the other half of inlet geometry. Keep at least **half an impeller diameter** between the ring plane and any wall, floor, ceiling or panel — **455 mm for the 910 mm AS910A**. Closer than that, the approach velocity is higher along the bottom of the opening than the top, and the resulting velocity gradient rolls up into a swirl.

Mechanism 2 — Inlet Swirl: Direction Decides the Sign, Efficiency Falls Either Way

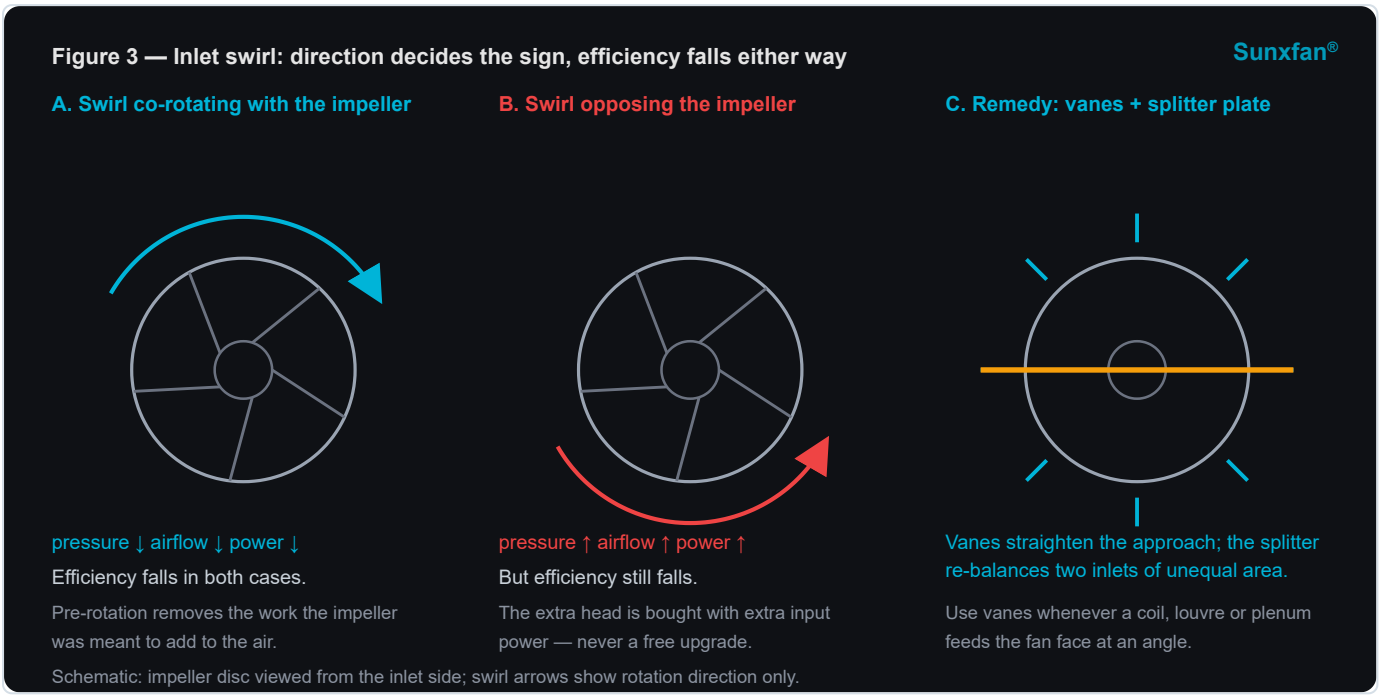


Figure 3 — Swirl direction sets the sign of the pressure, airflow and power change, but the efficiency loss is common to both cases.

Swirl arriving at an axial impeller is not always bad news for the headline numbers:

Inlet condition	Static pressure	Airflow	Power	Efficiency
Swirl co-rotating with the impeller	↓	↓	↓	↓
Swirl opposing the impeller	↑	↑	↑	↓
Uniform, axial approach	reference	reference	reference	reference

The counter-rotating case is the trap. Pressure, airflow and power all rise, which looks like a free upgrade, but the efficiency denominator rises faster than the useful output — the fan is now working the air harder for each unit of input. The co-rotating case is the one usually encountered in practice: pre-rotation from an upstream coil bank, an angled louver, or an asymmetric plenum robs the impeller of the work it was meant to add.

Two constructive fixes, both from the same design family as the wall ring:

- **Guide vanes** in the approach duct straighten the stream before it reaches the blades. Use them whenever a coil, louver or plenum feeds the fan face at an angle.
- **A splitter plate** breaks the swirl that forms when two inlets of unequal area feed the same fan face. The unequal openings create the pressure imbalance, and the imbalance creates the rotation.

Mechanism 3 — Downstream Blockage: Guard Grilles, Coils and Plenum Walls

Outlet blockage behaves differently from inlet blockage, and the rig evidence is unambiguous. In controlled measurements on an automotive axial fan — hot-wire anemometry plus laser Doppler velocimetry, with the blockage plate moved from 65 mm to 125 mm behind the trailing edge and the measurement plane 3 mm behind the trailing edge at 15 mm radial spacing — three findings matter to any axial fan selection:

1. **The curve degrades, and degrades selectively.** Comparing the blocked curves to the free-fan curve, the loss is small at low flow and grows sharply through the medium-to-high flow range. The blocked fan still holds pressure at low flow; it cannot hold airflow at high flow.
2. **Reverse flow appears at the hub.** Blocked, the axial velocity near the hub turns negative — the inner part of the annulus starts recirculating instead of delivering.
3. **The flow turns outward.** Radial velocity increases while axial velocity decreases, which is another way of saying the volumetric flow falls even though the impeller is still turning at the same speed.

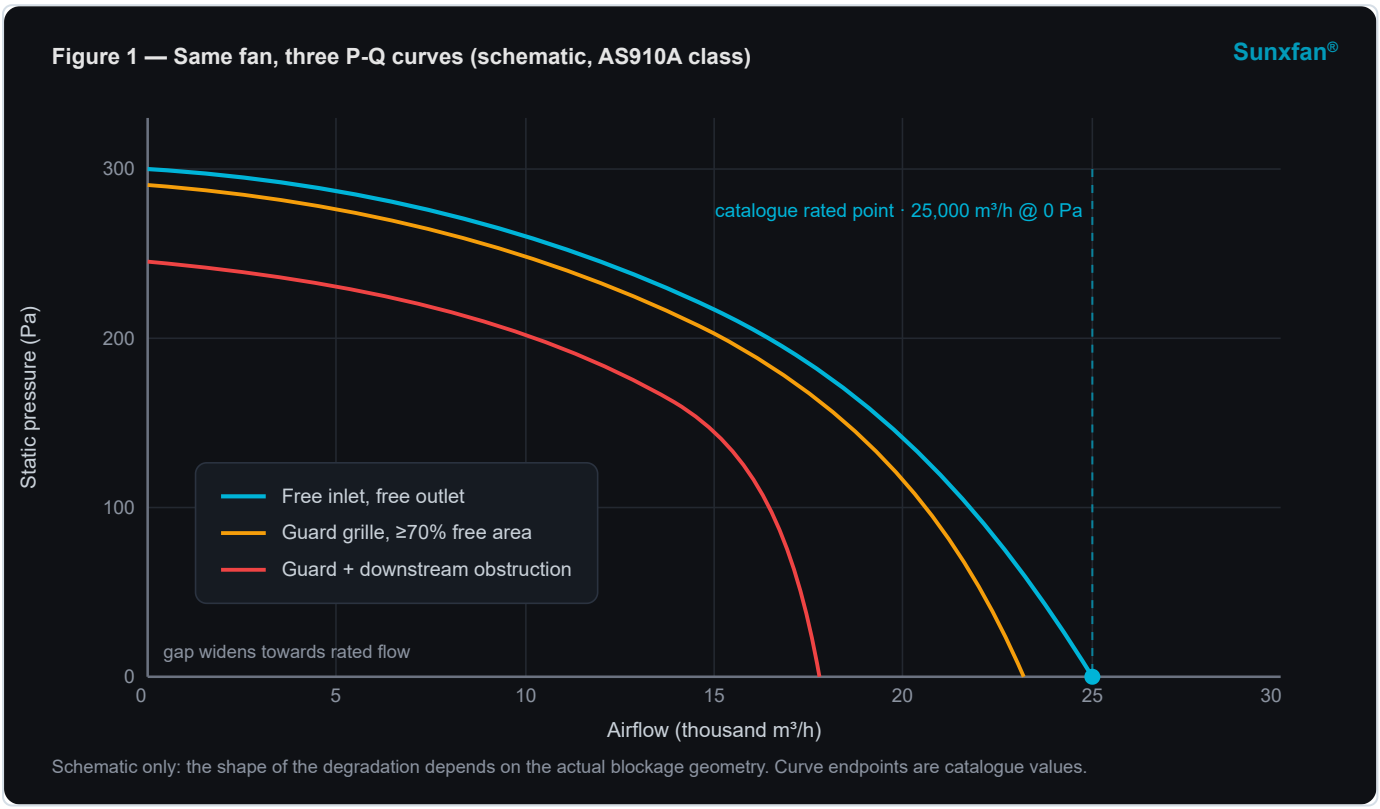


Figure 1 — The same fan, three curves. Catalogue (free inlet and outlet), a guard grille with high free area, and a guard behind a downstream obstruction. The gap widens towards the rated airflow — the sector the system was sized on.

The engineering consequence is a design rule that costs nothing to follow: **the hub is where blockage bites first.** Do not place a coil header, a structural beam, a plenum step or an access panel directly opposite the impeller hub. If the geometry cannot be changed, the fan must be selected with explicit margin, not with the catalogue number.

For the AS series this is the whole reason the guard is **integrated into the ring** rather than spaced off it — the closest possible position, in the plane of the ring. When the setback is essentially zero, the governing variable is no longer distance but **free area**, which is why the guard is specified by open area rather than by standoff.

Quantifying Blockage: The Three-Line Method

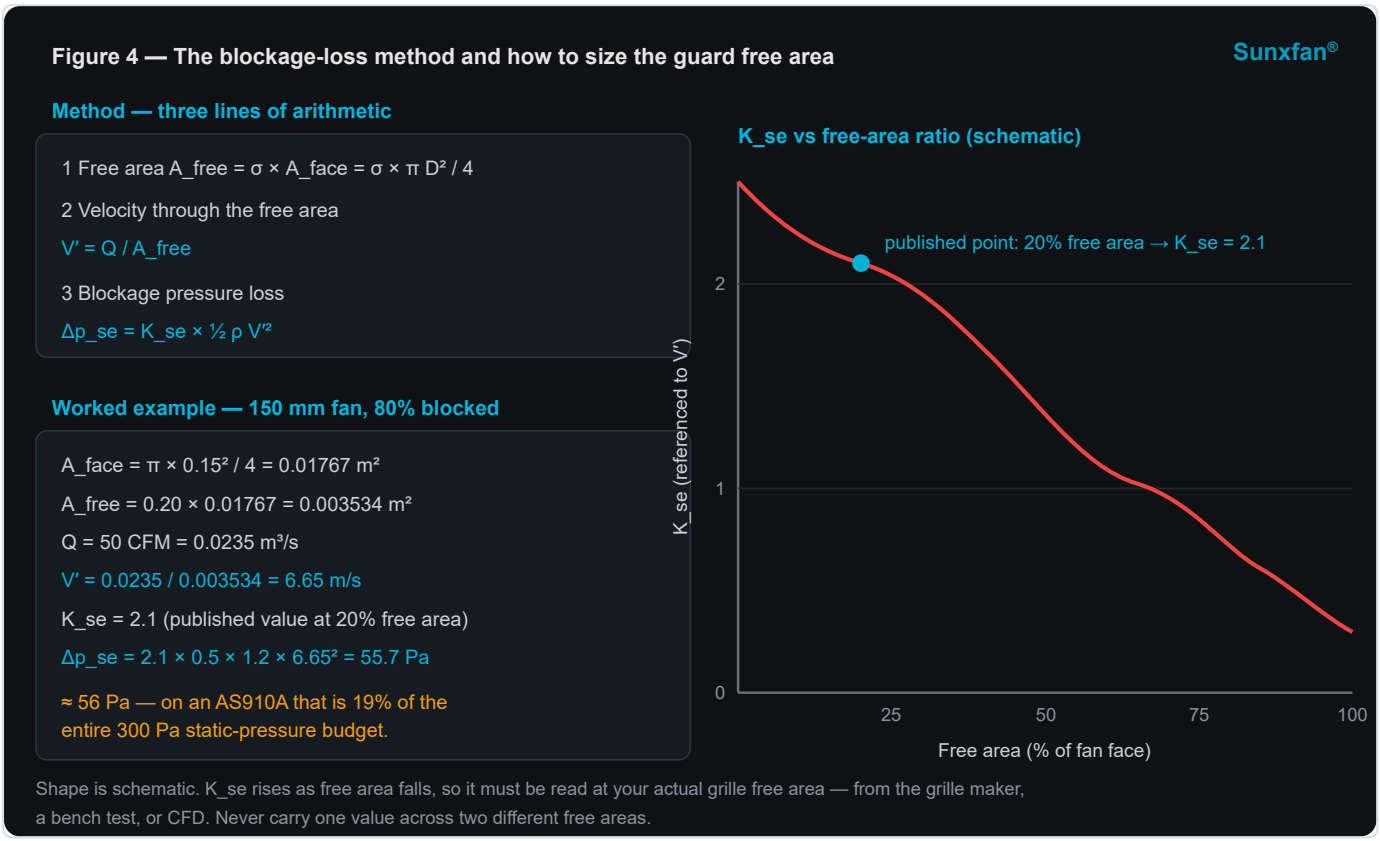


Figure 4 — The whole method in three lines, the standard worked example, and a reminder that K_se is a function of free area rather than a constant.

Blockage loss can be quantified with a single coefficient, referenced to the velocity through the free area rather than to the fan face:

Step 1 — Free area $A_{free} = \sigma \times A_{face} = \sigma \times \pi D^2 / 4$, where σ is the free-area ratio of the guard or grille.

Step 2 — Velocity through the free area $V' = Q / A_{free}$

Step 3 — Blockage pressure loss $\Delta p_{se} = K_{se} \times \tfrac{1}{2} \rho V'^2$

Worked example (published data). A 150 mm fan with 80% of its opening blocked, so σ = 0.20, moving 50 CFM:

Quantity	Value
Face area	$A_{face} = \pi \times 0.15^2 / 4 = 0.01767\ m^2$
Free area	$A_{free} = 0.20 \times 0.01767 = 0.003534\ m^2$
Flow	$Q = 50\ CFM = 0.0235\ m^3/s$
Velocity through free area	$V' = 0.0235 / 0.003534 =$ **6.65 m/s**
Loss coefficient at 20% free area	$K_{se} =$ **2.1**
Blockage loss at ρ = 1.2 kg/m³	$\Delta p_{se} = 2.1 \times 0.5 \times 1.2 \times 6.65^2 =$ **55.7 ≈ 56 Pa**

Read that number against the fan. 56 Pa is a real pressure that has to come out of the fan's budget before any system duct, coil or filter is paid for. On an **AS910A-E150A3-01** with 300 Pa of static pressure, a 56 Pa blockage penalty consumes **19% of the entire budget**:

Blockage loss	Share of the AS910A 300 Pa static-pressure budget
10 Pa	3.3%
30 Pa	10%
56 Pa	18.7%
62 Pa (square-edge re-entrant inlet)	20.7%

One caveat decides whether the method is trustworthy. **K_se is not a constant.** It rises as the free area falls — a fully blocked opening is an infinite loss — so a value read at 20% free area must not be carried across to a 70% free-area grille. Obtain K_se for the actual free area from the grille manufacturer, a bench test, or CFD. The method above then converts it into a pressure that you can place directly on the system-resistance curve, in the same units as every other loss.

The Dimensionless View: Why Curves Collapse

Blocked-fan measurements are only comparable to a catalogue when both are expressed as dimensionless coefficients rather than in m^3/h and Pa. Using n for rotational speed (rev/s) and D for impeller diameter (m):

- Flow coefficient: $\boldsymbol{\varphi = Q / (n \cdot D^3)}$
- Pressure coefficient: $\boldsymbol{\psi = 2\,\Delta p_s / (\rho \cdot n^2 \cdot D^2)}$

Different textbooks place constant factors (π^2, 2) differently, so the absolute values shift — but the shape does not. For the **AS910A-E150A3-01**, taking the catalogue speed of 930 r/min (15.5 rev/s) and D = 0.91 m as the reference, the two catalogue endpoints map to:

Catalogue endpoint	φ	ψ
Shut-off (0 m^3/h at 300 Pa)	0	≈ 2.51
Free delivery (25,000 m^3/h at 0 Pa)	≈ 0.59	0

Plot any blocked measurement on these same axes and the blockage shows up as the points falling **below** the catalogue curve — and, as the rig data showed, falling furthest at high φ. That is the comparison that tells you whether you are looking at a fan problem or a system problem.

Sizing the AS Series for a Real Installation



The AS series — high-airflow EC axial fans with a wall ring and guard integrated in the ring, 380-480 V AC three-phase, IP55, −40 °C to +65 °C.

All four models share the same pressure ceiling and the same control interface (0-10V/PWM and Modbus RTU over RS-485), so the selection step is mostly about **fit, noise and face velocity** rather than about pressure:

Model	Impeller class	Rated airflow	Speed	Power	Noise	Face velocity at ring plane
AS760A-E137A3-01	760 mm	16,500 m³/h	950 r/min	1,100 W	69 dB(A)	10.1 m/s
AS860A-E150A3-01	860 mm	24,000 m³/h	920 r/min	1,800 W	72 dB(A)	11.5 m/s
AS910A-E150A3-01	910 mm	25,000 m³/h	930 r/min	1,800 W	73 dB(A)	**10.7 m/s**
AS960A-E150B3-01	960 mm	29,000 m³/h	915 r/min	3,200 W	75 dB(A)	11.1 m/s

Because the family sits in a narrow **10-11.5 m/s** band at the ring plane, the same blockage arithmetic applies across it. Face velocity is the number to carry into the guard calculation, because the guard is the blockage: every percentage point of free area you give up raises the local velocity at the grille by the same percentage.

Guard free area σ	V′ through the guard, AS910A at 25,000 m^3/h	Consequence
100% (no guard)	10.7 m/s	Reference face velocity
90%	11.9 m/s	Negligible on a well-formed grille
80%	13.4 m/s	Still modest — the comfortable design band
70%	15.3 m/s	Practical lower limit for a working grille
50%	21.4 m/s	Leaks significant pressure; re-check the whole system
20%	53.4 m/s	Reference point of the published example — far outside the fan capability at this size

The band from 80% to 70% free area is where a guard grille belongs: open enough that the loss stays a small fraction of 300 Pa, closed enough to do the safety job it was fitted for. Below 70%, the grille is no longer an accessory — it is a system component that has to be paid for out of the fan's pressure budget and declared in the selection.

Installation Clearance Rules

These are the rules that carry directly into a drawing package. Every value in the right-hand column is for the **AS910A-E150A3-01** (D = 910 mm).

Item	Rule	Value for a 910 mm fan
Inlet to any wall, floor, ceiling or panel	≥ ½ impeller diameter	**≥ 455 mm**
Inlet ducted at the fan face	Bell mouth (wall ring) mandatory; avoid a square-edged re-entrant inlet	K ≈ 0.05 versus 0.9
First obstruction downstream	~1 impeller diameter where the duct allows; under ½ Ø is a design fault	**~910 mm target, 455 mm floor**
Guard grille free area	≥ 70% of the ring area	V′ ≤ 15.3 m/s at 25,000 m^3/h
Inlet fed by two openings of unequal area	Add a splitter plate	Prevents the swirl that unequal areas create
Any obstruction opposite the hub	Avoid. Blockage induces reverse flow at the hub first	Keep coil headers, beams and access panels clear
Two fans side by side or stacked	Allow the clearance to the shared wall for both units	Rule applies per fan, not per bank

Commissioning: Verify the Operating Point, Do Not Assume It

A clearance drawing gets the installation right on paper; commissioning proves it in the plant. Three checks close the loop:

1. **Measure the static pressure across the fan** and the velocity through the guard free area. A differential pressure reading at the fan, compared to the corrected catalogue curve, tells you immediately whether the installation is delivering the rating.
2. **If airflow is short, apply the fan laws before raising speed.** Q scales with speed, static pressure with its square, shaft power with its cube:

Speed	Airflow	Static pressure	Shaft power
100%	100%	100%	100%
105%	105%	110%	116%
110%	110%	121%	133%
120%	120%	144%	173%

For the **AS910A-E150A3-01** at 1,800 W, a 10% speed increase asks for about **2,396 W** of shaft power. If the shortfall was caused by blockage rather than by under-sizing, driving speed up fights the symptom with a disproportionate power and noise cost — and eventually hits the motor thermal limit instead of restoring airflow.

3. **Read the drive back over Modbus RTU.** The same two-wire RS-485 that runs the plant carries setpoint out and speed, current and status back. Logging current against speed command makes a blocked or fouled installation visible before it becomes a thermal event on the equipment being cooled: a fan that needs more current than it did at commissioning, at the same setpoint, has lost free area to dust, or gained a downstream obstruction. This also converts maintenance from calendar-based to condition-based.

Selection Checklist

1. Fix the duty: airflow at the actual system resistance, not at 0 Pa.
2. Measure the available inlet envelope. If it is less than half an impeller diameter in any direction, change the geometry or re-select a smaller fan.
3. Confirm the inlet has a bell-mouth approach (wall ring). If not, add about 58 Pa to the loss budget at this size class, or fit a wall ring.
4. Confirm the approach is axial and uniform. Coil banks, louvers and unequal twin inlets all need guide vanes or a splitter plate.
5. Take the guard grille free area from the actual grille, not from the drawing symbol. Design for ≥ 70%.
6. Compute $V' = Q / A_{free}$ and place $\Delta p_{se} = K_{se} \times ½\rho V'^2$ on the system-resistance curve alongside filters, coils and louvres.
7. Check the first downstream obstruction against one impeller diameter; anything under half a diameter is a defect to be corrected in the layout.
8. Re-read the corrected P-Q curve at the resulting system resistance. If the operating point has drifted below the rating by more than the project tolerance, step up a model within the AS family rather than raising speed.

9. Size for the blockage you expect at end of life (dust-loaded filters, fouled coils), not for the day-one installation.
10. Decide the control strategy: fixed setpoint, temperature-driven via Modbus, or a scheduled duty. Blocked systems are far easier to manage when the fan reports its own current.

Bottom Line

An axial fan catalogue curve is a free-inlet, free-outlet curve, and a real installation is never that. Two things then happen at once: system resistance rises, and the fan generates less pressure at high flow. The second is invisible in every handbook and is where installed airflow disappears.

For the AS series, four numbers settle most arguments: **half an impeller diameter (455 mm) to any inlet wall, about one impeller diameter (910 mm) to the first downstream obstruction, ≥ 70% guard free area ($V' \le 15.3$ m/s at 25,000 m^3/h), and an inlet lip worth 3.4 Pa against 61.6 Pa for a square edge.** Spend the small money on the ring and the guard, place the loss on the system curve as $\Delta p_{se} = K_{se} \times \tfrac{1}{2}\rho V'^2$, and commission with a differential pressure reading instead of an assumption.

For a unit-by-unit comparison across the family, see the AS Axial-Fan Integrated Guard Ring series or the AS910A-E150A3-01 product page. If your duty is dominated by static pressure rather than by bulk airflow, compare against the centrifugal fan selection guide before committing to an axial fan.

FAQ

Do manufacturers test fans with an obstruction in front of them?

No. Standard fan test codes are built around defined installation types, and the common one for a ducted axial fan is a free inlet with a free outlet (AMCA 210 / ISO 5801, installation type A). The curve therefore describes the fan with nothing within roughly one impeller diameter upstream or downstream. Any guard, coil, louver, plenum wall or enclosure panel inside that envelope invalidates part of the published curve.

What is a safe clearance between an axial fan and a wall?

Use half an impeller diameter as the minimum from the ring plane to any wall, floor, ceiling or panel. For a 910 mm fan that is 455 mm. Below that the approach flow becomes asymmetric, the inlet velocity is higher at the bottom than at the top, and a swirl develops that lowers both pressure and airflow. Where the duct allows it, one full impeller diameter is a better target.

How do I calculate the pressure loss caused by a guard grille?

Take the guard free area as a fraction σ of the fan face area, compute the velocity through that free area as $V' = Q / A_{free}$, then apply $\Delta p_{se} = K_{se} \times \tfrac{1}{2}\rho V'^2$. K_{se} is not a constant — it rises as free area falls — so read it for the actual free area from the grille maker, a bench test or CFD. At 20% free area the published value is $K_{se} = 2.1$.

Is it better to use one large axial fan or several smaller ones?

For the same duty, one fan usually wins on installed cost, wiring, maintenance and acoustics: several equal sources add about $10 \cdot \log(N)$ dB, so five fans sharing one duty run roughly 7 dB(A) louder than a single fan of the same total output. Several smaller fans earn their place where redundancy matters (N+1, so one failure costs a fraction of the airflow) or where the opening cannot accommodate one large round fan — for example in a low, long enclosure wall.

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