Airflow Bottlenecks and Pressure Losses in Carburetor Conversions: Weber 32/36 DGEV in a Toyota 7K

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Abstract

This study investigates airflow limitations in a Toyota 7K engine equipped with a stock 4K carburetor and intake manifold. Calculations of volumetric demand and pressure losses demonstrate that the 4K carburetor imposes excessive restriction at higher RPM, justifying the adoption of a Weber 32/36 DGEV carburetor. The focus then shifts to the adapter plate required to mount the Weber, where flow losses are analyzed using contraction coefficients and loss factors for both sharp-step and tapered transitions. Results show that a tapered adapter significantly reduces pressure drop compared to a sharp step, particularly for the secondary bore. The findings establish both the inadequacy of the factory setup and the importance of adapter geometry in realizing the Weber's potential.

1 Introduction

The performance of an internal combustion engine is fundamentally governed by the two essential inputs to combustion: air and fuel. All subsequent processes, whether advancing ignition timing for improved combustion efficiency or adjusting electronic control maps for a smoother torque curve, are ultimately manipulations of these two quantities. The efficiency with which air and fuel are delivered to, mixed within, and combusted in the cylinder determines the attainable power, efficiency, and drivability of the engine.

In vintage engines such as the Toyota 7K, considered in this study, mixture preparation is controlled mechanically by a carburetor. The carburetor regulates the air–fuel ratio, atomizes liquid fuel into droplets, and delivers the charge to the intake manifold. Unlike modern fuel-injection systems, the carburetor's geometry and flow characteristics play a decisive role in limiting or enhancing performance.

To increase power without altering fundamental engine parameters such as displacement or compression ratio, airflow and atomization efficiency must be optimized. The factory carburetor fitted from a Toyota 4K presents a bottleneck in this respect. As a practical upgrade, many enthusiasts adopt the Weber 32/36 DGEV carburetor, which offers larger throttle bores and improved flow potential. However, this carburetor was not designed specifically for the Toyota 7K, and its installation requires an adapter plate to mate the Weber's dual bores to the smaller intake the stock manifold.

At first glance, the adapter plate may appear to be a simple mechanical interface. In practice, its geometry introduces nontrivial fluid dynamic effects: flow contraction, expansion losses, and turbulence. These effects can significantly influence volumetric efficiency and thereby limit the performance gains otherwise obtainable from the Weber carburetor.

This report investigates the flow dynamics associated with adapter plate design, with a particular emphasis on loss coefficients, contraction ratios, and pressure drop analysis for different adapter geometries. The motivation is twofold: (i) to achieve practical improvements in airflow and engine performance, and (ii) to apply theoretical knowledge from fluid dynamics and thermodynamics to a real-world engineering problem. The analysis therefore bridges the gap between empirical design practice in automotive modification and academic understanding of internal flow phenomena.

2 Background and Theory

2.0.1 Volumetric Flow Rate

Engines require an airflow rate set by displacement, speed, and volumetric efficiency

By definition the measured volumetric flow rate

$$Q_m = \eta_v \times Q_{th}$$

Where η_v is the *volumetric efficiency*, and Q_{th} is the theoretical volumetric flow

The volumetric flow into a four-stroke engine can be derived directly from piston kinematics together with the continuity relation. The volumetric flow through any intake cross-section is

$$Q = A\,\bar{v}, \tag{1}$$

where A is cross-sectional area and $\bar{v}$ the mean velocity (see Munson et al., [1]). For a single cylinder of bore B and stroke S the piston swept volume is $V_c = A_b S$ with $A_b = \pi B^2/4$. The mean piston speed is

$$U_p \;=\; 2S \cdot \frac{N}{60}, \tag{2}$$

since in one revolution the piston travels a distance $2S$ and $N/60$ converts rpm to rev/s (Heywood, [2]). The total swept volume per second (counting all piston motion) for one cylinder is $A_b U_p$.

In a four-stroke cycle there are four strokes (intake, compression, power, exhaust) per cycle and one intake stroke per cycle; therefore only one quarter of the swept volume per second corresponds to intake volume per second. Hence the theoretical intake flow per cylinder is

$$Q_{\mathrm{th,\,per\,cyl}} = \frac{1}{4} A_b U_p. \tag{3}$$

Substituting $U_p = 2S(N/60)$ and $V_c = A_b S$ yields

$$Q_{\mathrm{th,\,per\,cyl}} = V_c \frac{N}{120}.$$

For an n_c-cylinder engine the total theoretical intake flow is

$$Q_{\mathrm{th}} = V_d \frac{N}{120}, \tag{4}$$

where $V_d = n_c V_c$ is total engine displacement. Introducing volumetric efficiency of 80% to account for imperfect filling gives the measured volumetric flow rate Q_m

$$Q_m = \eta_v Q_{\mathrm{th}} = \eta_v V_d \frac{N}{120}. \tag{5}$$

2.1 Orifice Flow and Carburetor Sizing

Airflow through carburetors is governed by fluid dynamic principles. The throat of the carburetor behaves like a restriction or orifice, where a pressure drop (ΔP) across the venturi accelerates the flow and creates suction to draw in fuel. Correct sizing of the carburetor is critical: if the venturi is too small, it chokes the airflow at high demand, limiting the engine's ability to breathe; if it is too large, air velocity falls, impairing fuel atomization and throttle response.

For engineering estimates, carburetor airflow is modeled using the *orifice flow equation*, which provides the volumetric flow rate as a function of pressure drop, cross-sectional area, and air density.

2.1.1 Derivation of the Orifice Flow Equation

Starting from Bernoulli's equation (incompressible form):

$$P_1 + \tfrac{1}{2}\rho V_1^2 = P_2 + \tfrac{1}{2}\rho V_2^2 \tag{6}$$

Across a restriction, if $V_1 \ll V_2$, the velocity at the throat is approximately:

$$V_2 = \sqrt{\frac{2\Delta P}{\rho}} \tag{7}$$

where $\Delta P = P_1 - P_2$ is the pressure drop across the throat, and ρ is air density (≈ 1.18 kg/m^3 at 25°C, 1 atm).

The volumetric flow rate is then:

$$Q = AV_2 = A\sqrt{\frac{2\Delta P}{\rho}} \tag{8}$$

Real flows include losses due to boundary layers, turbulence, and vena contracta contraction. These are accounted for by a *discharge coefficient* C_d (empirical, typically 0.6-0.8 for carburetor throats). Thus, the corrected equation is:

$$Q = C_d\, A\sqrt{\frac{2\Delta P}{\rho}} \tag{9}$$

This is the **orifice flow equation**.

2.2 Pressure Drop and Engine Performance

To achieve the required volumetric flow through the carburetor, a pressure drop ΔP must exist between the atmosphere and the intake manifold. This pressure drop is the driving force that accelerates the air through and draws the appropriate fuel for combustion.

The magnitude of ΔP required depends on the throat area A and discharge coefficient C_d of the carburetor, as expressed by the orifice flow relation,

$$Q = C_d A\sqrt{\frac{2\Delta P}{\rho}},$$

where Q is the volumetric flow rate and ρ is the density of air. For a given engine demand Q_{engine},

$$Q_{\text{engine}} = \frac{V_d\, N}{120} \cdot \eta_v,$$

the carburetor must sustain an equivalent flow, which determines the ΔP that arises across it.

If a lower ΔP is sufficient to meet Q_{engine}, the carburetor imposes less restriction on the intake system. This reduces pumping work, allows the cylinder pressure during the intake stroke to remain higher, and improves volumetric efficiency and power output. Conversely, if a high ΔP is required to achieve the necessary flow, the engine must expend additional work drawing air through the restriction. This reduces the effective pressure available for combustion and limits the maximum achievable horsepower.

Therefore, ΔP can be viewed as an indicator of how restrictive the carburetor is for a given operating condition: a small carburetor throat will demand a higher ΔP at high flows, while a larger throat reduces ΔP at the same flow. The tradeoff is that some level of pressure drop is also necessary to maintain sufficient air velocity for fuel atomization and throttle response at low engine speeds. The balance between low ΔP for high-rpm breathing and adequate velocity for low-rpm drivability is central to carburetor selection and engine performance.

3 Weber v. Stock 4K Carb

The Weber carburetor has an improved, more efficient design, from its jets, venturis, and linkage system to employ the sequential twin barrel system to counter power loss at high RPMs.

The stock 4K carburetor and intake, with 28mm throttle bores, are designed for the 4K engine, which has a 1.3L displacement compared to the 7K's 1.8L displacement. This results in an underpowered engine with a severe bottleneck that isn't performing at it's full potential.

I have practically observed this in the car that the 4K carb does not support the intake requirements for the 7K motor as the power output was not linear and fell off as the throttle opened more, choking the engine at high rpm.

I prove this explicitly and justify the need for a Weber:

3.1 Throat Area Comparison

Let's compare the throat area of a 4K carb vs a Weber mounted on a 4K intake at wide open throttle (WOT)

The area of the bores determines the volume of air-fuel mixture you can push into the engine. Therefore, power $\propto$ potential flow with both barrels open $\propto$ area.

After measuring the bore diameters using a digital vernier caliper to get accurate measurements instead of relying on factory numbers, we get the following data:

	Primary Bore (mm)	Secondary Bore (mm)
Weber	31.7	35.6
Stock 4K	27.4	27.7

Table 1: Throttle bore diamters in mm

Now, thinking about flow in terms of area, we get:

Weber 32/36 Total Area:

Primary(31.7mm) Area $\approx 790mm^2$
Seconary(35.6mm) Area $\approx 995mm^2$
Total Area: $1785mm^2$

On a 4-cylinder engine, that's an average $446mm^2$ per cylinder.

Stock 4K Total Area:

Primary(27.4mm) Area $\approx 590mm^2$
Seconary(27.7mm) Area $\approx 603mm^2$
Total Area: $1193mm^2$

On a 4-cylinder engine, that's an average $298mm^2$ per cylinder

$$\frac{446mm}{298mm} \approx 1.49$$

So the Weber has approximately 50% more total area at full throttle (both barrels opened to the maximum)

3.2 ΔP Comparison

Below I calculate the required volumetric flow at different rpms for the 7K motor, and the required ΔP across each of the two carburator options to provide that flow rate at WOT.

Q_m is calculated using [5] at various N, with $V_d = 0.0018\text{m}^3$,and η_v =0.8. ΔP is calculated by setting the orifice flow equation [9] equal to Q_m at each N, with C_d =0.6.

The area in [9] has been calculated in the preceding section.

RPM	Q_m (m^3/s)	ΔP_{Weber} (kPa)	ΔP_{4K} (kPa)
2000	0.024	0.295	0.646
3000	0.036	0.664	1.454
4000	0.048	1.181	2.585
5000	0.060	1.845	4.039
6000	0.072	2.656	5.816
7000	0.084	3.616	7.916
8000	0.096	4.723	10.339

Table 2: Required volumetric flow Q_m and pressure drop ΔP for Weber and stock 4K carburetors at WOT.

Note: Q_m is the mean volumetric flow required by the engine at the stated RPM (4-stroke, $Q_m = \eta_v V_d N/120$). ΔP values are computed from the orifice relation $\Delta P = \left(\frac{Q_m}{C_d A}\right)^2 \frac{\rho}{2}$ using the total venturi areas given above and assuming both throats are effectively open (WOT). These are steady, incompressible-orifice estimates (useful for comparison and sensitivity analysis); transient/pulsed and compressibility effects are not included here.

A carburetor becomes a flow-limiting element, or choke, under two primary conditions when the pressure drop (ΔP) required to meet the engine's demand becomes too large.

1. The engine cannot sustain the vacuum needed to pull the air through the restriction.
2. The high pressure drop introduces pumping losses or flow separation, reducing volumetric efficiency (VE) and limiting actual flow compared to the theoretical value.

From Table 2, the 4K carburetor requires a pressure drop of about 7.9–10.3 kPa at 7000–8000 rpm. These numbers are significant, because the carburetor is only one part of the intake path. Additional losses occur in the manifold, runners, and valves, typically adding another 2–4 kPa. When combined, the total pressure drop approaches what the engine can realistically generate, meaning the carburetor becomes the bottleneck. This explains why the engine struggles or feels weak at higher rpm.

By comparison, the Weber 32/36 only needs 3.6–4.7 kPa at the same engine speeds. Because the pressure drop is lower, more of the available intake depression is left for the rest of the system. This reduces pumping losses, helps the engine maintain VE, and avoids the sharp falloff in airflow seen with the smaller carb. In practice, this means more usable power and smoother high-rpm operation.

Therefore, purely from the required ΔP for the mean flow, the 4K carb becomes increasingly stressed at high rpm and is likely to be the limiting element; the Weber provides significantly lower ΔP for the same demanded flow and therefore should produce more power and better drivability at high rpm.

4 Adapter Plate Flow Dynamics

When upgrading from the stock carburetor to the Weber 32/36, an adapter plate is required to match the carburetor's throttle outlets to the engine intake manifold. While the adapter appears to be a simple spacer, its geometry has a direct impact on airflow dynamics, mixture quality, and ultimately engine performance.

The Weber has a 32mm and a 36mm throttle bore, compared to the intake's 28mm and 28mm plenum diameter. This mismatch has a significant impact on the flow dynamics of the air-fuel mixture going into the intake. Such a drastic drop in the cross-section that the mixture impedes the carburetor from delivering peak, efficient performance.

4.1 Sudden Area Changes and Flow Separation

When a flow passes through a duct with an abrupt change in cross-sectional area, flow separation and turbulence occur. Two common cases are relevant:

- **Sudden Contraction (Step-Down):** Flow accelerates sharply through the smaller opening. The contraction causes a vena contracta to form downstream of the step, reducing the effective flow area and increasing local velocity. The loss coefficient is approximated as:

$$K_c \approx \left(\frac{1}{C_c} - 1\right)^2 \tag{10}$$

  where C_c is the contraction coefficient (typically $C_c \approx 0.62$ for sharp-edged orifices) [3].

- **Sudden Expansion (Step-Up):** Flow decelerates into the larger section. If the expansion is abrupt, the jet detaches, creating vortices and energy loss. The head loss is:

$$h_L = \frac{(V_1 - V_2)^2}{2g} \tag{11}$$

  where V_1 and V_2 are velocities before and after the expansion [1].

Both cases result in additional pressure drop, ΔP_{loss}, that is not accounted for in the ideal orifice equation. These losses increase the effective ΔP required across the carburetor, reducing the margin for airflow at higher engine speeds.

4.2 Tapered Transitions

To minimize turbulence and losses, the adapter should provide a gradual transition between different bore diameters. Experimental and design practice suggests that a taper angle of 7°–12° is sufficient to prevent flow separation in both nozzle (contraction) and diffuser (expansion) configurations [4].

A tapered transition serves two purposes:

1. Guides the airflow smoothly, reducing formation of recirculation zones.
2. Recovers static pressure more efficiently in expansions, maintaining mixture density.

Thus, the adapter plate should not be designed as a flat plate with sharp edges, but rather as a short diffuser/nozzle section.

Another critical aspect is alignment between carburetor throats, adapter plate, and intake manifold plenum. Any misalignment introduces "steps" that behave as sudden contractions or expansions, generating turbulence and additional ΔP_{loss}. Port-matching, machining or grinding to ensure smooth alignment, is a standard technique to maximize effective flow area.

If the adapter is poorly designed (sharp step, poor alignment), the added ΔP_{loss} may offset much of the Weber carburetor's theoretical advantage over the stock 4K carburetor. Conversely, a well-designed adapter with tapered transitions preserves the Weber's lower pressure-drop characteristics, enabling higher volumetric efficiency, more consistent fuel atomization, and improved high-rpm drivability.

Taper (diffuser/nozzle) design guideline. For a simple conical taper between two concentric circular sections, the required taper length L (axial length of the cone) is

$$L = \frac{R_1 - R_2}{\tan\theta}, \tag{12}$$

where R_1 and R_2 are the upstream and downstream radii respectively, and θ is the cone half-angle.

4.3 Contraction Losses from Carburetor to Plenum

When flow contracts from the carburetor throats into the plenum entries of 27.4mm and 27.7mm (measured to be the same as the 4K carburetor throttle bores which is expected), pressure losses occur. These losses depend strongly on whether the contraction is abrupt (a sharp step) or gradual (a well-designed tapered cone). Here, I show the effect of an adapter between the Weber and intake manifold plenum by estimating the pressure loss in each case using the Bernoulli equation with loss terms:

$$\Delta P = K_c \frac{1}{2}\rho v^2, \tag{13}$$

where v is the local velocity through the restriction.

Note: This as a demonstration of scale; specific vehicle-installed losses (manifold, runner bends, valve curtain) must be added separately in system calculations.

The engine considered is a 1.8 L four-cylinder, operating at $N = 7000$ rpm with a volumetric efficiency of $\eta_v = 0.80$. The mean volumetric demand is

$$Q = \frac{V_e \cdot \eta_v \cdot N}{120}, \tag{14}$$

where $V_e = 1.8 \text{ L} = 1.8 \times 10^{-3} \text{ m}^3$. This gives

$$Q = \frac{1.8 \times 10^{-3} \cdot 0.80 \cdot 7000}{120} \approx 0.084 \text{ m}^3/\text{s}. \tag{15}$$

The total throat area of the carburetor is

$$\begin{aligned} A_p &= \frac{\pi (0.0317)^2}{4} \approx 7.89 \times 10^{-4} \text{ m}^2, \\ A_s &= \frac{\pi (0.0356)^2}{4} \approx 9.95 \times 10^{-4} \text{ m}^2, \\ A_{\text{tot}} &= A_p + A_s \approx 1.78 \times 10^{-3} \text{ m}^2. \end{aligned}$$

Flow splits proportionally to throat area:

$$\begin{aligned} Q_p &= Q \frac{A_p}{A_{\text{tot}}} \approx 0.037 \text{ m}^3/\text{s}, \\ Q_s &= Q \frac{A_s}{A_{\text{tot}}} \approx 0.047 \text{ m}^3/\text{s}. \end{aligned}$$

4.3.1 Area Ratios

The plenum entry diameter is $d_{2p} = 27.4$ mm and $d_{2s} = 27.7$mm. We define the contraction ratio as:

$$\beta = \frac{d_2}{d_1}. \tag{16}$$

$$\beta_p = \frac{27.4}{31.7} = 0.864,$$
$$\beta_s = \frac{27.7}{35.6} = 0.778.$$

4.3.2 Contraction Coefficients

For abrupt contractions, the contraction coefficient is given empirically [5, 6] by:

$$C_c = 0.62 + 0.38\beta^3. \tag{17}$$

For a tapered (well-shaped cone), the contraction coefficient is given as an empirical constant [5]

$$C_c \approx 0.98$$

.

Primary throat:

$$C_c^{\text{abrupt}} = 0.62 + 0.38(0.864)^3 \approx 0.865,$$
$$C_c^{\text{tapered}} \approx 0.98.$$

Secondary throat:

$$C_c^{\text{abrupt}} = 0.62 + 0.38(0.778)^3 \approx 0.799,$$
$$C_c^{\text{tapered}} \approx 0.98.$$

4.3.3 Loss Coefficient K

The loss coefficient is

$$K = \left(\frac{1}{C_c} - 1\right)^2. \tag{18}$$

Primary:

$$K_p^{\text{abrupt}} \approx 0.024,$$
$$K_p^{\text{tapered}} \approx 0.0004.$$

Secondary:

$$K_s^{\text{abrupt}} \approx 0.063,$$
$$K_s^{\text{tapered}} \approx 0.0004.$$

4.3.4 Pressure Losses

The dynamic head is $\frac{1}{2}\rho v^2$, with $\rho = 1.225$ kg/m^3. Velocities at the plenum entry are

$$\begin{aligned}
A_{plenum\ primary} &= \frac{\pi(0.0274)^2}{4} \approx 5.89 \times 10^{-4} \text{ m}^2, \\
A_{plenum\ secondary} &= \frac{\pi(0.0277)^2}{4} \approx 6.03 \times 10^{-4} \text{ m}^2, \\
v_p &= \frac{Q_p}{A_{plenum}} \approx \frac{0.037}{5.89 \times 10^{-4}} \approx 62.8 \text{ m/s}, \\
v_s &= \frac{Q_s}{A_{plenum}} \approx \frac{0.047}{6.03 \times 10^{-4}} \approx 77.9 \text{ m/s}.
\end{aligned}$$

The corresponding pressure losses are

$$\begin{aligned}
\Delta P_p^{\text{abrupt}} &= K_p^{\text{abrupt}} \cdot \tfrac{1}{2}\rho v_p^2 \approx 0.024 \cdot \tfrac{1}{2}(1.225)(62.8^2) \approx 58.0 \text{ Pa}, \\
\Delta P_p^{\text{tapered}} &= 0.0004 \cdot \tfrac{1}{2}(1.225)(62.8^2) \approx 0.97 \text{ Pa},
\end{aligned}$$

$$\begin{aligned}
\Delta P_s^{\text{abrupt}} &= K_s^{\text{abrupt}} \cdot \tfrac{1}{2}\rho v_s^2 \approx 0.063 \cdot \tfrac{1}{2}(1.225)(77.9^2) \approx 234.2 \text{ Pa}, \\
\Delta P_s^{\text{tapered}} &= 0.0004 \cdot \tfrac{1}{2}(1.225)(77.9^2) \approx 1.48 \text{ Pa}.
\end{aligned}$$

	ΔP Abrupt (Pa)	ΔP Tapered (Pa)
Primary Throat (31.7mm)	58.0	0.97
Secondary Throat (35.6mm)	234.2	1.48

Table 3: Pressure Loss Comparison: Abrupt vs. Tapered

The results show that contraction losses are negligible with a well-designed tapered adapter. Without it, the abrupt contraction at the secondary throat produces a significant loss (≈ 234.2 Pa), which is almost 160x larger than with a tapered adapter (≈ 1.48 Pa). This justifies the use of a dual-taper adapter plate matching each carburetor throat to its respective plenum entry.

Practical Analysis:

- Machine a tapered profile in the adapter so the cone half-angle θ is in the 7°–12° range. For the diameters above, taper length of 15 mm is therefore sufficient to achieve the low K assumed above while remaining compact. This will result in the following primary and secondary cone angles:

$$\theta_p = \arctan\left[\frac{\left(\frac{0.0317}{2}\right) - \left(\frac{0.0274}{2}\right)}{0.0150}\right] = 8.16^\circ \tag{19}$$

$$\theta_s = \arctan\left[\frac{\left(\frac{0.0356}{2}\right) - \left(\frac{0.0274}{2}\right)}{0.0150}\right] = 15.3^\circ \tag{20}$$

- Ensure axial and angular alignment between the carb, adapter, and manifold; remove sharp lips and guarantee a smooth fillet radius at intersections.

5 Adapter Plate Design

In developing the adapter plate, the process began with precise measurement of both the Weber carburetor flange and the intake manifold interface. Initial hand sketches and dimensional diagrams were prepared to explore possible layouts, which were subsequently translated into accurate 2D CAD blueprints. These blueprints formed the basis for the final 3D model of the adapter.

A key design challenge arose from the mismatch in bolt patterns between the carburetor and the manifold. The spacing was such that the holes were too close to permit entirely separate stud locations, yet too far apart to reuse the same studs directly. To resolve this, the design was adjusted so that two of the bolt holes were aligned, allowing shared use of studs at those positions. This alignment provided sufficient clearance to create four additional, independent holes: two dedicated to the manifold side and two for the Weber carburetor. The result was a secure and functional compromise that preserved structural integrity while ensuring proper fitment.

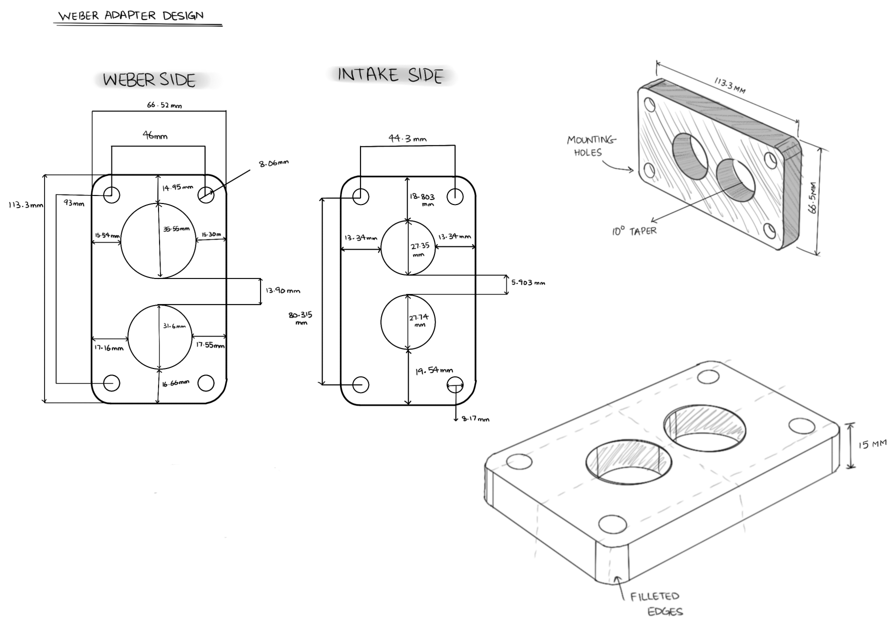
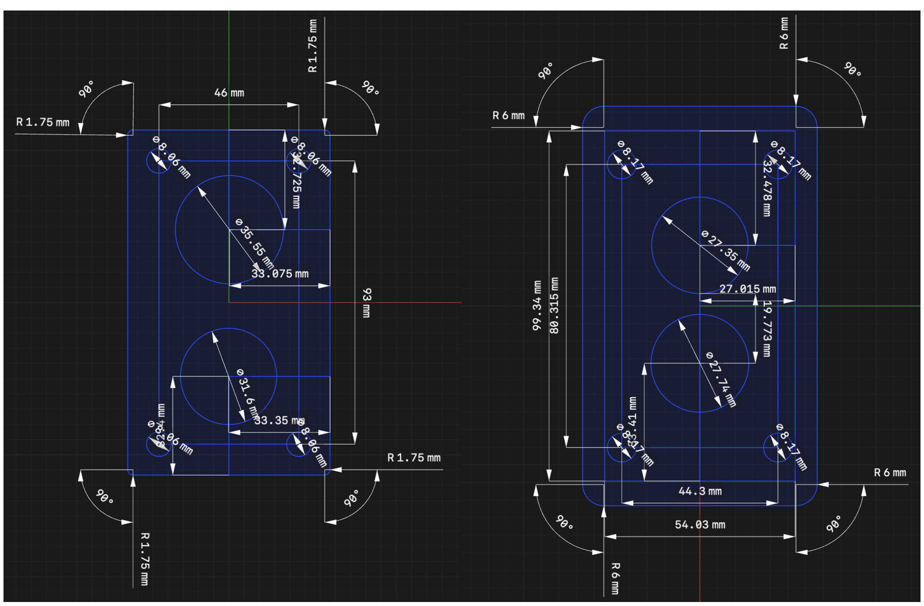


Figure 1: Initial Sketches + Measurements



(a) Weber Side

(b) Intake Side

Figure 2: 2D CAD Blueprints

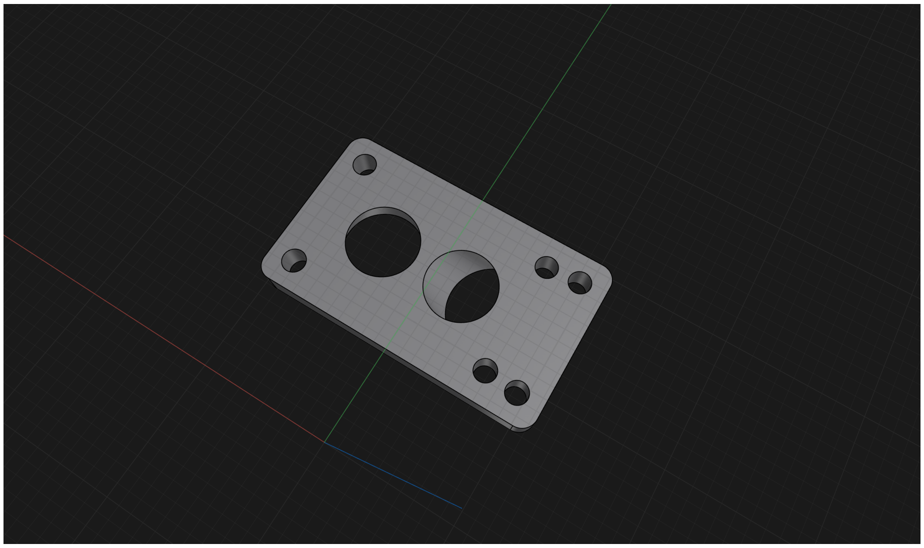


Figure 3: 3D Model Intake Side

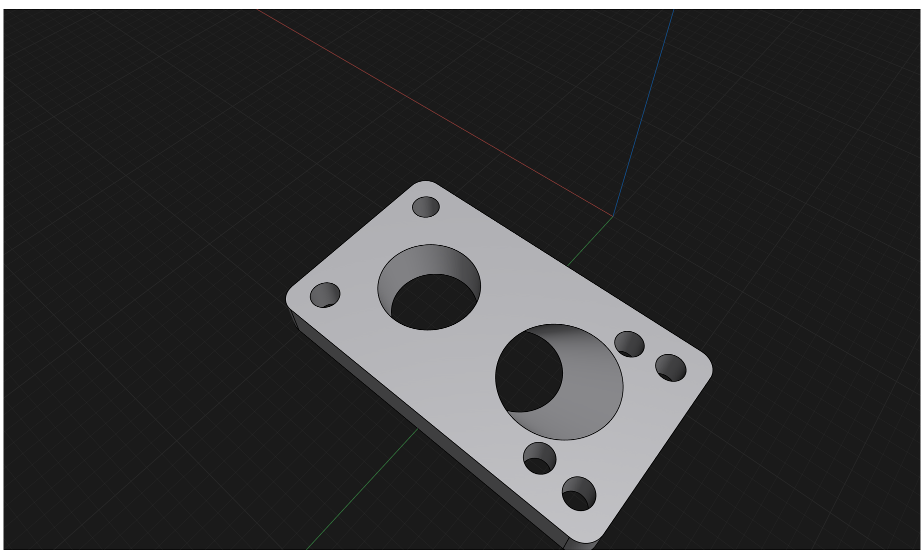


Figure 4: 3D Model Weber Side

6 Scope and Boundaries of Analysis

All baseline calculations in this report (orifice flow, throat-area comparisons, and the adapter loss example) use steady, incompressible flow assumptions with a discharge coefficient C_d. The orifice relation used throughout is

$$Q = C_d\, A \sqrt{\frac{2\Delta p}{\rho}},$$

and its rearranged form for pressure drop

$$\Delta p = \left(\frac{Q}{C_d A}\right)^2 \frac{\rho}{2}.$$

These relations provide simple, conservative engineering estimates that are easy to compare across geometries and operating points. However, they omit several real-world phenomena:

- **Compressibility:** The incompressible assumption becomes less accurate as local Mach number rises (see below). For local Mach 0.3–0.4, compressibility corrections and choked-flow effects should be considered. See Munson [1] and White [3].
- **Transient / pulsed flow:** The intake of a multi-cylinder four-stroke engine is inherently pulsatile. The steady mean flow Q conceals short-duration peaks during an individual cylinder's intake stroke. Pulsation increases local instantaneous velocities and can provoke separation or unsteady losses not predicted by steady formulas.
- **Resonance and acoustic effects:** Intake runner length and geometry produce standing-wave/Helmholtz-like effects that alter cylinder filling (volumetric efficiency) at specific RPM bands. These acoustic phenomena can either augment or reduce instantaneous filling relative to the steady prediction.
- **Geometry-dependent loss coefficients:** In the adapter/taper worked example we used a loss coefficient K and contraction coefficient C_c to estimate local losses ($\Delta p_{\text{loss}} = \frac{1}{2}\rho V^2 K$). These K and C_c values are empirical and depend strongly on edge radii, taper angles and alignment.

These effects mean the true usable mass flow at a given rpm is controlled not only by static throat areas but also by dynamic behaviour.

Other sources of loss: The earlier comparison of carb throat areas and corresponding Δp values treated each carb in isolation (i.e., "in a vacuum"). In a real engine the carb is only the first element in the chain and the following components also consume part of the available pressure drop:

$$\Delta p_{\text{total}} = \Delta p_{\text{carb}} + \Delta p_{\text{adapter}} + \Delta p_{\text{plenum}} + \Delta p_{\text{runners}} + \Delta p_{\text{port/valve}} + \Delta p_{\text{exhaust}}.$$

Because of the items above, steady incompressible estimates should be used as conservative design guidance and to establish relative comparisons (e.g., which carb has more throat area). Final validation requires flow-bench and on-engine MAP/dyno measurements.

7 Instantaneous Runner Velocities and Minimizing Loss

As outlined earlier, the carbureator is only one source of bottleneck in the system. It feeds the air-fuel mixture into an intake manifold, with *runners*, that ultimately feed into the engine. This, being the final component that may limit the engine's breathing potential, requires a thorough analysis.

7.1 Local Mach number and why a velocity target matters

The velocity of intake air within an internal combustion engine must be carefully balanced. Excessively high velocities cause detrimental effects such as flow separation, turbulence, and compressibility losses. As Heywood (1988) [2] notes, once port velocities approach Mach 0.3 (102.9m/s), the assumption of incompressible flow breaks down, leading to sharp pressure losses and reduced volumetric efficiency. Moreover, in carbureted systems, unstable high-speed flow can worsen fuel atomization and promote wall wetting.

Conversely, excessively low velocities are also undesirable. Low-speed flow reduces mixture homogeneity, allows fuel dropout, and diminishes the beneficial inertia charging effect that drives additional cylinder filling after bottom dead center. Stone (2012) [7] emphasizes that adequate port velocity is essential for atomization, suspension of droplets, and strong throttle response.

For design purposes, industry guidelines and experimental data converge on a target mean port velocity of approximately Mach 0.3 - 0.35 at peak volumetric demand. This range ensures that air momentum is sufficient for mixing, without incurring the penalties of compressibility. While modern engines employ computational fluid dynamics and advanced strategies (variable valve timing, direct injection), the Mach 0.3 criterion remains a widely accepted rule of thumb in both carbureted and contemporary EFI engines.

7.2 Calculating Runner Velocities for 30 mm Runners

To quantify the instantaneous flow behaviour and local Mach number in the runner we use the per-cylinder instantaneous intake flow during the intake stroke and the runner cross-sectional area. This provides an engineering check that the **runner diamater of 30mm**(on the stock manifold with which the engine is equipped) keeps local Mach in an acceptable range.

Definitions and relations: For a four-cylinder, four-stroke engine, the required mean volumetric flow is given by Equation [5]. The instantaneous volumetric flow for a single runner is derived similarly as follows:

$$V_c = \frac{V_d}{n_{\text{cyl}}}$$

is the geometric volume per cylinder and

$$V_{\text{charge}} = V_c \cdot \eta_{\text{v}}$$

is the actual charge volume drawn in per intake event when volumetric efficiency is η_v. Approximating the intake duration as 180° crank (0.5 crank rev), the intake time per event is

$$t_{\text{intake}} = \frac{0.5}{N/60} = \frac{30}{N} \text{ (s)},$$

where N is engine speed in rpm. The instantaneous volumetric flow for a single runner during its intake stroke is therefore

$$Q_{\text{inst}} = \frac{V_{\text{charge}}}{t_{\text{intake}}} = \frac{V_c\, \eta_v\, N}{30}.$$

Note that this is the same as Equation [5] with using V_c instead of V_d, divided by 4, since in a 4 stroke engine, only 1 stroke corresponds to an intake stroke per cycle.

The instantaneous runner velocity is

$$V_{\text{inst}} = \frac{Q_{\text{inst}}}{A_{\text{runner}}},$$

with $A_{\text{runner}} = \pi D^2/4$ for a circular runner of diameter D. The local Mach number is

$$M = \frac{V_{\text{inst}}}{a},$$

where a is the local speed of sound (taken as $a \approx 343$ m/s at standard conditions).

Numeric evaluation for the 7K (30 mm runner) Using the 7K geometry and the design target:

The instantaneous flow becomes

$$Q_{\text{inst}} = \frac{V_c \cdot \eta_v \cdot N}{30} = 1.2 \times 10^{-5}\, N \quad (\text{m}^3/\text{s}),$$

so the instantaneous runner velocity is

$$V_{\text{inst}} \approx \frac{1.2 \times 10^{-5}\, N}{7.0686 \times 10^{-4}} \approx 0.01698\, N \text{ (m/s)}.$$

Resulting in the following data:

RPM	V_{inst} (m/s)	$M = V_{\text{inst}}/a$
4000	67.92	0.198
5000	84.90	0.247
6000	101.88	0.297
7000	118.86	0.346

Interpretation and justification From the table:

- With a 30 mm runner, local Mach is approximately $M \approx 0.30$ at $\sim$6000 rpm and ≈ 0.35 at 7000 rpm.
- The commonly used engineering guideline is that compressibility effects and density-variation corrections become relevant for local Mach numbers above ~ 0.3; flows below $M \approx 0.3$ are typically treated as effectively incompressible with small error [3, 1].

Practical conclusion for the 7K A local Mach of ≈ 0.35 at 7000 rpm is not ideal, but because the 7K is a long-stroke, torque-oriented engine, its factory design emphasises mid-range (3-5k rpm) torque, with peak power at 4600 rpm, rather than extreme high-rpm power. This makes the 30mm runners a perfect compromise:

- it keeps instantaneous Mach numbers below or near the conservative threshold up to the high end of the design operating range
- it preserves reasonable air velocity for low- and mid-range torque (avoiding the severe low-speed atomization issues associated with excessively large runners).
- A mach number of 0.35 is only slightly above the advised limit, thus the runners will still prevent severe transient choking at high rpm. Moreover, the engine will only sit at 7000 rpm only momentarily minimizing adverse affects associates with high Mach numbers further

Hence, with a properly tapered adapter and port-matching, a 30 mm runner provides adequate area to realize the Weber's increased throat capacity in the target rpm band while keeping local Mach and compressibility effects at acceptable levels. Final verification should be performed with flow-bench measurements and MAP/dyno testing to capture pulsation and acoustic effects that are not represented in steady estimates [1, 3, 2].

8 Conclusion

This report has examined and explicitly proven that the initial system of a Toyota 7K engine with a 4K carbureator was bottlnecked, justifying the need for a Weber 32/36 DGEV. Examination of the flow dynamics of adapter plates in the context of fitting a Weber 32/36 DGEV carburetor to the Toyota 7K engine was conducted. By analyzing contraction ratios, loss coefficients, and pressure drop across both sharp-step and tapered transitions, the study has demonstrated how seemingly minor geometric details can produce measurable differences in volumetric efficiency and overall airflow delivery.

The findings suggest that the design of the adapter plate plays a decisive role in minimizing flow losses. A well-tapered adapter reduces contraction effects significantly compared to a sharp step, thereby preserving the Weber carburetor's flow advantage. Treating each throat independently also highlights the larger secondary bore as a critical source of potential loss, reinforcing the need for smooth transitions.

While the theoretical calculations establish a framework for understanding these effects, they cannot capture the full complexity of real-world conditions such as turbulence, mixture distribution between cylinders, or the influence of fuel atomization. Therefore, the logical next step is to fabricate the optimized adapter plate and conduct controlled engine tests to compare theoretical predictions with experimental outcomes. Measuring airflow, pressure drop, and ultimately performance metrics such as torque and power output will allow for validation of the analysis and reveal how much of the theoretical efficiency gain translates into practice.

In this way, the study not only provides a deeper academic perspective on adapter plate design but also sets the stage for practical experimentation, bridging theory and real-world application.

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