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Application of a Springless Valve Actuation System to a High-Performance Sports Car Engine

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FINAL VERSION



Masters in Mechanical Engineering

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Abstract

This work investigates the retrofit of a springless, desmodromic valve actuation system to the Porsche 9A1 cylinder head (911 GT3 RS, 992) with the goal of increasing the safe engine speed from 9 000 to 10 500 rpm. The objective is to demonstrate geometric and manufacturing feasibility while preserving the original valves and a production-style carrier architecture.

The study begins with a metrology campaign on the 9A1 cylinder head and a reference desmodromic assembly using a ZEISS (Coordinate Measuring Machine (CMM) with Calypso and targeted 3D scanning. These data drive parametric skeletons in Creo for both assemblies. Valve angles and positions, cam-to-cam packaging and access envelopes are compared. The analysis shows workable packaging, with camshaft-axis offsets of 12.4–12.7 mm (intake/exhaust), confirming the feasibility of dimensional adaptation. The concept is iterated into a closed carrier with two continuous shafts and repeating rocker modules. A block-out strategy from billet is defined to enable machining of continuous flanges, bores and service windows.

The final layout coexists with the original valves without interference and provides coherent assembly and inspection access. The remaining open item is the rocker-on-shaft axial retention and end-float definition. Feasibility is therefore established, contingent on closing the fixation details and completing structural, tribological, lubrication/thermal and dynamic validation to 10 500 rpm.

Key Words: Desmodromic System, Springless Valve Actuation, Porsche 9A1, 911 GT3 RS, 10,500 RPM, ZEISS CMM Metrology, 3D Scanning, Parametric CAD In Creo, Rocker Arms, Camshafts, Structural And Tribological Validation

Resumo

Este trabalho investiga a adaptação de um sistema de atuação de válvulas sem molas, de arquitetura desmodrómica, à cabeça do motor Porsche 9A1 (911 GT3 RS, 992), com o objetivo de elevar a rotação máxima de 9 000 para 10 500 rpm. Pretende-se demonstrar a viabilidade geométrica e de fabrico, preservando as válvulas originais e um porta-válvulas com carácter de produção.

O estudo inicia-se com levantamento geométrico na cabeça do 9A1 e num conjunto desmodrómico de referência, usando uma MMC ZEISS com Calypso e digitalização 3D dirigida. Estes dados alimentam modelos paramétricos em Creo para ambos os conjuntos. Comparam-se ângulos e posições das válvulas, o empacotamento entre eixos de comando (veios de cames) e os envelopes de acesso. A análise confirma um empacotamento viável, com desvios entre eixos de comando de 12,4–12,7 mm (admissão/escape), validando a viabilidade da adaptação dimensional. O conceito evolui para um porta-válvulas fechado, com dois veios contínuos e módulos repetitivos de balancins, e é convertido para uma abordagem “block-out” a partir de tarugo, definindo flanges contínuas, furos e janelas de serviço usináveis.

A configuração final coexiste com as válvulas originais sem interferências e assegura acessos coerentes para montagem e inspeção. Permanece em aberto a retenção axial dos balancins no veio e a definição da folga terminal. Assim, a viabilidade fica estabelecida, condicionada ao fecho dos detalhes de fixação e à validação estrutural, tribológica, de lubrificação/térmica e dinâmica até 10 500 rpm.

Palavras Chave: Sistema Desmodrómico, Atuação de Válvulas Sem Mola, Porsche 9A1, 911 GT3 RS, 10 500 rpm, Metrologia CMM ZEISS, Digitalização 3D, CAD paramétrico em Creo, Balancins, Cames, Validação Estrutural e Tribológica

UN Sustainable Development Goals

The United Nations Sustainable Development Goals (**SDGs**) establish a global framework to deliver prosperity within planetary limits by 2030 [1]. In road transport—responsible for a substantial fraction of energy-related CO₂ emissions—engineering interventions that reduce fuel consumption and tailpipe emissions are central to SDG delivery [2]. Within this context, the present thesis proposes a springless (desmodromic) valve actuation architecture for a high-performance spark-ignition engine (Porsche 9A1), aiming to reduce mechanical losses while enabling operation at elevated rotational speeds. The discussion below replaces purely formal alignment with an evidence-oriented projection that quantifies plausible efficiency and CO₂ benefits and links them explicitly to SDGs 7, 9, 11, 12 and 13.

Purpose and caveat: At this stage, no fuel-consumption or regulated-emission tests have been performed with the desmodromic system. The sustainability assessment is therefore framed as a first-order projection grounded in standard engine performance relationships and literature values for friction partitioning [3–5]. All figures are indicative and will require confirmation by motored and fired testing in the validation phase.

Method: first-order energy model For steady operating points,

$$\text{BMEP} = \text{IMEP} - \text{FMEP} - \text{PMEP}, \quad (1)$$

where IMEP is indicated mean effective pressure, FMEP is friction MEP, and PMEP is pumping MEP [3]. For a small reduction ΔFMEP at fixed brake torque (fixed BMEP), the fractional fuel reduction is, to first order,

$$\frac{\Delta\dot{m}_f}{\dot{m}_f} \approx \frac{\Delta\text{FMEP}}{\text{IMEP}} = \frac{\Delta\text{FMEP}}{\text{BMEP} + \text{FMEP} + \text{PMEP}}, \quad (2)$$

holding cycle efficiency constant [3, 4]. Literature indicates that the valvetrain contributes roughly 10–25% of total FMEP, increasing with speed as spring compression work and cam–follower losses scale with lift rate and frequency [3, 5]. By removing coil-spring compression work and reducing closing-flank normal loads, a desmodromic system may plausibly reduce the valvetrain portion of FMEP by about 15–30%, conditional on contact and lubrication design [6–8].

Assumptions (transparent and conservative)

- **Operating points:** Urban cruise (part-load), Highway (higher BMEP), and Track (high speed/high load). Baseline BMEP–FMEP–PMEP values are chosen within typical ranges for high-performance spark-ignition engines [3, 4].
- **Valvetrain share of FMEP:** 20% at Urban/Highway, 25% at Track [3, 5].

Table 1: First-order projection of friction and BSFC/CO₂ savings.^a

Operating point	BMEP	FMEP	PMEP	Valvetrain	ΔFMEP	ΔBSFC/CO ₂
	[bar]	[bar]	[bar]	share [%]	[bar]	[%]
Urban cruise	3.00	1.20	0.30	20	0.072	1.60
Highway load	12.00	1.80	0.10	20	0.108	0.78
Track / high speed	13.00	2.20	0.05	25	0.165	1.08

^a Assumes 30% reduction of valvetrain friction; Delta FMEP = $0.30 \times (\text{valvetrain share}) \times \text{FMEP}$. Percentage saving computed as Delta FMEP divided by (BMEP + FMEP + PMEP), using Equation (2) [3, 4].

- **Desmodromic benefit:** 30% reduction of the valvetrain share of FMEP (upper bound within the 15–30% range) [6, 7].

Projected results (BSFC and CO₂): Applying the relation in Equation (2), Table 1 reports projected brake-specific fuel consumption (BSFC) and CO₂ percentage reductions for each duty point. Because tailpipe CO₂ is essentially proportional to fuel used, the percentage CO₂ reduction mirrors the BSFC reduction [3]. For illustration (not a duty-cycle average): a vehicle consuming 10.0 L/100 km over 10,000 km (approximately 1,000 L per year) would save about 8–16 L per year for the range below; using a representative petrol factor of 2.31 kg CO₂ per litre yields about 18–37 kg CO₂ per year avoided [9].

Context of the projected BSFC/CO₂ effect The estimated 0.8–1.6 % BSFC/CO₂ reduction is modest in absolute terms but consistent with efficiency gains reported when valvetrain losses are reduced by advanced actuation. Studies on pneumatic or fully flexible actuation show sub-percent to low single-digit improvements at steady points once spring work and surge margins are removed or lowered [4, 10, 11]. The present architecture pursues the same mechanism by eliminating spring compression work; added sliding losses are contained by materials, coatings and film management [8, 12].

Explicit SDG linkage with quantified estimates The projections above enable an evidence-oriented linkage to specific SDG targets:

- **SDG 7 — Affordable and Clean Energy (Target 7.3, efficiency).** The projected 0.8–1.6% BSFC reduction across representative points constitutes a direct end-use energy-efficiency gain, contingent on delivering the friction deltas in rig testing [3–5].
- **SDG 9 — Industry, Innovation and Infrastructure (Target 9.4).** The architecture eliminates coil-spring hysteresis and reduces closing-flank loads; the assumed 15–30% cut in the valvetrain portion of FMEP is consistent with desmodromic positive closure provided materials/coatings and lubrication are engineered appropriately [6–8].
- **SDG 11 — Sustainable Cities and Communities (Target 11.6).** While regulated pollutants are not predicted here, lower frictional losses reduce fuel burned for a given service, hence proportionally lower CO₂; pollutant impacts require empirical characterisation with the final timing strategy [3, 5].
- **SDG 12 — Responsible Consumption and Production (Target 12.2).** Fewer wear-intensive springs and reduced parasitic work can lower material and maintenance footprints over life; durability and maintainability will be established via the stage-gated validation plan [8].

Table 2: Contribution to selected SDGs: targets addressed, mechanism of impact, and example metrics.

SDG	Target (excerpt)	Contribution of springless (desmodromic) valvetrain	Example indicators/metrics
SDG 7	7.3 — Double the global rate of improvement in energy efficiency	Reduces mechanical (valvetrain) losses, improving fuel economy and end-use energy efficiency; see Table 1 [3, 4].	Percent reduction in valvetrain FMEP; BSFC change at key points; energy per kilometre.
SDG 9	9.4 — Upgrade infrastructure/industry for resource-efficient, clean technologies	Eliminates coil springs and their hysteretic loss pathway; lowers closing-flank loads; integrates within existing head architecture, easing retrofit [6, 8].	Component mass reduction; process energy for manufacture; kg CO ₂ per engine produced.
SDG 11	11.6 — Reduce cities' environmental impact; improve air quality	Lower fuel use for a given service reduces CO ₂ ; pollutant outcomes to be determined with final timing strategy [3, 5].	Tailpipe CO ₂ (g/km); WLTP fuel use; regulated pollutant indices after optimisation.
SDG 12	12.2/12.5 — Efficient resource use; reduce waste via prevention and durability	Fewer springs and wear-prone parts; potential to extend overhaul intervals via lower parasitic work and tailored tribology [8].	Mass of replaced components per 100,000 km; overhaul interval; recyclability of materials.
SDG 13	13.2 — Integrate climate measures (lower emissions)	Direct proportional reduction in CO ₂ with BSFC; pathway to lower carbon intensity of performance ICEs pending validation [3, 9].	Percent CO ₂ reduction at points; annual kg CO ₂ avoided per vehicle; fleet-scale extrapolation.

- **SDG 13 — Climate Action (Target 13.2).** The BSFC reduction maps 1:1 to tailpipe CO₂. Using the simple illustration above, savings of CO₂ per vehicle per year are plausible under the assumed usage; fleet-level impacts depend on duty cycle and adoption [3, 9].

SDG targets, contribution and metrics (summary table) Table 2 consolidates the relevant SDG targets, the contribution of the proposed system, and measurable indicators appropriate for postgraduate research and early-stage industrialisation.

Sensitivities, limitations and required validation The dominant sensitivities are: (i) the valvetrain share of total FMEP (assumed 20–25%) and its speed dependence; and (ii) the achievable reduction (15–30%) of that share with desmodromic closure given surface finish, coatings and oil viscosity at temperature. The projection neglects second-order interactions (for example, oil-pump power and accessory drives) and does not model combustion/emissions changes; these must be addressed experimentally [3, 4, 8]. Immediate steps to convert projection into evidence are:

1. Motored friction mapping on a cam-in-head rig with torque instrumentation to measure Δ FMEP versus speed and confirm the assumed reduction of the valvetrain share [3].
2. Tribology screening of closing-flank and pivot contacts (elastohydrodynamic film parameter λ , friction coefficients) to validate the targeted reduction with the selected materials/coatings/oil [7, 8].

3. Engine/chassis dynamometer BSFC and CO₂ mapping (steady-state and WLTP-aligned) to replace projections with measured deltas and close the SDG claims with evidence [3, 4, 9].

The linkage to SDGs is here recast on a quantitative basis: a plausible 0.8–1.6% BSFC/CO₂ reduction across representative points is estimated, subject to validation. This reframes sustainability from a formal alignment to an evidence-oriented estimate consistent with the proof-of-concept status of the present work [3–9].

Per-vehicle CO₂ savings are small. The main benefit is strategic. The springless valvetrain raises safe speed to 10 500 rpm and keeps high-revving, naturally aspirated engines relevant for sports use during the shift to electrification. It preserves qualities valued in this niche: sharp response, linear delivery, and repeatable durability. It also prepares a clear base for future work with low-carbon fuels. In short, the environmental case is indirect: the system helps this class remain viable while opening paths to later lifecycle gains through fuel choice and duty-cycle optimisation.

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Beatriz Miranda

"Simply lovely"

Max Verstappen

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List of Acronyms

CMM	Coordinate Measuring Machine
SDG	Sustainable Development Goals
BMEP	Braking Mean Effective Pressure
FMEP	Friction Mean Effective Pressure
IMEP	Indicated Mean Effective Pressure
PMEP	Pumping Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CAD	Computer-Aided Design
RPM	Rotation Per Minute
VVT	Variable Valve Timing
VVL	Variable Valve Lift
OHV	Overhead Valve
OHC	Overhead Camshaft
SOHC	Single Overhead Camshaft
DOHC	Double Overhead Camshaft
VTEC	Variable Valve Timing and Lift Electronic Control
BMW	Bayerische Motoren Werke
VANOS	Variable Nockenwellensteuerung
VVTL-i	Variable Valve Timing and Lift Intelligent System
CCVTL	Continuously Variable Valve Timing Lift Systems
VVEL EMVA	Electromagnetic Valve Actuation
ICE	Internal Combustion Engine
GP	Grand Prix
DLC	Diamond Like Carbon
GD&T	Geometric Dimensioning and Tolerancing
EHL	Elastohydrodynamic Lubrication
ADT	Abstract Data Type
ANDF	Architecture-Neutral Distribution Format
API	Application Programming Interface
CASE	Computer-Aided Software Engineering
PS	Pferdestärke (Horsepower)
FEA	Finite Element Analysis

Chapter 1

Introduction

Motorsport engines demand maximum speed, efficiency, and reliability, yet conventional spring-based valvetrains impose limits through float, surge, and energy loss. This work addresses that constraint by evaluating a springless, desmodromic system applied to the Porsche 9A1 cylinder head. The study defines scope, objectives, and constraints, providing a structured basis before reviewing high-performance valve actuation in motorsport.

1.1 High-performance valve actuation in motorsport

High-revving, naturally aspirated sports-car engines place exceptional demands on the valvetrain: precise timing at very high rotational speeds, low reciprocating mass, and durable contact interfaces under aggressive accelerations. Conventional spring-closed systems are ultimately constrained by valve float, surge and frictional losses in the springs, which motivate alternatives that preserve positive control of the valve throughout the lift event. Desmodromic (springless) actuation—where opening and closing are cam-controlled—has been proven in top-tier motorcycle applications to sustain higher engine speeds by permitting higher valve accelerations with reduced risk of float and reduced parasitic losses. Transferring these advantages to a high-performance automotive cylinder head is therefore a compelling pathway to increase safe engine speed while maintaining reliability.

1.2 Project context and scope

This dissertation was developed during an academic mobility placement at the Kolbenmaschinen-Labor (K-MA LAB) of Hochschule Heilbronn, in Germany. The laboratory context provided access to specialised metrology equipment, which allowed the project to be framed with industrial relevance and supported by experimental evidence.

Two baseline artefacts define the project scope: (i) a production 9A1 Gen 5 cylinder head (4.0 L flat-six) with known geometry and valvetrain arrangement; and (ii) a desmodromic head (motorcycle) used to derive rocker and cam topologies, clearances and stacking constraints for

a springless mechanism. The thesis frames the problem as a packaging and kinematics challenge—preserving the production envelope whilst substituting a positive-closure valvetrain—and identifies lubrication, thermal management and structural verification of any new carrier as integral to feasibility. A consistent nomenclature of parts (cams, long/small rockers, rocker shafts, valve stack, stopper) is established to ensure unambiguous communication across measurement, CAD and assembly tasks.

1.3 Motivation

The motivation is twofold. First, to retire the dominant speed-limiting failure modes of spring-based valvetrains (valve float, surge, retainer loads) by enforcing positive valve control, as supported by fundamentals and by sustained motorsport practice in desmodromic engines. Second, to demonstrate that such a mechanism can be engineered into a production-grade, high-specific-output automotive head without compromising manufacturability, serviceability or system interfaces (timing drive, lubrication, cooling). The potential pay-offs—higher safe rpm, improved top-end breathing and reduced parasitic losses—are directly aligned with the performance character of a GT3-class power unit.

1.4 Dissertation Objectives

The work pursues measurable objectives that convert the high-level ambition into engineering deliverables:

- Increase the safe maximum engine speed from 9,000.00 rpm to 10,500.00 rpm via springless (desmodromic) actuation, subject to production envelope constraints;
- Define a complete desmodromic assembly (cams, opening/closing rockers, shafts, valve stack and ancillary stoppers) and the associated datum/parameter framework to drive CAD and metrology.

1.5 Document Structure

To provide a clear understanding of the steps taken throughout the work, this document is organized into eight chapters, each of which is described below. Chapter 2 surveys valve-actuation systems with emphasis on desmodromic architectures, positioning the proposed concept against established spring-closed and alternative springless solutions. Chapter 3 defines the engineering problem, baseline hardware and success criteria, including the rpm target and the nomenclature used throughout. Chapter 4 documents measurements on the production head and on the desmodromic donor hardware, acquired on a ZEISS CMM with Calypso, and reduced to the controlling dimensions needed for packaging and kinematics. Chapter 5 presents the concept-development

iterations and CAD parameterisation, leading to a packaged desmodromic layout and a final assembly study, with explicit discussion of unresolved design risks to be retired in future work. Finally, in Chapter 6, the conclusions for this work are made and the future works described.

1.6 Design Flow

The work follows an evidence-led sequence: define a traceable geometric baseline on the 9A1 head (ZEISS/Calypso); create top-down Creo skeletons for the 9A1 and the desmodromic donor; iterate concepts within measured constraints to address cam support, rocker-shaft alignment, manufacturability and assembly, logging unresolved items (e.g., rocker axial retention) as risks; and verify spatial compatibility with the original valves via an integrated assembly study, carrying residual uncertainties into detailed design.

This thesis reports a demonstrated adaptation of a springless, desmodromic valve actuation system to the Porsche 9A1 cylinder head while retaining the original valves and a production-style carrier. It establishes a traceable geometric baseline, encodes it in parametric CAD skeletons, and delivers an integrated packaging and manufacturability study. The work confirms geometric feasibility for operation targeted at 10 500 rpm and defines the remaining design risks for closure in structural, tribological and dynamic validation.

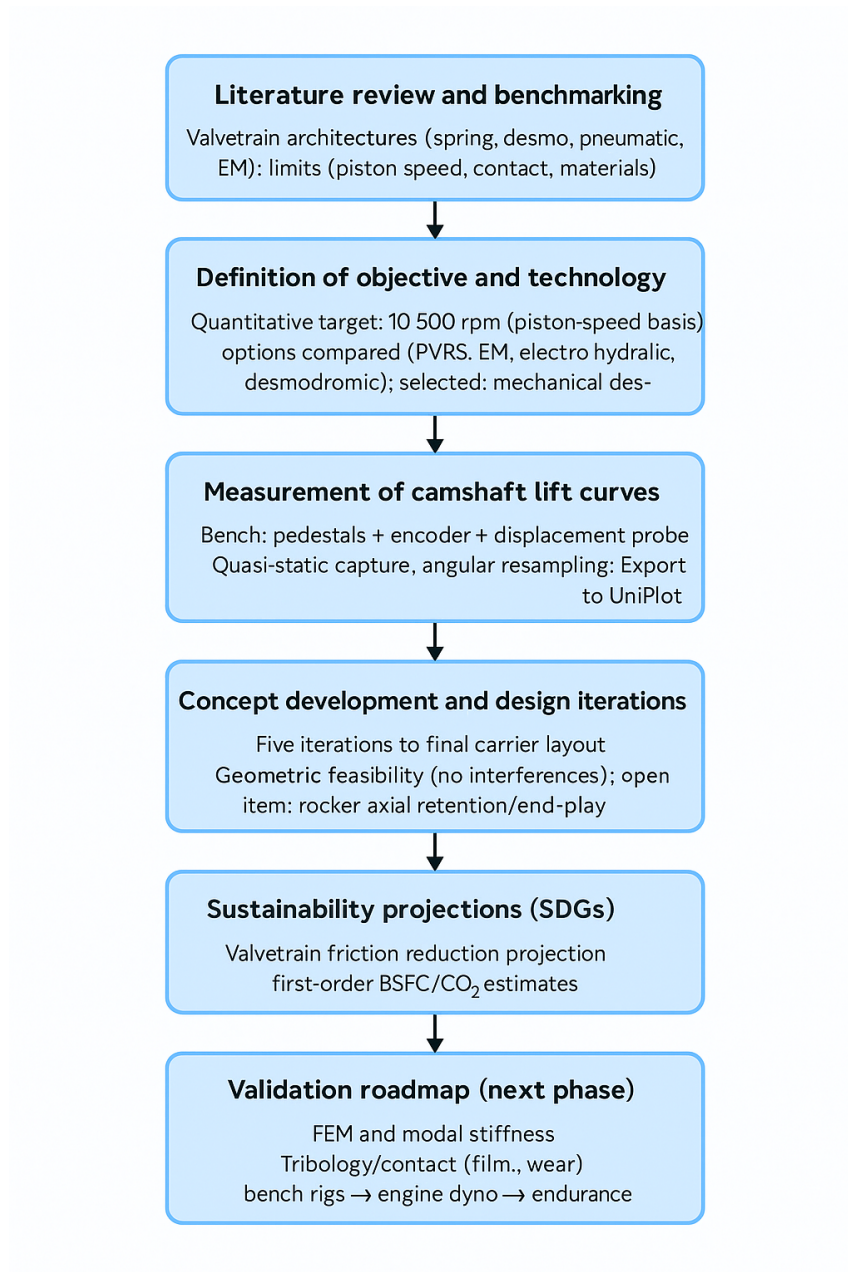


Figure 1.1: Design flow of the project

Chapter 2

State of Art

This chapter sets the scene for the work by outlining how valve actuation has evolved, where conventional spring-based systems reach their limits at high engine speed, and why a springless, desmodromic approach is an attractive alternative. It also places the project in context by touching on Porsche's GT3 RS powertrain and the packaging realities of adapting a racing-style mechanism to a production-derived head. In short, it explains what exists today, what the trade-offs are, and why a desmo concept is worth testing here.

2.1 Introduction

Valve actuation systems play a crucial role in controlling the intake and exhaust processes of an internal combustion engine. Over the years, various valve actuation methods have been developed to enhance engine performance, reduce valvetrain energy losses, and increase overall efficiency. Among these, the desmodromic valve actuation system stands out as a unique solution that eliminates the need for traditional valve springs, instead using positive mechanical closure of valves. This chapter reviews the evolution, classification, and impact of different valve actuation technologies, with a specific focus on desmodromic systems as a springless valve actuation approach in high-performance engines.

2.2 Background on Sports Context

In recent decades, high-performance sports car engines have undergone considerable advancements aimed at maximising power output, improving reliability, and optimising efficiency. Modern engine development featured lightweight components and sophisticated electronic controls, alongside innovative valvetrain designs [5]. For example, Porsche's 911 GT3 RS epitomises these trends with a highly refined valvetrain capable of sustaining over 9000 rpm, thanks to lightweight valve components and a rigid valvetrain geometry. Such design allows exceptional power density and throttle response in track-oriented vehicles while maintaining reliability at extreme engine

speeds.

Parallel to Porsche's approach, Ducati's Desmodromic valvetrain system offers an example of specialised valvetrain engineering. The Desmodromic system uniquely employs cam-driven opening and closing actions, eliminating conventional spring mechanisms, thus allowing precise valve timing at high rotational speeds without the common limitation of valve float [13]. This design significantly reduced mechanical losses and improved reliability at high RPM, characteristics highly desirable in competitive racing contexts. Ducati's success with the Desmodromic system underscores the potential advantages of springless valve actuation, particularly in engines that regularly operate under extreme conditions encountered in motorsport environments [14].

In the broader context, the continued evolution of valve train technology, including innovations such as springless actuation systems, holds substantial promise for future sports engine development. Integrating a springless valve actuation system, as exemplified by Ducati's Desmodromic arrangement, into high-performance automotive engines similar to those in the Porsche GT3 RS presents a promising avenue for further enhancement of performance. Such systems could potentially deliver high precision, improved durability, and greater engine speeds, directly contributing to competitiveness in motorsport applications [5].

2.3 Overview of Valve Actuation Systems

Valve actuation mechanisms are responsible for opening and closing the intake and exhaust valves in synchronisation with the engine cycle.

Typical traditional systems rely on the movement of a camshaft-drive system, where the lobes of the camshaft push rocker arms/lifters to open and close the valves at precise intervals. Most engines use coil springs to return the valves to their closed position, ensuring proper sealing and maintaining compression. However, when reaching higher speeds, the valve tends to float - a condition where the valve does not close properly due to inertia- which can occur, limiting performance, considering this results in improper sealing and loss of efficiency.

To counteract these limitations, manufacturers have developed advanced valve actuation systems, such as variable valve timing (VVT) and variable valve lift (VVL) technologies. These systems can dynamically adjust the timing or lift of the valves depending on the load and speed of the engine.

An alternative approach to conventional valve actuation is the development of springless valvetrains, which aim to eliminate the limitations of mechanical springs. Technologies such as desmodromic valve actuation, used by Ducati, employ a secondary cam and rocker to forcibly

close the valve, preventing its floating and allowing higher rpm operation. More recent advancements include electromagnetic, electrohydraulic and pneumatic valve actuation systems, which allow fully independent control of each valve without the constraints of camshafts or springs. These systems offer greater flexibility in optimising combustion efficiency, improving throttle response, and reducing mechanical losses in high-performance and racing applications.

2.4 Historical Evolution of Valve Actuation Systems

The history of valve actuation systems dates back to the earliest internal combustion engines, which primarily used side-valve (flat head) configurations [15]. In these early designs, the intake and exhaust valves were positioned beside the piston, leading to compact and simple engine layouts. However, this design suffered from poor airflow dynamics and inefficient combustion. As engine technology advanced, manufacturers transitioned to overhead valve (OHV) systems, where valves were positioned in the cylinder head and operated by pushrods connected to a camshaft located in the engine block [5, 16]. This layout significantly improved airflow and efficiency, enabling higher power outputs.

By the 1950s and 1960s, high-performance and racing applications demanded further refinement. This led to the widespread adoption of overhead camshaft (OHC) systems, where camshafts were placed directly above the valves. Single overhead camshaft (SOHC) systems were common in mass-market vehicles, while double overhead camshaft (DOHC) configurations gained popularity in sports and performance engines. DOHC layouts allowed the use of four valves per cylinder and improved high-RPM breathing due to reduced valve-train mass and more precise valve timing [5]. Despite their advantages, these systems had fixed valve timing, lift, and duration, which forced compromises between low-end torque and high-end power [17].

The need to improve fuel efficiency and meet stricter emissions regulations from the 1980s onwards drove the development of variable valve timing (VVT) technologies. These systems introduced the ability to alter the camshafts' rotational position relative to the crankshaft, thereby changing the timing of valve opening and closing events. Early implementations, such as Alfa Romeo's system in 1980, Honda's VTEC in 1989, and BMW's VANOS in 1992, enabled engines to operate more efficiently across a broader range of speeds and loads [18, 19]. These VVT systems improved drivability, emissions, and fuel consumption without sacrificing peak performance [19].

By the late 1990s and early 2000s, manufacturers began combining variable timing with variable valve lift and duration [20]. These systems allowed for more adaptive control of engine breathing. Honda VTEC's system, for example, introduced multiple cam profiles that the engine could switch between under operating conditions. At low RPM, a low-lift profile would be used

for efficiency, while at higher speeds, the system would switch to a high-lift profile to increase performance. Other manufacturers, including Toyota with VVTL-i and Mitsubishi with MIVEC 2.1, adopted similar dual-profile switching systems [20]. These solutions provided improved versatility but remained limited by the discrete nature of the cam profile transitions.



Figure 2.1: Mitsubishi MIVEC Engine [21].

To overcome these limitations, the automotive industry advanced towards continuously variable valve timing and lift systems (CVVTL). Unlike systems that operated in discrete steps, CVVTL technologies allowed seamless modulation of valve optimisation. BMW's Valvetronic, introduced in 2001, provided continuous control over intake valve lift, enabling throttle-less operation and significant reductions in pumping losses. Nissan's VVEL and Hyundai's CVVL employed mechanical linkages and control shafts to achieve continuous lift and duration variation, often working in concert with traditional cam phasing systems [22].

In 2009, Fiat introduced the MultiAir system 2.2, a notable innovation using electrohydraulic actuators between the camshaft and the intake valves. This design enabled cylinder-by-cylinder, cycle-by-cycle control of valve events, allowing the system to optimise performance and emissions dynamically. It represented an important milestone on the path toward camless operation [23].



Figure 2.2: Fiat MultiAir System [24].

The most recent and revolutionary development in valve actuation is the emergence of fully variable, camless systems. These systems eliminate the mechanical camshaft and control valve motion through electromagnetic, electro-hydraulic, or pneumatic means [25].

One of the most advanced applications of camless technology is the FreeValve system 2.3, developed by Koenigsegg [26]. FreeValve employs a pneumatic-hydraulic-electric actuation method that allows full, independent control of each valve in real time. This enables highly advanced engine operation strategies, including per-cylinder valve events, skip-firing, and hybrid combustion cycles. Although still rare in production vehicles due to cost and complexity, FreeValve demonstrates the immense potential of camless, springless valve actuation in high-performance engines [27].

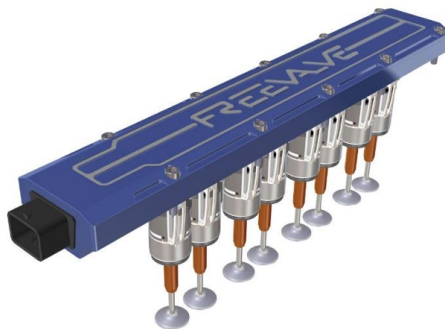


Figure 2.3: Honda VTEC system [28].

Pneumatic valve return systems have also been used extensively in high-revving racing engines, such as those in Formula 1. These systems replace conventional valve springs with compressed air to return the valve, allowing extremely high engine speeds without valve float. Although not camless in nature, these pneumatic systems represent an essential step in reducing valvetrain limitations at the upper extremes of engine performance [29].

The historical trajectory of valve actuation technology reveals a steady shift from fixed mechanical control towards highly adaptive, software-managed systems [20]. This evolution reflects broader trends in engine design, where mechanical limitations are increasingly replaced by electronically governed precision. In the context of high-performance sports car engines, the integration of springless, camless valve actuation systems stands as a logical next step in achieving peak efficiency, rapid responsiveness, and full real-time control of the combustion process [27].

2.5 Types of Valve Actuation Systems

A practical way to frame valve-actuation technologies is to group them by how much freedom they give over the valve event and by what provides the return force. First are fixed-timing cam–spring systems (OHV/pushrod, SOHC, DOHC) where a lobe with a single, non-changing profile opens the valve and coil springs close it—simple, robust, but inherently a compromise across the rev range. Next are variable systems, which modify timing, lift, or both: cam-phasing VVT, discrete or continuous VVL (e.g., multi-lobe switching vs. eccentric/lever mechanisms), and combined VVT+VVL schemes that broaden torque and efficiency while adding actuators and control complexity. Finally are springless/camless approaches that seek to escape spring-inertia limits altogether—pneumatic valve return, mechanically positive closure (desmodromic), and electronically controlled camless concepts (electromagnetic/electro-pneumatic) promising per-valve, real-time control but with significant cost, packaging, and durability challenges.

2.5.1 Fixed Timing Systems

Fixed timing systems in internal combustion engines represent the most traditional approach to valve actuation. These systems operate in the intake and exhaust valves according to a predefined cam profile that is mechanically linked to the crankshaft, without any capability for altering valve timing, duration, or lift dynamically during engine operation [16]. Although modern engine design increasingly favours variable valve timing systems, fixed timing configurations remain relevant for their mechanical simplicity, reliability, and historical significance. Within fixed timing architectures, three principal configurations are prevalent: cam-in-block (pushrod) systems, single overhead camshaft (SOHC) engines, and double overhead camshaft (DOHC) engines [5]. Each of these has distinct mechanical layouts and performance characteristics.

2.5.1.1 Cam-in-Block Systems (Pushrod Engines)

Cam-in-block systems, more commonly referred to as pushrod engines, position the camshaft within the engine block, typically close to the crankshaft. Valve actuation in this arrangement relies on lifters or tappets that ride directly on the camshaft lobes. These transfer motions vertically through long pushrods to rocker arms mounted on the cylinder head, which in turn open

and close the intake and exhaust valves. This design was particularly prevalent in American automotive engineering, with General Motors' small-block V8 engines- first introduced in 1955- standing as one of the most iconic examples [17, 30]. The architecture's main advantage lies in its compactness: by keeping the camshaft in the block, the cylinder head remains relatively small and low-profile, which benefits vehicle design and contributes to a lower centre of gravity [20]. This makes pushrod engines especially well-suited for rear-wheel-drive vehicles with long hoods, such as traditional muscle cars and trucks. In addition, the reduced number of components in the cylinder head simplifies manufacturing and, in certain applications, can improve durability and serviceability [15].

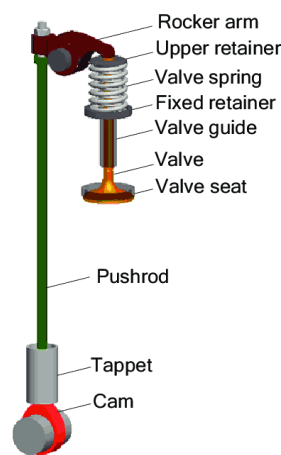


Figure 2.4: Pushrod System.

Despite their compact packaging and torque-rich performance characteristics at low RPM, pushrod engines suffer from mechanical limitations when operated at higher speeds. The longer valvetrain, including lifters, pushrods, and rocker arms, introduces greater inertia, which restricts maximum RPM due to the risk of valve float and timing imprecision [31]. Moreover, because the pushrod system actuates valves indirectly and lacks the spatial flexibility of overhead configurations, incorporating more than two valves per cylinder is significantly more complex. These limitations ultimately constrain both breathing efficiency and volumetric performance [5]. Nevertheless, pushrod systems remain in use in certain high-displacement V8s in North America, particularly where simplicity, cost-efficiency, and torque output are prioritised over peak power and variable valvetrain technology [30].

2.5.1.2 Single Overhead Camshaft (SOHC)

The single overhead camshaft configuration was developed in the middle of the 20th century in response to demands for more precise valve timing, improved fuel efficiency, and higher RPM operation [16, 20]. The camshaft is situated in the cylinder head, just above the combustion chamber, in this design. Usually, it drives the intake and exhaust valves directly through bucket

tappets or finger followers, or indirectly through rocker arms. The stiffer, lighter valvetrain that SOHC engines achieve by doing away with pushrods allows for more dependable operation at higher engine speeds. In addition to improving valve timing precision, the fewer components between the camshaft and valves also lower parasitic losses.

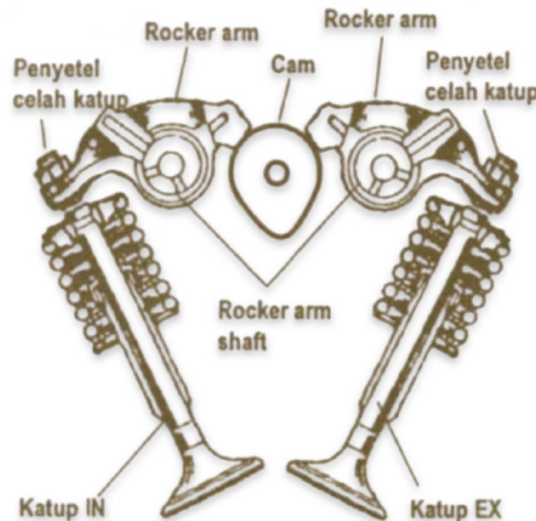


Figure 2.5: Single Overhead Camshaft System [32].

Additionally, SOHC engines provide crossflow cylinder head designs and easier access to more efficient combustion chamber shapes, which enhance power output and thermal efficiency [15], [5]. Furthermore, three or four valves per cylinder arrangements can be supported by modifying the SOHC design [16]. However, the tuning flexibility is naturally constrained because a single camshaft must regulate both intake and exhaust valve actions. At higher performance levels, the limitation of independently optimising valve opening profiles becomes more apparent [20, 31]. With a relatively basic mechanical layout and better performance, mechanical efficiency, and fuel economy than pushrod designs, SOHC engines are therefore a compromise, but they still can't match the possibilities of more sophisticated, dual-cam variants [15].

2.5.1.3 Double Overhead Camshaft (DOHC)

The double overhead camshaft system addresses many of these shortcomings by introducing a second camshaft in the cylinder head, allowing independent actuation of intake and exhaust valves [16, 33]. This design provides several mechanical and thermodynamic advantages, particularly when configured with four valves per cylinder, a layout that significantly enhances airflow and supports efficient high-RPM applications [5], including the Peugeot Grand Prix car from 1912, which featured a four-cylinder DOHC engine with four valves per cylinder. This pioneering design demonstrated a marked improvement in breathing capability and engine output, setting the stage for the adoption of similar architectures in motorsport and eventually in production vehicles

[34].

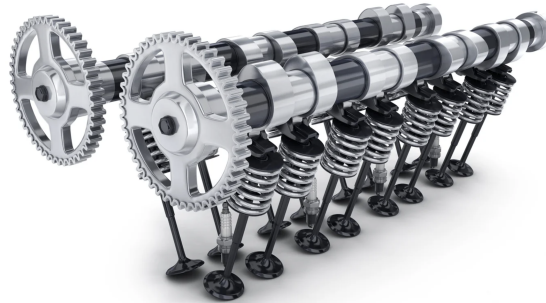


Figure 2.6: Double Overhead Camshaft System [35].

With separate camshafts controlling intake and exhaust valves, engineers gain a greater degree of freedom in valve timing and duration. This precision allows for better matching of valve operation to the combustion characteristics required at different engine loads and speeds, even when constrained to a fixed timing profile [33]. Moreover, DOHC heads are inherently better suited for the implementation of variable valve timing and lift technologies, as each camshaft can be actuated independently through phasers or other control mechanisms. The use of DOHC configurations has become ubiquitous in modern multi-valve engines, particularly those used in performance vehicles. Notable examples include Toyota's 4A-GE engines and BMW's M-series inline-sixes and V8s, which achieve renown for their high specific output and linear power delivery [36, 37].

DOHC systems, however, are not without disadvantages. They are more complex to manufacture, heavier, and often taller than their SOHC or pushrod counterparts, factors that can affect both production cost and vehicle design constraints [17]. The increased number of components in the valvetrain - including two camshafts, extra bearings, timing chains or belts, and sometimes additional tensioners - also introduces potential failure points and maintenance requirements [15]. Still, these trade-offs are frequently justified by the performance and efficiency gains that DOHC designs offer, particularly in high-compression, high-speed and forced-induction applications [5, 20].

In the evolution of fixed valve timing technologies, the progression from pushrod to SOHC and finally to DOHC systems reflects a continuous effort to enhance combustion efficiency, volumetric performance, and operational flexibility within the limits of mechanical control. Each of these architectures introduced new levels of sophistication in valve actuation, paving the way for today's electronically controlled, variable valvetrain systems. Despite the increasing prevalence of variable valve timing, the legacy of fixed timing remains deeply embedded in engine development history, and these systems continue to serve as the basis for understanding more advanced

mechanisms in contemporary high-performance and conventional powertrains [5, 16].

The motor in the study has a Double Overhead Camshaft geometry.

2.5.2 Variable Valve Systems (VVx)

As previously mentioned, Variable Valve Systems (VVS) have significantly advanced internal combustion engine (ICE) technology, which has allowed a maximisation of performance across various operating conditions. Initially, simpler mechanisms such as cam-phasing were introduced, modifying the relative position of the camshaft concerning the crankshaft, thus altering valve timing events without changing valve lift or duration. Although beneficial, these early systems were limited in flexibility and posed challenges related to complexity, cost, and restricted operational effectiveness [20, 38].

To address these early limitations, advanced VVS technologies have been developed. Honda's Variable Valve Timing and Lift Electronic Control (VTEC) system (Figure 2.7) represents a significant innovation, incorporating multiple camshaft profiles that enable discrete shifts between low-lift, economical operation at lower speeds, and high-lift performance-oriented settings at higher RPM ranges through the use of hydraulic actuators [33]. Additionally, Fiat introduced the MultiAir technology, employing an electro-hydraulic actuation system to facilitate fully variable and precise cylinder-specific control of intake valve events. This approach enhances combustion efficiency and substantially reduces exhaust emissions, demonstrating considerable improvements over previous systems [24]. The progression towards camless technologies, exemplified by electromagnetic valve actuation systems, offers even more extensive flexibility by eliminating mechanical cams, thereby allowing complete and independent control over each valve's timing, lift, and duration for continuous optimisation [39].



Figure 2.7: Honda VTEC System [28].

The development of camless engine technologies continues to gain momentum as research advances into electromagnetic and electro-hydraulic valve mechanisms. Studies by some investigators illustrate how these camless systems achieve superior operational flexibility and enable engines to function optimally at all speed and load conditions, providing measurable improvements in efficiency and emissions compared to traditional cam-based solutions [40]. Furthermore, investigations into composite electromagnetic valve mechanisms have indicated promising potential due to their rapid response, precision control, and high reliability, underscoring their suitability for future engine designs aimed at stringent efficiency and environmental regulations [41].

Extensive research has also been dedicated to understanding the combustion implications of variable valve timing. [41] demonstrated that precise manipulation of valve timing can notably enhance combustion stability, reduce pollutant emissions, and optimise fuel consumption in spark-ignition engines, reinforcing the significant role of advanced valve timing in meeting contemporary emission standards and efficiency goals [42]. Complementarily, it has been established that precise adjustments in the intake valve opening angle significantly influence engine performance and efficiency, indicating that minor alterations in valve timing parameters could effectively enhance engine responsiveness and lower emissions [43].

Finally, current research continues to investigate the comprehensive effects of VVS on engine thermodynamic efficiency, particularly in relation to pumping losses and volumetric efficiency. [43] provided analytical insights highlighting how variable valve timing significantly reduces pumping losses and increases the net indicated work of engines, enhancing overall thermal efficiency [44]. In a similar vein, this investigation into the breathing characteristics of spark-ignition engines underscored how precisely controlled valve events substantially improve volumetric efficiency, thereby optimising power output and fuel economy. These ongoing advancements emphasise the critical role of VVS technologies in future high-performance, environmentally sustainable internal combustion engine designs.

2.5.2.1 Variable Timing Systems (VVT)

Variable Valve Timing (VVT) systems are designed to dynamically adjust the timing of valve opening and closing events in internal combustion engines. This variability allows the engine to respond efficiently across a wide range of operating conditions. Early VVT systems primarily involved simple cam phasing mechanisms, which could adjust the relative angle between the camshaft and crankshaft, thus changing the timing of intake or exhaust valve events. Such systems improved efficiency moderately, but were constrained by limited variability and a relatively narrow adjustment range.

With the evolution of technology, more sophisticated and capable VVT systems have emerged. BMW's VANOS system (Figure 2.8), for example, introduced continuous camshaft adjustment,

allowing smoother, infinitely variable timing within specific operational limits. VANOS and similar systems could better optimise performance across various engine speeds and loads, leading to significant improvements in combustion efficiency and emissions reduction. Toyota's Variable Valve Timing with intelligence (VVT-i) is another prominent example that significantly improved engine responsiveness and reduced fuel consumption, particularly in urban driving conditions, where low-speed torque and transient response are critical [20].

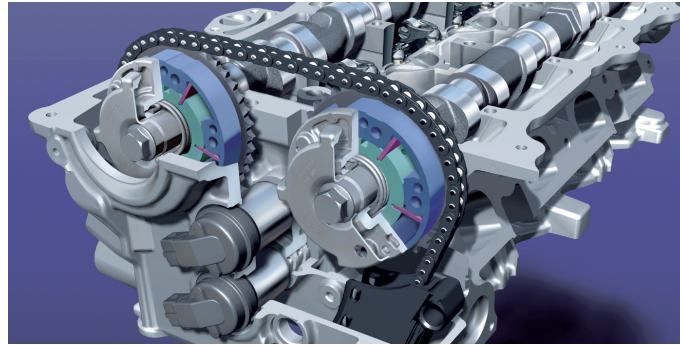


Figure 2.8: BMW VANOS System [45].

Recent advancements in VVT systems have introduced dual independent camshaft phasing (D-VVT), enabling separate timing adjustments for intake and exhaust valves. This innovation provides greater control over the valve overlap period, optimising gas exchange processes and further enhancing combustion efficiency. For example, Ford's Twin Independent Variable Camshaft Timing (Ti-VCT) system independently adjusts both intake and exhaust camshafts, achieving more effective emissions reduction and fuel economy gains [46].

Research studies have consistently demonstrated the benefits of precise valve timing adjustments on engine performance. [41] evaluated the influence of variable valve timing on combustion, showing clear improvements in stability and emission reductions, particularly for hydrocarbons and carbon monoxide, through optimised valve overlap and timing. Similarly, [43] emphasised the importance of valve timing adjustments to reduce pumping losses, thus directly improving thermal efficiency and fuel economy in modern engines.

The ongoing development in VVT technology continues to focus on enhancing electronic control systems for faster response and higher precision. Future advancements are anticipated to further integrate VVT with complementary technologies, such as turbocharging and direct injection, to enable even greater improvements in performance, emissions, and fuel efficiency. Researchers predict continued refinement and wider adoption of dual independent and continuously variable valve timing systems as fundamental components of advanced internal combustion engines [47].

2.5.2.2 Variable Lift Systems (VVL)

Variable Valve Lift (VVL) systems enable the adjustment of valve opening height, dynamically controlling airflow into and out of the combustion chamber. This capability provides significant improvements in volumetric efficiency, engine torque, and overall combustion performance. Early VVL systems, such as Honda's VTEC, introduced discrete steps in valve lift, typically employing separate cam profiles that switched based on engine speed and load. These systems achieved considerable improvements but were still limited to distinct, pre-set lift profiles [33].

Subsequent developments have introduced continuously variable lift mechanisms, such as BMW's Valvetronic system (Figure 2.9), which employs a shaft-and-rocker-arm arrangement to precisely adjust intake valve lift and duration continuously. This technology enabled throttle-less engine operation, dramatically reducing pumping losses and significantly improving fuel economy and emissions control compared to conventional throttled engines [24]. Nissan's Variable Event and Lift (VEL) system also provides continuously variable lift, achieving similar efficiency gains and smoother engine performance through precise electronic control [48].

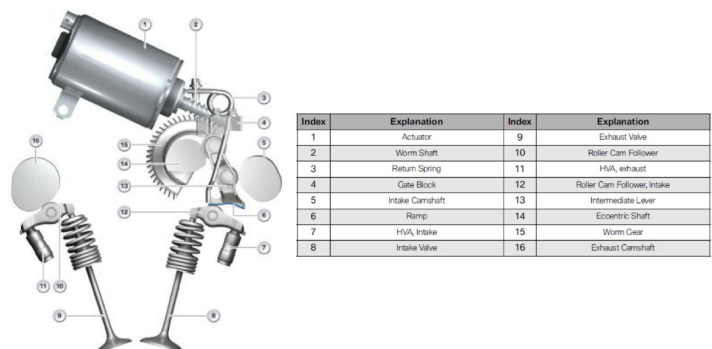


Figure 2.9: BMW Valvetronic System [49].

Research by [33] demonstrated the advantages of continuously variable valve lift in optimising engine breathing characteristics, showing notable improvements in fuel efficiency and torque, particularly in partial load conditions typical of everyday driving scenarios. Their study highlighted the capability of a continuously adjustable lift to reduce pumping losses and improve combustion quality.

Further innovations continue to emerge in VVL technology, driven by increasing demands for fuel economy and stringent emission standards. Hyundai's Continuous Variable Valve Lift (CVVL) system, for example, integrates mechanical and electronic controls to provide variable lift and duration, improving engine response, emissions, and efficiency over a wide operating range [50]. Additionally, Fiat's MultiAir technology has set new standards by implementing electro-hydraulic variable lift control, providing exceptionally precise cylinder-by-cylinder valve actua-

tion [24].

Future research in VVL technologies aims to enhance reliability, response time, and control precision, thereby supporting their integration into hybridised and advanced combustion engine platforms. Continued developments in electronic actuation systems and mechanical reliability will further solidify VVL as a cornerstone technology in advanced ICE design.

2.5.2.3 Variable Duration Systems (VVD)

Variable Valve Duration (VVD) systems are specifically designed to control the duration of the engine valve's opening during each combustion cycle. Precise duration control can optimise combustion conditions, influencing engine efficiency, emissions, and overall power output. While earlier systems indirectly varied valve duration through variable lift and timing, dedicated variable duration technologies offer more direct control over the combustion process [42].

Advanced VVD systems have become prominent due to their unique ability to tailor valve-open periods independently. Hyundai's CVVD (Continuously Variable Valve Duration) system, for instance, achieves dynamic duration changes via a rotating camshaft mechanism, enabling more efficient and cleaner combustion by matching valve duration precisely to engine speed and load conditions [50]. This system provides substantial improvements in both performance and fuel efficiency, particularly under varied and real-world driving scenarios.



Figure 2.10: Hyundai CVVD [51].

Some researchers highlighted the effectiveness of variable valve duration systems in managing engine performance by precisely controlling intake valve open durations. Their findings showed that adjusting valve duration significantly influences engine breathing and load control while achieving efficiency improvements.

2.5.2.4 Camless Valve Actuation (Advanced Systems)

Camless valve actuation systems signify a transformative shift in internal combustion engine (ICE) technology. By employing electronic, hydraulic, or pneumatic actuators to control valve events,

these systems offer unparalleled flexibility in valve timing, lift, and duration. This flexibility enables real-time optimisation of engine performance, fuel efficiency, and emissions across a broad range of operating conditions [52].

One notable implementation of camless technology is Koenigsegg's FreeValve system. This innovative approach utilises electro-pneumatic-hydraulic actuators to achieve independent control of each valve, allowing for advanced strategies such as cylinder deactivation, variable valve timing, and multi-lift profiles. The FreeValve system has demonstrated significant improvements in engine responsiveness and efficiency, particularly in high-performance applications [26].

Research into electromagnetic valve actuators (EMVAs) (such as the one present on Figure) has further advanced the potential of camless systems. EMVAs offer rapid response times and precise control over valve movements, contributing to enhanced combustion stability and reduced fuel consumption. Studies have explored various EMVA designs, including hybrid configurations that combine permanent magnets with electromagnetic coils to optimise performance and energy efficiency [53].

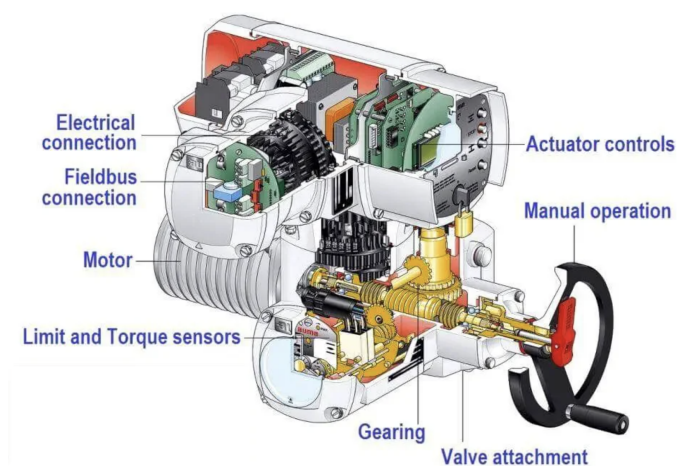


Figure 2.11: Electromagnetic Valve Actuator [54].

The adoption of camless valve actuation also presents opportunities for improved engine reliability and reduced maintenance. By eliminating components such as camshafts, timing belts, and associated hardware, the system reduces mechanical complexity and wear points. This simplification can lead to longer service intervals and lower maintenance costs over the engine's lifespan [55].

Future developments in camless technology focus on enhancing actuator durability, minimising energy consumption, and integrating intelligent control algorithms. These advancements aim to further refine engine performance and support the broader goals of automotive efficiency and sustainability. As research progresses, camless valve actuation is poised to play a pivotal role in

the evolution of high-performance ICEs [52, 53].

Practical positioning: camless vs desmodromic on a production-derived head

Camless actuation (electro-pneumatic, electro-hydraulic, electromagnetic) offers full authority over timing, lift and duration, enabling per-valve strategies and rapid transient shaping [10, 11, 56, 57]. On a compact DOHC, production-derived head, this flexibility brings substantial integration tasks: actuator packaging and local cooling, non-trivial power draw at high cycle rates, fast closed-loop motion control with robust default-to-close behaviour, and durable sensing/diagnostics for series use [11, 56, 57].

By contrast, a mechanical desmodromic layout is a near-term route for the 9A1 architecture: it removes coil springs and their surge ceiling, enforces positive mechanical closure, reuses the existing timing drive and oil system, and fits within a machined carrier. High-speed viability is supported in the literature when contact, lubrication and shaft-line stiffness are engineered appropriately [12, 58].

Table 2.1: Desmodromic vs. camless on a 9A1-type head: near-term retrofit vs. longer-horizon redesign (thesis context).

Criterion	Desmodromic (this work)	Camless (advanced)
Primary benefit	Positive mechanical closure; removes spring surge; reduced spring work	Full authority of timing/lift/duration; per-valve strategies [10, 11, 57]
Packaging in 9A1 envelope	Carrier within existing head; reuses timing/oil circuits	Actuator volume, wiring, power electronics and local cooling to integrate [56, 57]
Energy/thermal	Lower spring work; contact losses managed by tribology	Actuator power and heat rejection at high engine speeds [10, 11]
Controls/safety	Conventional cam phasing; mechanical default-to-close	Fast closed-loop motion control; robust default-to-close strategy [11, 56]
High-rpm relevance	Directly addresses seating control/float (current limiter)	Flexibility high; production-viable > 10,000 rpm on compact heads remains challenging [10, 11]
Maturity/risk	Near-term retrofit with classical validation (contacts, stiffness, oil) [12, 58]	Prototype maturity improving; series-integration risks (durability, cost, service) remain [56, 57]

Implication for this thesis Given the 9A1 packaging and the specific objective (credible 10 500 rpm operation), the desmodromic route is positioned as a short- to medium-term, production-compatible evolution; camless systems remain a longer-horizon redesign with higher integration and validation burden on this architecture [11, 12, 56, 58].

2.6 Mechanical Components in Conventional Valvetrain Systems

In internal combustion engines, the valvetrain is responsible for controlling the timing and operation of the intake and exhaust valves. Traditional valvetrain systems comprise several mechanical components that transmit motion from the camshaft to the valves. Understanding these components is crucial as it provides a baseline for evaluating potential improvements in performance.

2.6.1 Lifters/ Tappets

Lifters, also known as tappets, are components that follow the camshaft lobes and transmit their motion to the subsequent valvetrain elements. There are primarily two types of lifters: solid (mechanical) and hydraulic. Solid lifters are rigid components that require periodic adjustment to maintain proper valve clearance. Hydraulic lifters, on the other hand, use engine oil pressure to automatically adjust for thermal expansion and wear, maintaining zero valve clearance and reducing maintenance requirements.

Roller lifters (Figure 2.12) incorporate a rolling element at the interface with the camshaft lobe, reducing friction and wear. This design is particularly beneficial in high-performance applications where reducing valvetrain inertia and friction is crucial [59].



Figure 2.12: Roller Lifters [60].

The choice between hydraulic and solid lifters depends on the specific engine application. High-performance engines often prefer solid lifters due to their ability to handle higher RPMs without the risk of "pump-up" issues associated with hydraulic lifters. However, hydraulic lifters offer quieter operation and reduced maintenance, making them suitable for everyday vehicles [61].

Proper maintenance of lifters is essential to ensure optimal engine performance. Worn or malfunctioning lifters can lead to valve train noise, reduced engine efficiency, and potential damage to other components. Regular inspection and timely replacement are recommended, especially in

high-mileage engines.

2.6.2 Pushrods

In overhead valve (OHV) engines, where the camshaft is located within the engine block, pushrods are used to transmit motion from the lifters to the rocker arms situated in the cylinder head. Pushrods are slender rods that must be precisely manufactured to ensure proper valvetrain geometry and timing. Some designs incorporate hollow pushrods to facilitate the flow of engine oil to the rocker arms, aiding in lubrication and cooling.

The stiffness and mass of pushrods are critical factors influencing valvetrain dynamics. In high-speed engines, pushrod deflection can lead to timing inaccuracies and reduced performance. Therefore, materials with high-strength-to-weight ratios are preferred to minimise these effects.

Advancements in material science have led to the development of pushrods made from chromoly steel and other high-strength alloys. These materials offer improved rigidity and reduced weight, contributing to enhanced engine performance and durability.

Proper pushrod length is crucial for maintaining correct valvetrain geometry. Incorrect pushrod length can result in improper valve timing, reduced engine efficiency, and increased wear on valvetrain components. Engine builders often use adjustable pushrods during the assembly process to determine the optimal length before final installation.

2.6.3 Rocker Arms

Rocker arms act as levers that transfer motion from the pushrods (or directly from the camshaft in overhead camshaft engines) to the valves. They pivot on a shaft or a stud, converting the upward motion of the pushrod into the downward motion required to open the valve. The design of rocker arms, including their ratio and mass, significantly affects valve lift and engine performance.

Materials used for rocker arms range from stamped steel in standard applications to aluminium and forged steel in high-performance engines. The incorporation of roller tips at the valve contact point reduces friction and wear, enhancing durability and efficiency.

The rocker arm ratio, defined as the ratio of valve lift to camshaft lobe lift, plays a significant role in determining the engine's breathing characteristics. Higher ratios can increase valve lift, improving airflow and potentially enhancing engine performance. However, higher ratios also increase the stress on valvetrain components, necessitating the use of stronger materials and precise

engineering.

Proper alignment and geometry of rocker arms are essential to ensure efficient valve operation and minimise wear. Misaligned rocker arms can lead to uneven valve lift, increased friction, and premature failure of valvetrain components. Engine builders must carefully set up and verify rocker arm geometry during assembly to achieve optimal performance.

2.6.4 Finger Followers

Finger followers are commonly used in overhead camshaft (OHC) and double overhead camshaft (DOHC) engines. They are pivoting levers that directly transmit camshaft motion to the valves. This design reduces the number of components and the overall mass of the valvetrain, allowing for higher engine speeds and improved performance.

Roller finger followers, which include a rolling element at the camshaft interface, further reduce friction and wear. The reduced mass and friction in finger follower systems contribute to more precise valve timing and increased engine efficiency.

The design of finger followers must account for factors such as contact stress, lubrication, and material selection to ensure durability and performance. Advanced manufacturing techniques, such as precision forging and surface treatments, are employed to produce high-quality finger followers capable of withstanding the demands of high-performance engines.

Incorporating finger followers into engine designs can also facilitate the implementation of variable valve timing and lift systems. By adjusting the pivot point or using hydraulic mechanisms, engineers can achieve variable valve actuation, enhancing engine efficiency and performance across a broader operating range.

2.6.5 Valve Springs

Valve springs are responsible for closing the valves after they have been opened by the camshaft-induced motion. They must provide sufficient force to ensure the valve returns to its seat promptly and remains closed during the compression and combustion strokes. The design of valve springs, including their stiffness and damping characteristics, is crucial to prevent valve float and ensure reliable engine operation at high speeds.

In high-performance engines, dual or beehive valve springs are often used to provide the necessary force while minimising mass and reducing the risk of resonance-induced failures. However, valve springs also contribute to parasitic losses due to the energy required to compress them during

valve opening, which is a consideration in the development of springless valve actuation systems.

Advancements in valve spring materials, such as the use of chrome-silicon and other high-strength alloys, have improved the fatigue life and performance of valve springs. Additionally, surface treatments like shot peening and nitriding enhance the durability and resistance to stress-induced failures.

Proper installation and maintenance of valve springs are essential to ensure optimal engine performance. Factors such as installed height, coil bind clearance, and spring pressure must be carefully measured and verified during assembly. Regular inspection and replacement of valve springs are recommended, especially in high-performance or high-mileage engines.

2.7 Desmodromic Valvetrain

This work will focus on the Desmodromic Valvetrain, a system mainly developed by Ducatti that does not depend on coil springs to return the valve to its closed position.

2.7.1 Definition and Working Principle

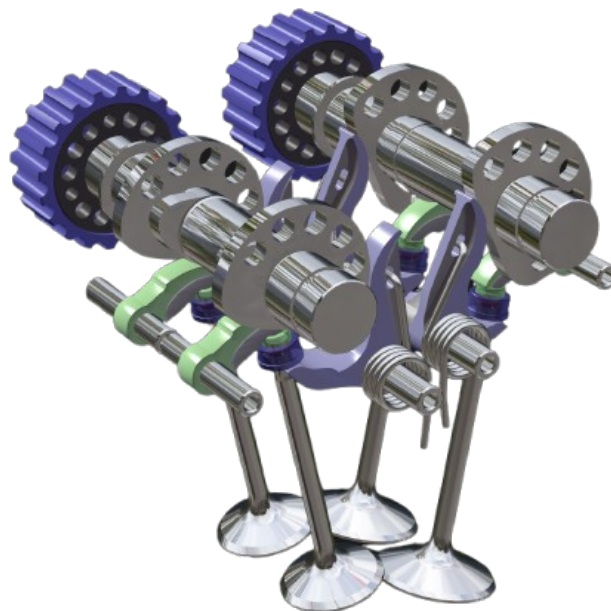


Figure 2.13: Desmosystem [vielzutun2025].

Each conjugate cam disc contacts a rocker, and both rockers interact with a backlash adjuster at the valve tip. The cam discs exert force on the adjuster through the rockers in opposite directions: one accelerates the valve in the opening direction (positive cam disc/ rocker), while the other manages closure (negative cam disc/ rocker). Though commonly referred to as "opening" and "closing" cam discs/ rockers, these terms are technically imprecise.

A preloaded helical spring is installed around the negative rocker pin, ensuring proper contact between the valve head and seat during the dwell phase when the cam discs momentarily separate from the rockers. Unlike conventional valve trains with closing springs, desmodromic systems enable higher valve accelerations without requiring excessively stiff springs, preventing follower lift-off from the cam. However, the mechanical complexity of desmodromic systems is justified primarily in high-speed, single-cylinder-head engines, such as those used in Ducati racing motorcycles.

2.7.2 Historical Development and Key Innovations

The origins of Desmodromic valve control date back to early twentieth-century mechanical engineering experimentation, though the concept only gained practical prominence during the mid-1950s through Ducati's application in high-performance racing motorcycles. The term Desmodromic derives from the Greek *desmos* (controlled) and *dromos* (course), highlighting the fundamental characteristic of the system: the positive mechanical control of valve motion in both opening and closing phases, eliminating the need for return springs.

Early theoretical patents and mechanical sketches proposed springless valve mechanisms as a means to reduce mechanical losses and enhance reliability. These concepts were largely experimental, but they laid the groundwork for more refined implementations. By the early twentieth century, it was understood that traditional valve springs exhibited limitations at high engine speeds due to harmonic oscillations, resulting in valve float — a condition in which the valve fails to close in synchrony with the camshaft, potentially causing power losses and mechanical damage. To address this, some aviation engines, particularly in radial configurations, trialled mechanically closed valvetrains to improve reliability under high-RPM and high-altitude conditions. However, their complexity hindered widespread adoption ([62]). One of the early applications of this desmodromic valvetrain was in 1912 in the Peugeot L76 ([63]).

A significant breakthrough occurred in Formula One during the 1950s. Mercedes-Benz developed a Desmodromic system for the 1954 W196 and the 1955 300 SLR race cars, aiming to eliminate valve float at operating speeds nearing 9000 RPM. This innovation provided a clear advantage in performance and durability, contributing to the team's dominance during the 1954–1955 seasons under drivers such as Juan Manuel Fangio and Stirling Moss ([64]). Concurrently, in 1956,

Fabio Taglioni at Ducati introduced a Desmodromic system for motorcycles, marking the beginning of its extensive use in the company's racing and production vehicles. By the 1970s, the technology had become a hallmark of Ducati's engineering philosophy, providing superior valve control at high RPMs and improving both reliability and engine response. To this day, models such as the Ducati Panigale V4R exemplify the continued relevance of the Desmodromic system in motorsport and commercial superbikes ([65]).

The principal innovation of the Desmodromic architecture lies in its elimination of valve springs — components which, although essential in conventional designs, introduce significant limitations in high-performance applications. Valve springs are prone to dynamic surge and fatigue, particularly under high cyclic loading. These effects can lead to delayed valve closure, diminished combustion efficiency, and ultimately mechanical failure. Desmodromic systems bypass these issues through cam-driven closure, allowing precise valve actuation at engine speeds exceeding 15,000 RPM [64]. This configuration also enables the use of sharper cam profiles, improving the rate of valve lift and enhancing the tuning of air-fuel exchange processes. The result is improved engine breathing, throttle response, and thermal efficiency. Additionally, by removing the energy expenditure associated with compressing valve springs, Desmodromic systems offer a reduction in parasitic losses, contributing to better fuel economy and engine responsiveness [66].

Despite its technical advantages, the widespread application of Desmodromic valve control remains limited. Ducati remains the sole mass-production manufacturer to adopt the system, using it as a technological differentiator in both MotoGP and consumer markets. However, the inherent complexity, manufacturing cost, and maintenance requirements have restricted broader adoption, particularly in the automotive sector. Furthermore, the emergence of camless valve actuation systems — such as electromagnetic [67] and electrohydraulic architectures [68] — presents viable alternatives that offer similar benefits with improved flexibility and programmability. These technologies are capable of variable timing, lift, and duration, without the mechanical intricacy of Desmodromic linkages.

Nevertheless, the core principles of Desmodromic control may find renewed relevance within the evolution of camless engines. By combining the deterministic nature of mechanical closure with the flexibility of electronically controlled actuators, future systems may achieve optimal valve control across a wide operating range. As shown by recent studies into electromagnetic actuators [67], these systems face their challenges, including high power requirements and thermal management, which echo the trade-offs long associated with Desmodromic systems.

In conclusion, although the Desmodromic valvetrain has seen limited adoption beyond Ducati and historic Formula One applications, its influence on high-performance engine design remains significant. It offers a compelling case study in mechanical innovation, addressing fundamental limitations of spring-based actuation through positive control mechanisms. As research continues

into next-generation combustion engines and adaptive valve technologies, the legacy of Desmodromic systems is likely to inform future solutions, particularly in contexts where precision, reliability, and high-RPM performance are paramount.

2.7.3 Mechanical Design and Components

Mechanically, a desmodromic system is more complex than a traditional spring valvetrain. The primary elements in a Ducati-style desmodromic valvetrain are the camshaft (with paired lobes for each valve), an opening rocker, a closing rocker, and the valve itself with a split collar or tappet that the closing rocker engages. The camshaft is driven by belt or chain (belts in many earlier Ducati engines, chains or gear trains in later ones) much like a DOHC drive. In addition to the usual opening lobe, the cam must carry a precise closing lobe. The opening lobe resembles a conventional profile but is typically optimised for rapid lift and a clean hand-over to closing control. The closing lobe is shaped to “catch” the closing rocker and bring the valve back to its seat at a controlled velocity. Correct valve seating without bounce is critical: if the valve is slammed shut under high cam force it may bounce or accelerate wear. Accordingly, desmodromic cams use a carefully engineered closing ramp to seat the valve gently.

The rockers are small, hardened steel levers. The closing rocker commonly carries a screw or shim adjuster to set running clearance to the valve (a small gap must exist when the valve is fully closed to accommodate thermal expansion). Clearances in desmodromic systems are typically tighter on one side than in comparable spring valvetrains and require periodic adjustment by trained technicians; historically this has been viewed as a maintenance downside. Modern Ducati designs adopt shim-under-rocker solutions to stabilise clearance at high rpm and recommended service intervals have increased (for example, to around 24 000 km on recent models). Even so, precise assembly and regular checking of valve clearances remain characteristic of desmodromic engines.

From a materials standpoint, rockers are usually surface-treated (nitrided or DLC-coated) to cope with stress, and cams are hardened steel. Some engines employ finger followers for opening (in racing applications) with a separate claw for closing. To minimise friction, valve-stem sealing is often lighter than in spring engines, which can lead to slightly higher oil consumption. Overall, the valvetrain is more involved: each valve uses two rocker shafts and two rockers in addition to the cam lobes, plus small return springs on the rockers (hairsprings or light coils that maintain contact when clearance appears). Despite the extra parts, packaging is compact: Ducati V-twin heads, for example, accommodate two desmodromic valves per cylinder with little size penalty, and V4 heads package four desmodromic valves per cylinder with a more complex rocker layout. Positive closure introduces more friction interfaces (two cam contacts instead of one), which is mitigated by low-friction coatings and careful oil control.

From a design-energy perspective, opening a valve in a desmodromic engine requires less work than in a spring engine because there is no stiff spring to compress—only the acceleration of the valve and rocker mass. Closing is not “free”: the closing cam must decelerate the valve and force it shut, whereas a spring engine returns stored energy. In practice, the overall work may be similar but distributed differently. Several analyses suggest a small net reduction in valvetrain friction with desmodromic closure because the continuous spring compression losses are removed; this benefit is most evident at lower engine speeds where a stiff spring would otherwise waste energy.

2.7.4 Advantages and disadvantages

The principal advantage is the prevention of valve float at high engine speed. Because the valve is actively closed by the cam, it follows the intended profile even at extreme rpm where a spring system may float or bounce. This permits reliably higher rev limits. Modern desmodromic V4 street engines operate in the 14 000–16 500 rpm band, a level that would be challenging for large-valve, high-lift spring systems. In racing, desmodromic engines sustained very high rpm long before pneumatic springs became commonplace, which provided a decisive reliability advantage.

A second benefit is the freedom to employ faster opening and closing profiles. Without the constraint of spring dynamics, the cam can be made “sharper”—valves open quicker and can remain fully open longer for a given duration to improve flow. High opening and closing accelerations increase the area under the lift curve, improving cylinder filling even at modest speeds. In effect, the designer can open and close the valve more aggressively without risking float, which raises breathing efficiency.

A further benefit is the reduction of spring-related losses. Conventional engines spend appreciable work compressing valve springs; some is returned elastically, but a significant portion is lost to hysteresis and friction. With no large springs to compress, the cam drive sees less resistance during opening, reducing camshaft drive power and friction—particularly at low to mid rpm where a stiff spring would be unnecessarily compressed. The absence of large springs also removes spring fatigue or sag as a failure mode (while other components still require wear management).

Finally, the achievable rpm ceiling is often set by factors other than the valvetrain (ignition system capability, bottom-end stress), because positive closure removes float as the limiting mechanism. Historically this enabled Ducati race engines to run higher speeds than spring-valve rivals; in modern MotoGP, desmodromic systems continue to perform on par with pneumatic-spring solutions.

The desmodromic layout is undeniably more complex. Each valve requires roughly twice as many cam lobes and additional hardware, which increases manufacturing and assembly cost. Camshafts must be ground with two matched profiles per valve in precise phase, and rocker mechanisms must be built to tight tolerances from high-grade materials. The system also imposes a

higher set-up burden in production and service, since accurate clearance setting on both opening and closing paths is essential for reliability and noise control.

2.7.5 Applications of the desmodromic system

Desmodromic valvetrains have played a crucial role in motorsport, particularly in high-performance motorcycles. Ducati has utilised desmodromic actuation in MotoGP and World Superbike Championship motorcycles. The system ensures precise valve control at extremely high RPMs, eliminating valve float and enhancing power output. This level of precision is essential in racing, where every millisecond counts in optimising engine performance and durability [69].

Beyond motorcycles, desmodromic systems have also been explored in automotive and experimental engine applications. While the technology offers significant performance advantages, its adoption in cars has been limited due to its complexity and high production costs. Some high-performance and research-focused engines have tested desmodromic actuation, but conventional camshaft-driven systems remain the standard in the automotive industry. The balance between efficiency, reliability, and manufacturing feasibility has made widespread implementation challenging [70].

Looking ahead, advancements in material science and computational engineering could expand the potential of desmodromic systems. High-performance alloys, composite materials, and advanced coatings are being used to reduce downtime, improve safety, and extend valve lifespans [71]. Additionally, integrating these systems with modern fuel efficiency and emissions-reducing technologies could make them more attractive for next-generation internal combustion engines, particularly in specialised motor sport and performance sectors.

Comparing desmodromic systems with conventional valve actuation highlights key trade-offs in efficiency, durability, and complexity. While desmodromic systems excel in high-RPM scenarios, electronic valve actuation technologies are emerging as a promising alternative. These electronic systems provide even greater control over valve timing and lift, improving efficiency and emissions. Future trends may focus on hybrid desmodromic-electronic solutions, leveraging the benefits of both approaches while addressing cost and complexity challenges. The integration of lightweight materials and new actuation methods could shape the next evolution of high-performance engine design [72].

2.8 Porsche GT3 RS

The Porsche 911 GT3 RS (992 generation) represents the pinnacle of Porsche's motorsport-derived engineering in a road-legal platform. As its core lies in a naturally aspirated 4.0-litre flat-six engine, delivering 525 PS (386 kW) at 8500 RPM and 465 N·m of torque at 6300 RPM. This high-

revving unit, shared in concept with the 911 GT3 Cup race car, features individual throttle bodies and a rigid valve drive system, enabling an exhilarating 9000 RPM redline and immediate throttle response [73]. Paired with a 7-speed dual-clutch PDK transmission with shorter gear ratios than the standard GT3, the GT3 RS accelerates from 0 to 100 km/h in just 3.1 seconds and reaches a top speed of 296 km/h. The powertrain tuning prioritises track performance and linear response, making it a standout among naturally aspirated production engines [74].



Figure 2.14: Porsche GT3 RS [74].

Aerodynamic development plays a central role in the GT3 RS's performance profile. The car produces 409 kg of downforce at 200 km/h and up to 860 kg at 285 km/h, more than doubling the figures achieved by its predecessor, the 991.2 GT3 RS [75]. Key to this achievement is the hydraulically actuated Drag Reduction System (DRS), adapted from Formula 1 technology, integrated into the two-part rear wing. Additionally, numerous active and passive aerodynamic elements - such as air deflectors, underbody channels, and a single central radiator replacing the standard GT3's tri-radiator layout - contribute to enhanced cooling and aerodynamic balance under high-load conditions. These systems work cohesively to provide the high-speed stability and cornering grip necessary for competitive track use.

The suspension architecture is equally sophisticated, reflecting Porsche's experience in motorsport. The GT3 RS adopts a double-wishbone front axle with motorsport-derived geometry, optimised to reduce pitch during hard braking and ensure maximum tyre contact under lateral loads. At the rear, a multi-link suspension setup balances ride quality with high-speed control. These elements are supported by active systems such as Porsche Active Suspension Management (PASM), rear-axle steering, and adjustable anti-roll bars and dampers that can be independently tuned via steering-wheel-mounted rotary selectors—features typically found in race cars [75]. These adjustments allow drivers to tailor the handling characteristics dynamically based on circuit conditions.

To complement its powertrain and chassis, the GT3 RS incorporates extensive lightweight construction techniques. Carbon fibre-reinforced plastic (CFRP) is used in the doors, roof, bonnet, front wings, and rear wing, reducing the overall kerb weight to approximately 1,450 kg despite the inclusion of added aero and mechanical systems. Lightweight glass, minimal sound insulation, and an optional Weissach package—featuring additional magnesium components—further improve the car’s power-to-weight ratio [76]. As a result, the 911 GT3 RS achieves a Nürburgring Nordschleife lap time of 6:44.848, affirming its place among the fastest naturally aspirated production cars ever built. In sum, the 992 GT3 RS exemplifies the technical translation of track development into road-legal performance without compromise.

2.9 Measurement methodology

This section introduces the measurement tools and software that underpin all dimensions used later on. The ZEISS DuraMax CMM with the VAST XXT sensor provides reliable point and scanning measurements on machined features, while ZEISS Calypso manages alignments, features and reports. Together, they give a consistent, repeatable way to locate axes and planes, check distances and angles, and build the datum framework that drives the CAD skeletons and tolerance checks.

2.9.1 Measurement by points

This part focuses on discrete CMM probing—the day-to-day method used for tight, controlling features. Using simple patterns (planes, circles and lines) and direct software distances, the setups establish the basics: valve angles, camshaft spacing, deck heights and support diameters. Point-based work was preferred wherever precision on machined surfaces was needed, with scanning reserved for broader context and non-critical shapes.

2.9.1.1 ZEISS coordinate measuring machine

The assembly used to compose the coordinate measuring machine consisted of two components: Zeiss DuraMax and Zeiss Vast XXT.

The Zeiss DuraMax is a compact and robust coordinate measuring machine (CMM) designed for industrial metrology applications in shop-floor environments. Engineered to withstand challenging conditions such as temperature variations, vibrations, and airborne contaminants, the Zeiss DuraMax provides reliable and precise measurements without requiring a controlled laboratory setting. Its integration into manufacturing workflows enables real-time quality control, reducing downtime associated with off-site inspections and improving overall production efficiency. This

capability makes it particularly well-suited for high-precision dimensional verification in automated and inline quality control processes.

Equipped with ZEISS Vast XXT scanning technology, the Zeiss DuraMax (Figure 2.15) supports both single-point tactile probing and continuous scanning, ensuring accurate assessment of complex geometries and critical form tolerances. Such versatility is essential in the automotive sector, where dimensional precision of components like engine blocks or transmission housings is crucial to ensuring system performance and reliability. The high measurement repeatability and the ability to acquire data directly on the production line make the Zeiss DuraMax a strategic tool for maintaining quality consistency and traceability in highly demanding manufacturing environments.



Figure 2.15: ZEISS DuraMax [77].

The Zeiss Vast XXT (Figure 2.16) is a high-performance scanning sensor specifically designed for lightweight and intricate parts. Unlike traditional tactile probing systems, the XXT enables continuous scanning, capturing detailed surface data with high-speed accuracy. This makes it particularly useful for measuring intricate engine components, such as camshafts and cylinder heads, where minute deviations can significantly impact performance. The sensor supports multi-point probing and dynamic measurement, allowing for efficient inspection of complex geometries.

Building upon its core capabilities, this system also distinguishes itself through its modularity and adaptability within automated inspection systems. Its lightweight construction not only reduces dynamic loads on the measuring system, but also enhances the responsiveness and stability

of scanning routines, particularly when integrated into high-speed, high-throughput environments.



Figure 2.16: ZEISS Vast XXT [78].

An important advantage of the Vast XXT lies in its compatibility with a variety of stylus configurations, including star and angled setups, which enables access to difficult-to-reach features without the need for re-clamping or repositioning the workpiece. This is particularly advantageous in quality control workflows for complex engine assemblies, where maintaining alignment and reference systems is critical for dimensional traceability.

In addition, the sensor's ability to integrate with ZEISS software platforms allows for advanced path planning, real-time data analysis, and efficient error compensation strategies. These features support closed-loop manufacturing processes, where measurement results are used to inform corrective actions upstream, ensuring continuous optimisation of production quality.

2.9.1.2 Calypso software

Zeiss Calypso is a powerful and flexible coordinate measuring software developed by Carl Zeiss Industrial Metrology for use with their range of coordinate measuring machines (CMMs). It provides a comprehensive platform for dimensional inspection, enabling automated and highly accurate measurement of complex geometries. Calypso is distinguished by its parametric and object-oriented approach, allowing users to define features, tolerances, and alignment strategies directly from CAD models. The software supports both contact and non-contact probing systems and integrates seamlessly with Zeiss CMMs such as the DuraMax series. Its user interface is designed to facilitate both novice and expert users by streamlining routine tasks while enabling advanced

control strategies for high-precision applications.

One of the most important features of Calypso is its full compatibility with ISO GPS standards and its capability to interpret geometric dimensioning and tolerancing (GD&T) directly from embedded CAD data. It includes automated path planning, collision detection, and support for scanning strategies using sensors such as the VAST XXT or VAST Gold. Moreover, the software allows for real-time process monitoring, report generation, and integration into statistical process control systems (SPC). In high-performance sectors like automotive and aerospace, Calypso is widely recognised for enabling traceable, repeatable, and efficient dimensional verification workflows. The ability to simulate measurement strategies before execution ensures minimal machine downtime and significantly enhances throughput in production and quality control environments.

2.9.2 3D scanning

This section explains why optical 3D scanning was brought in alongside tactile probing: it captures the complex, free-form shapes of the cylinder head and donor parts quickly, giving rich surface detail that helps with packaging checks, clearance studies and CAD surfacing. The workflow described in the thesis—scan capture, alignment, mesh cleaning and export—was chosen to supply dependable geometry for reverse-engineering and visual deviation maps before committing to machining or rework.

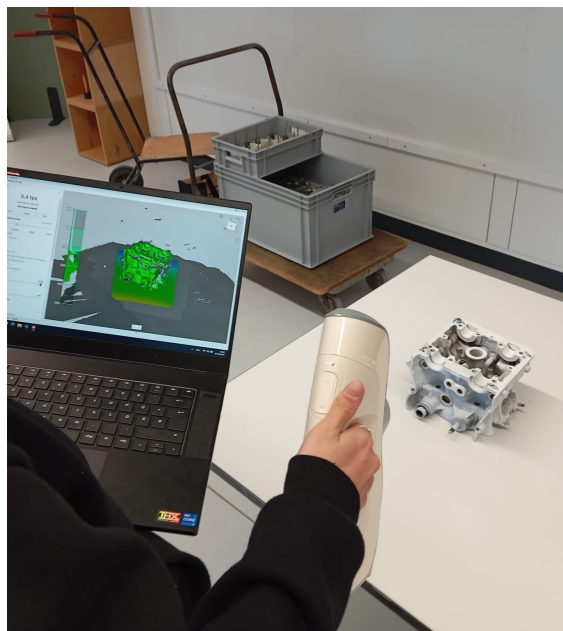


Figure 2.17: 3D Scanning performed.

2.9.2.1 Artec Eva and Artec Studio

The Artec Eva handheld structured-light 3D scanner has been extensively evaluated in peer-reviewed studies for its accuracy in capturing human anatomy and complex geometries. It was investigated for its performance against a high-precision Romer laser scanner using transtibial and transfemoral limb models, finding a mean volume error below 2 and reproducibility under 5, with cross-section deviations under 1 mm—well within clinically acceptable margins. Another clinical study confirmed its effectiveness when scanning human hands, demonstrating comparable fidelity to the more precise Artec Space Spider while maintaining practicality at larger sizes. These outcomes support Eva's utility in applications requiring anatomical precision, such as prosthetic design and orthotic fitting.

Additional research underscores its performance in craniofacial measurement tasks. Authors examining surface area and volumetric accuracy in head and neck scans reported sub-millimetric error irrespective of subject position, highlighting the Eva's repeatable and reliable data capture in applications with stringent requirements. Clinicians in plastic surgery have further validated its output quality, noting that structured-light capture at 16 fps provides high-fidelity texture and geometry maps suitable for planning and postoperative assessment. Through these studies, it is clear that the Eva excels in medical and clinical metrology environments, offering a balance between speed, accuracy, and ease of use.

The Artec Studio software that accompanies the Eva has been evaluated for its metrological capabilities in both research and applied contexts. Its Autopilot workflow streamlines data alignment and fusion, with studies highlighting its contributions to simplified yet robust processing pipelines suited for clinical workflows. Supporting reverse-engineering tasks, the software's mesh-to-CAD features—including automated surface generation and inspection tools—have been employed successfully in engineering contexts. Industry sources credit recent versions (Studio 18/19) with advancing AI-driven texture enhancement, multi-resolution data fusion, and improved performance in quality inspection tasks.

Finally, in the realm of industrial metrology and manufacturing, Artec Studio paired with Eva or the new Artec Point scanner offers a credible alternative to traditional laser systems. Trade and engineering analyses note that Studio supports deviation mapping and dimensional inspection, exporting to standard CAD formats and interfacing with tools like Geomagic Control X and ZEISS INSPECT for traceable quality assurance. The latest software releases have received formal metrology certifications (e.g., PTB), reinforcing their suitability for demanding industrial applications.

2.9.3 Metrology and tolerances

This subsection answers three practical questions. First, how accurate are our measurements? Second, what tolerances make the adaptation viable? Third, how do we ensure the solution is reproducible in manufacture?

Measurement capability. The ZEISS CMM (CALYPSO) verified to ISO 10360-2 delivers an expanded uncertainty (95 %, U_{95}) of about ± 0.008 mm for the camshaft-axis separation routine used here (bank reference, plane–axis–point datums). The optical 3D scanning workflow verified to VDI/VDE 2634-2 provides $U_{95} \approx \pm 0.14$ mm for large, best-fit distances. The CMM is therefore used for critical machined features; the scanner is used for non-critical cast surfaces and qualitative clearance maps [79–81].

Interpreting the 12.4–12.7 mm offset. The reported 12.4–12.7 mm value is a *design offset* between axes when comparing the donor desmodromic layout with the Porsche head. It is not a tolerance. What matters for assembly is how tightly we control the *new* axis locations and alignments once the carrier is machined and installed.

Critical tolerances adopted. Based on line-boring practice, bearing fits, and chain drive alignment, the following limits are used. The ± 0.03 mm target on axis separation keeps contact loads and wear within credible limits; ± 0.10 mm would be too loose for reliable high-speed running [3, 4, 6].

Table 2.2: Critical features and tolerances (verification method and simple decision rule).

Feature (KPI)	Tolerance T	Method	Decision rule
Camshaft-axis separation (per bank)	± 0.03 mm	CMM	Accept if $ m + g \leq T$, with $g = U_{95}/2$ [82].
Mutual parallelism of shaft lines	0.01 mm	CMM	Accept if $m + g \leq T$.
Shaft straightness (full span)	0.02 mm	CMM	Accept if $m + g \leq T$.
Deck-to-shaft datum distance	± 0.05 mm	CMM	Accept if $ m + g \leq T$.
Non-critical cast surfaces (clearance map)	± 0.5 mm	3D scan	Visual map; spot-check distances on CMM if clearance is marginal.

Simple uncertainty statement. Uncertainties are expressed as $U_{95} = 2u_c$ (95 %), combining Type A (repeatability) and Type B (specifications, calibration, temperature, alignment) terms per ISO GUM. For the CMM routine above we obtain $U_{95} \approx 0.008$ mm; for the scanner distances, $U_{95} \approx 0.14$ mm. These values comfortably separate instrument noise from the large geometry changes being managed [79–81].

Reproducibility in manufacture. To ensure the solution is repeatable:

- *Datums and fixturing:* use a stable A–B–C datum scheme referenced to machined faces; line-bore both cam bores in one setup.

- *In-process control*: probe datums and bore positions after roughing; finish-bore to size with compensated tool offsets.
- *Final verification*: 100 % CMM check of axis separation, parallelism and straightness against Table 2.2.
- *MSA and SPC*: short gauge R&R on the CMM routine (≤ 10 % of tolerance) and capability targets of $C_{pk} \geq 1.33$ on the shaft-separation feature in a pilot run [79, 83].

In summary, the admissible tolerance for the shaft-to-shaft distance is ± 0.03 mm (not ± 0.10 mm). With the stated measurement capability and a basic control plan (fixtures, in-process probing, CMM verification and gauge R&R), the adaptation is both buildable and reproducible for the intended high-speed operation [3, 4, 6, 82].

2.10 Conclusion

Valve actuation systems have evolved significantly to meet the demands of modern engines. While conventional camshaft-driven and variable valve timing systems dominate mass-market applications, desmodromic technology remains a critical innovation for high-performance applications. Continued research and development in valve actuation could lead to novel solutions that blend the precision of desmodromic systems with the flexibility of electronic actuation methods, paving the way for more efficient and powerful engines.

Chapter 3

Problem Description

For the integration of the desmodromic system in the Porsche GT3 RS engine, two main components were provided. The first is the cylinder head of the vehicle in question, a **9A1 Gen 5** high-performance 4.0-litre naturally aspirated flat-six unit. This engine descends from the 9A1/9A2 families. Its specifications are listed in Table 3.1.

Table 3.1: Key Specifications of the Porsche 911 GT3 RS (992) 9A1 Gen 5 Engine [porsche992gt3rs]

Specification	Value
Engine Type	4.0 L Naturally Aspirated Flat-Six (9A1 Gen 5)
Displacement	3,996 cm ³
Bore × Stroke	102.0 mm × 81.5 mm
Compression Ratio	13.3:1
Maximum Power Output	386 kW (525 PS) @ 8,500 rpm
Maximum Torque	465 Nm @ 6,300 rpm
Maximum Engine Speed	9,000 rpm
Fuel Type	Supreme Unleaded (98 RON)
Fuel Injection	Direct Fuel Injection
Lubrication System	Dry Sump with External Oil Tank
Valve Control	VarioCam (Intake and Exhaust Camshaft Adjustment)
Valvetrain	DOHC, Finger Followers
Exhaust System	Valve System, Two Catalytic Converters, Titanium Rear Silencer, Dual Titanium Tailpipes

The second one is a head of Desmo system coming from a motorbike. The details from the prototype were not shared, but the system as a whole is presented below:



Figure 3.1: Desmodromic System

The main target from this project is to increase the maximum engine speed from its current value of 9000 RPM to a new value of 10500 RPM. This target was set based on the following:

- Integrating the Desmodromic system means that **lighter** components will be integrated, and the return of the valve train will not be conditioned by the stiffness and mass of high-speed valve springs;
- The elimination of the valve springs also eliminates the high risk of valve float, making the movement more precise and less likely to suffer wear.

3.1 Implications

Integrating a springless, positive-control valvetrain within the 9A1 head has direct implications that must be verified during the validation phase.

Lubrication and oil-film integrity Critical contacts (cam–follower, rocker pivots/shafts, thrust faces) must sustain elastohydrodynamic (EHL) films across the duty at 10,500 rpm. This requires adequate supply pressure/flow, controlled inlet temperature and suitable viscosity to maintain a film ratio $\lambda \geq 1$ at the most loaded zones. Oil-jetting and gallery routing may need local tuning to address starvation risk at high speed and to aid heat removal from the carrier and covers [3, 4, 8].

Local heating Sliding zones concentrate frictional work; the resulting flash temperatures must remain below scuffing thresholds for the chosen material/coating system. Thermal paths from lobes and rocker shoes into the carrier should be sufficient to prevent thermal growth that would close end-play or distort shaft alignment [4, 8].

Vibration and dynamics Shaft-line bending and torsional ripple from phase-shifted lobe torques can excite modes in the carrier and covers. Support stiffness and mode placement require verification so that dynamic deflection does not compromise contact stability or timing accuracy. The desmodromic layout removes spring surge as the primary limiter but introduces sensitivity to axial control of rockers and to local compliance in the closing path [3, 6, 84].

Manufacturing and inspection Line-bored saddles/caps, bearing fits, and thrust-face quality must meet the tolerance budget defined for shaft parallelism, shaft separation and deck-to-shaft distance. End-play per rocker must be measurable at build and after endurance to confirm stability [8].

Controls and diagnostics (rig and dyno) Rig and engine phases should instrument contact/torque proxies, oil temperatures/pressures, and vibration at the carrier to detect loss of closure or abnormal heating early [3, 4].

3.1.1 Integration with the existing lubrication circuit

The springless valvetrain introduces additional oscillating pivots and thrust faces that must be supplied by the 9A1's dry-sump pressure stage without compromising gallery pressure at high engine speed. A simple flow mapping and a first-order flow/pressure budget help to evidence feasibility and to identify risks early.

Baseline flow path (schematic) Pressure pump → main oil gallery → branch to bottom-end (main/rod bearings and piston jets) + branch to cylinder-head galleries → cam bearing feeds and leakage-to-drain → drains to scavenge stages. Similar flow architectures and design rules (pump sizing, branch balancing, restrictors) are standard in production engines and are amenable to one-dimensional network analysis of losses and distribution [85, 86].

Added feeds for the desmodromic carrier Within the head galleries, the carrier requires (i) metered feed to rocker journal bushes, (ii) metered feed to rocker thrust faces (or flanged bushes), and (iii) optional cam-lobe wetting (controlled bleed/slinger) where warranted by contact conditions. Film formation and wear limits for cam/roller and oscillating bearings set the functional targets (e.g., $\lambda \geq 1$ at critical contacts; controlled flash temperatures) [87–89].

Flow/pressure budget (first order) The added demand is limited by orifices at each new feed to keep the incremental head-flow within the pump/regulator margin. A 1D network model- nodes at pump outlet, main gallery, head upstream, carrier inlet, and drains- with standard orifice/line losses can quantify the pressure drop and sensitivity to orifice size, oil viscosity and temperature [85, 86]. Target behaviour is that head-gallery pressure at the carrier inlet remains within the accepted band across the speed range (no regulator run-out). Where dry-sump operation is subjected to high g

loading, surge/aeration control must be checked because it affects pressure dynamics at the branch to the head [90, 91].

Contact/lubrication considerations Dynamic EHL analyses and experiments for cam–follower contacts and oscillating journals show that minimum film thickness and friction depend on load, entrainment speed and oil temperature; acceptable operation requires adequate supply and thermal control at the head [87, 89, 92, 93]. For cam/roller contacts, both measurement and modelling of film thickness and thermal effects are established [88, 94]. For oscillating bushes/thrusts, squeeze and replenishment under small amplitudes must avoid false brinelling/fretting, which is sensitive to film starvation—favouring metered feeds and smooth drains [89, 95].

Temperature management Additional sliding at thrust faces concentrates frictional heat. Head-oil temperature at the carrier inlet should be monitored and, if necessary, reduced via oil-jet optimisation or gallery routing; nozzle sizing is known to influence gallery charge and heat pick-up in engine oil circuits [96, 97]. The design objective is to keep bulk oil and flash temperatures below scuffing thresholds for the selected material/coating/finish system.

Risks and mitigations (summary)

- *Pressure shortfall at carrier inlet:* mitigate via orifice sizing (limit added head flow), verification of regulator capacity, and, if needed, a local restrictor/accumulator strategy; confirm with a 1D network model and head-gallery pressure instrumentation [85, 86].
- *Aeration/surge under high g:* verify dry-sump surge control and scavenge capacity; adapt pickup routing and baffles; confirm on track-representative manoeuvres [90, 91]
- *Film starvation at pivots/thrusts:* use metered feeds to oscillating journals and thrust faces; validate film proxies and wear after endurance [89, 95].
- *Excess thermal rise:* monitor head-inlet oil temperature; if required, adjust nozzle sizes/routing to balance cooling and pressure drop [96, 97].

Verification plan (instrumented checks) Install pressure transducers at: pump outlet, main gallery, head upstream of the carrier branch, and carrier inlet; add thermocouples at carrier inlet/outlet. Use a calibrated restrictor rig to measure incremental flow vs. orifice size and temperature. Correlate with EHL proxies on cam/roller and with post-test inspection of rocker pivots/thrusts (blueing/wear maps). This provides evidence that the added feeds do not compromise gallery pressure and that films remain acceptable across the 10,500 rpm duty [87–89].

3.2 Nomenclature

For the following process of explaining the design, a number list of the components will be followed according to the figure present below:

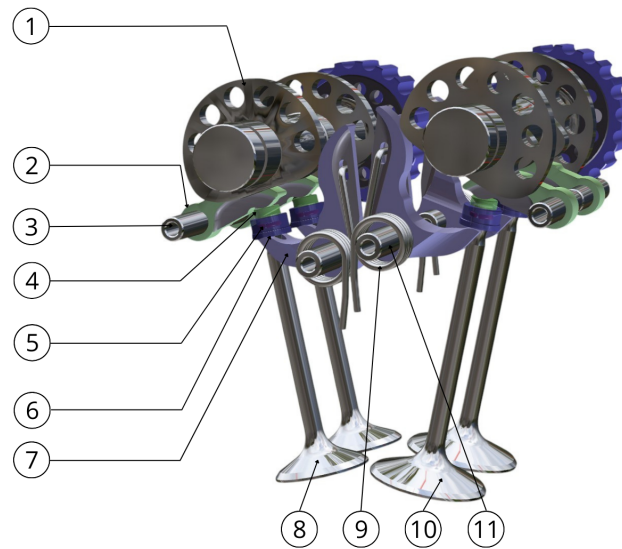


Figure 3.2: Nomenclature defined

This nomenclature represents the following elements:

Table 3.2: Component numbering used throughout the desmodromic assembly

Number	Description
1	Desmodromic Camshaft
2	Small Rocker
3	Small Rocker's Shaft
4	Valve Top
5	Valve Fixer
6	Valve Topper
7	Long Rocker
8	Exhaust Valve
9	Angular Spring
10	Intake Valve
11	Long Rocker's Shaft

An additional component, not shown in Figure 3.2, is referenced later as item 12 (spring assembly stopper).

3.3 Quantitative target justification and technology down-selection

Why 10 500 rpm? For stroke $S = 81.5$ mm, the mean piston speed is $V_p = 2Sn/60$. This gives $V_p \approx 24.5$ m/s at 9 000 rpm and $V_p \approx 28.5$ m/s at 10 500 rpm. Authoritative texts place durable road engines typically around 18–22 m/s, with short-stroke high-performance and racing engines operating acceptably in the 25–30 m/s band when component design, lubrication and cooling are addressed. The 10 500 rpm target therefore sits at the upper edge of proven practice for production-derived performance engines and is credible provided the valvetrain—the usual limiter at these speeds - is de-bottlenecked.

Table 3.3: Piston-speed context for the 9A1 stroke (why 10 500 rpm is credible).

Case	Speed (rpm)	Mean piston speed (m s^{-1})	Practice band
Baseline limit	9 000	24.5	Within performance range
Proposed target	10 500	28.5	Upper end of 25–30 band

This value lies at the upper bound of accepted practice for short-stroke performance engines. To anchor credibility, Table 3.4 lists three reference applications with bore $\times$ stroke, reference speed and mean piston speed, showing that 28.5 m/s^{-1} sits alongside demonstrated high-speed engines.

Table 3.4: Benchmarking the 10 500 rpm target using mean piston speed ($\bar{U}_p = 2Sn/60$).

Engine	Bore $\times$ Stroke (mm)	Ref. rpm	$\bar{U}_p$ (m/s)
Porsche 9A1 (target)	102×81.5	10 500	28.5
Porsche 911 GT3 Cup (992) [98]	102×81.5	8 750	23.8
Ducati Panigale V4 R [99, 100]	81.0×48.4	16 500 (6th)	26.6
F1 V6 turbo-hybrid [101, 102]	$80.0 \times \approx 53.0^*$	15 000 (limit)	26.5

* Stroke inferred from 1.6 L total displacement and fixed 80 mm bore: $S = \frac{V}{(\pi B^2/4) \times 6} \approx 53.0$ mm.

Options to lift the speed limit (comparative view) Multiple concepts could extend the safe speed window. The table below compares capability and integration on the 9A1 head. Figures and remarks reflect widely cited practice and the constraints of a road-capable, production-derived cylinder head.

Why the desmodromic route fits the 9A1 (1) It addresses the true limit at 10500 rpm: valve motion control, not the crank–rod–piston group. (2) It can be packaged within the existing head, reusing the timing drive and fluids layout, which is far more practical than PVRs or large electromagnetic actuators on a road-going head. (3) By removing spring compression work, it targets a 15–30% cut of the *valvetrain* portion of FMEP (projection), mapping to about 0.8–1.6% BSFC/ CO_2 benefit at representative points pending rig validation. (4) The residual risks (contact

Table 3.5: Alternatives to raise valvetrain speed on the 9A1 head: capability vs. integration.

Concept	High-rpm capability (typical)	Integration on 9A1 head (key points)	Verdict for this thesis
Cam + coil springs (base-line)	Up to about 9000 rpm before surge/float risk rises with aggressive profiles [3, 6]	Mature, compact and low cost; to push higher needs much stiffer springs, raising friction and contact stress; NVH and durability penalties at very high speed [3, 4, 6]	Not sufficient for 10500 rpm
Pneumatic valve return (PVRs, “F1 style”)	Demonstrated well beyond 15000 rpm in racing [3–5]	Requires accumulators, regulators, seals, gas management and fail-safe strategy; packaging and serviceability challenging for road duty; high cost/complexity [4, 5]	High potential but poor fit for high road 9A1
Electromagnetic camless)	(fully Flexible lift/phasing; research demos focus on mid-range speeds; thermal/power limits at very high speed [4, 103]	Large actuators and power electronics; control and fail-safe closure complexity; difficult to package in compact DOHC head [4, 103]	Not suitable for 10500 rpm on this platform
Electro-hydraulic hydraulic (e.g., MultiAir-type)	/ Improves flexibility but retains springs; not aimed at extreme rpm [4]	Substantial oil-system and control changes; does not remove spring-surge ceiling [4]	Not addressing the real limiter
Mechanical desmodromic (selected)	Proven at very high speeds in motorcycles when contact, lubrication and stiffness are engineered [5, 6]	Removes coil springs (no surge); reduces closing-flank normal loads; feasible to package within the 9A1 head using a carrier and rocker/slider mechanisms; first-order literature projection suggests 15–30% reduction of the valvetrain share of FMEP (to be validated), helping offset contact losses and supporting stability [3–6]	Best fit: tackles the limiter with manageable integration

stress, wear, lubrication, axial retention/clearances) are classical mechanical-design problems with clear mitigation paths via materials, coatings, stiffness and endurance testing.

Energy balance and trade-offs The net fuel benefit depends on the balance between reduced spring work and added contact/oil losses in the closing path. In FMEP terms, the expected sign is $\Delta \text{FMEP}_{\text{vt}} \approx -\Delta W_{\text{spring}}/(2\pi) + \Delta W_{\text{contact}}/(2\pi) + \Delta W_{\text{oil}}/(2\pi)$, with the first term dominant if contact stress, finish and lubrication keep the latter two small [4, 8]. Literature on flexible/pneumatic actuation reports that the net is often in the sub-2 % band at representative points [10, 11]. Accordingly, this thesis treats 0.8–1.6 % as a credible, conservative target pending rig and engine validation.

Evidence-based target statement *Target.* Extend the valvetrain’s safe speed from 9 000 to 10 500 rpm. *Justification.* Piston-speed remains within published high-performance practice; the selected desmodromic architecture removes the spring-surge ceiling; alternatives impose materially higher integration burden on a road-capable 9A1 with uncertain durability pay-off. *Stage success criteria (geometric proof-of-concept).* No hard interferences in CAD; manufacturable tolerances with instrumented measurement capability; defined validation path for contact, stiffness and oil-film performance at the target speed (FEM, bench rig and dynamometer steps).

3.3.1 Order-of-magnitude dynamics check for 10 500 rpm

This study serves to provide a simple, defensible calculation that the desmodromic (positive-closure) mechanism can keep the valve under control (no separation) at the target speed of 10 500 rpm, using literature-consistent kinematics and loads [3, 4, 6, 84].

Assumptions (transparent and conservative)

- Engine speed $n = 10\,500$ rpm; four-stroke DOHC camshaft speed $\omega_c = \pi n/60 \approx 550 \text{ rad s}^{-1}$ [3].
- Maximum valve lift $L = 13 \text{ mm}$ (performance DOHC range) [4].
- Closing flank idealised as simple harmonic motion (SHM) over cam angle $\beta = 140^\circ$ (i.e., 280° crank equivalent for lift event), giving continuous velocity/acceleration and zero seating speed [6].
- Equivalent moving mass at the valve: intake $m_i = 0.08 \text{ kg}$, exhaust $m_e = 0.09 \text{ kg}$ (valve, retainer and effective rocker inertia) [6, 84].
- Resistive gas load during closing (conservative): pressure differential $\Delta p = 0.10 \text{ MPa}$ across a 40 mm valve ($A = \pi d^2/4 = 1.257 \times 10^{-3} \text{ m}^2$) $\Rightarrow F_{\text{gas}} \approx 126 \text{ N}$ opposing closure [3].
- Effective cam moment arm near peak load $r_{\text{eff}} = 18 \text{ mm}$ [6].
- Safety factor on normal force $S_F = 1.5$ to cover friction, profile deviation and phasing uncertainty.

Kinematic envelope (SHM, cam angle β)

For SHM over cam angle $\phi \in [0, \beta]$,

$$x(\phi) = \frac{L}{2} \left(1 - \cos \frac{\pi \phi}{\beta} \right), \quad a_{\text{valve, max}} = \frac{L}{2} \left(\frac{\pi}{\beta} \right)^2 \omega_c^2, \quad v_{\text{valve, max}} = \frac{L}{2} \left(\frac{\pi}{\beta} \right) \omega_c,$$

with β in radians and ω_c the camshaft angular speed [3, 6]. Using $L = 0.013 \text{ m}$, $\beta = 140^\circ$ and $\omega_c \approx 550 \text{ rad s}^{-1}$:

$$a_{\text{valve, max}} \approx 3.25 \times 10^3 \text{ m s}^{-2}, \quad v_{\text{valve, max}} \approx 4.6 \text{ m s}^{-1}.$$

3.3.2 Preliminary tribology and wear assessment (qualitative)

The desmodromic layout introduces three critical contact families: (i) cam–follower (roller or pad), (ii) rocker journal pivots (oscillatory hydrodynamic contacts), and (iii) rocker thrust faces. Published measurements and models for cam contacts report minimum EHL films in the few-tenths of a micrometre at elevated oil temperatures, consistent with mixed/EHL boundary conditions depending on speed and viscosity [87, 88, 104]. Oscillatory journal bearings exhibit film

collapse near velocity reversal and therefore operate predominantly in mixed/boundary regimes unless specifically fed and finished for squeeze-film support [105]. False brinelling/fretting at low amplitudes is a known risk in oscillating/thrust interfaces if film replenishment is limited [106].

EHL proxy and λ -ratio targets For design screening, we target a combined roughness $\sigma = \sqrt{\sigma_1^2 + \sigma_2^2}$ of $\approx 0.10\text{--}0.15\ \mu\text{m}$ for cam and follower surfaces (ground/polished), and seek $\lambda = h_{\min}/\sigma \geq 1$ at the most loaded zones. Hamrock–Dowson film correlations (line/elliptical EHL) provide $h_{\min} = f(U, G, W, k) R_x$ with the speed (U), material (G) and load (W) groups; these are standard proxies to tune viscosity and contact geometry [107, 108]. Measured cam–tappet films of order $0.2\text{--}0.6\ \mu\text{m}$ at $\sim 100^\circ\text{C}$ indicate that $\lambda \sim 1\text{--}3$ is realistic with production-grade finishes; this places the conjunction at the EHL/mixed boundary and justifies material/coating support [87, 88].

First-order pressure envelope Cam–roller and cam–pad conjunctions are non-conformal; Hertzian peak pressures are typically in the gigapascal range under valve-train loads. Mixed-TEHL analyses for cam–roller contacts (including thermal and non-Newtonian effects) support design in the $p_0 \sim 1\text{--}2\ \text{GPa}$ band, which is consistent with the contact conditions assumed for high-speed valve trains [104, 109].

Contact-specific qualitative assessment and countermeasures

- *Cam–follower (roller or pad)*: Mixed/EHL regime at high speed and hot oil. DLC on the follower (or pad/roller) reduces friction and wear, with numerous studies showing robust improvements in cam–tappet conjunctions; pairing with a hardened/nitrided cam is a common stack to balance scuff resistance and fatigue [110–112]. Aim for ground/polished $R_a \leq 0.2\ \mu\text{m}$ and monitor film via EHL proxies [87, 88].
- *Rocker journal pivots (oscillatory)*: Predominantly mixed/boundary at reversals; require metered feeds and gentle groove/drain design to maintain replenishment. Bronze/steel pairs with suitable finishes and oil chemistry are tractable; endurance must screen for friction hysteresis and temperature sensitivity [105].
- *Rocker thrust faces*: Oscillating thrust contacts are prone to fretting/false brinelling if lightly fed; use hardened/nitrided thrust washers or DLC-coated thrust faces to mitigate adhesive and fretting wear, with proven benefits from nitriding/DLC in engine hardware [112–114].

Provisional targets for validation (i) Cam–follower: achieve $\lambda \geq 1$ at critical lobes across the speed/temperature window using Hamrock–Dowson proxies with measured oil viscosity–temperature data; verify by comparison to literature film measurements [87, 88, 107, 108]. (ii) Pivots/thrusts: confirm mixed-regime robustness by endurance testing with pressure/temperature logging and

post-test wear maps; absence of false brinelling scars is a pass criterion [105, 106]. (iii) Coatings/heat-treat: down-select DLC grade and nitriding parameters based on published wear/friction data for lubricated steel contacts at comparable loads/temperatures [110, 111, 113].

Required closing normal force and cam torque

Inertial demand $F_{\text{in}} = m a_{\text{valve}}$; the desmodromic closing path must supply a normal force N such that $N \geq F_{\text{in}} + F_{\text{gas}}$ at all times (margin via S_F) [6, 84].

Scope and limitations The values reported in Table 3.6 are *order-of-magnitude screening estimates* based on SHM kinematics and conservative loads. They are intended to size the test rigs and to guide concept iteration. They do **not** replace component-level finite element analysis (stress, stiffness and modal) or multibody valvetrain dynamics with profile-specific kinematics and contact models, which are scheduled in the next phase [3, 4, 6, 84].

Table 3.6: Order-of-magnitude loads at 10 500 rpm (SHM, $\beta = 140^\circ$).

Quantity	Intake (0.08 kg)	Exhaust (0.09 kg)	Notes
Inertial force F_{in} [N]	≈ 260	≈ 292	$m a_{\text{valve,max}}$
Gas force F_{gas} [N]	≈ 126	≈ 126	$\Delta p \cdot A$ (conservative)
Required $N_{\text{req}} = F_{\text{in}} + F_{\text{gas}}$ [N]	≈ 385	≈ 418	Separation avoided if $N \geq N_{\text{req}}$
Design normal $N_{\text{des}} = S_F N_{\text{req}}$ [N]	≈ 580	≈ 630	$S_F = 1.5$
Peak cam torque per valve T_{peak} [N m]	≈ 10.4	≈ 11.3	$T \approx N_{\text{des}} r_{\text{eff}}$

The resulting peak lobe torque per valve (Table 3.6) is of the same order as high-performance spring systems' open-load torque, but here it *guarantees positive closure* with zero reliance on spring surge characteristics [3, 6]. Because valve events are phase-shifted across cylinders, not all lobes reach T_{peak} simultaneously; the shaft sees a ripple within a practicable mean torque envelope [3].

Interpretation and limits

(i) The $N_{\text{des}} \approx 0.58\text{--}0.63$ kN per valve indicates that, with appropriate rocker leverage and cam profile, the desmodromic path can maintain contact at 10 500 rpm [6, 84]. (ii) The SHM assumption gives *lower jerk* near seating and zero seating velocity, which is favourable for impact control; real profiles (polynomial or spline) can be tuned to similar or lower jerk [6]. (iii) A full validation requires profile-specific $N(\phi)$, contact stress and elastohydrodynamic film checks, which are planned in the next stage (FEM/tribology and bench rig) [3, 4]. Nevertheless, these numbers are consistent with published desmodromic practice in high-speed applications [84].

Conclusion

Under conservative assumptions, a mechanical desmodromic layout can plausibly sustain 10 500 rpm without valve separation. The required closing normal force is $\mathcal{O}(0.6$ kN) per valve and

the implied peak lobe torque is $\mathcal{O}(10\text{--}11\text{ N m})$ per valve, which is within the range managed by performance valvetrains when materials, coatings and lubrication are engineered accordingly [3, 4, 6, 84].

Success criteria for Chapter 3

For the present concept to be carried forward as viable, the following criteria should be met during analysis and rig testing:

1. **Positive valve control at 10,500 rpm:** no separation on the closing path (verified by kinematic/force proxies on the rig) and controlled seating speed consistent with the selected profile [3, 6].
2. **Contact stress envelope:** peak Hertzian line/contact pressure at the cam–follower and rocker pivots within the allowable range for the chosen material/coating and oil, with a provisional cap of $p_0 \leq 2.0\text{ GPa}$ pending material-specific confirmation; target film ratio $\lambda \geq 1$ at critical contacts [4, 8].
3. **Geometry and clearances:** no hard interferences in CAD and on rig; rocker end-play per unit within the specified window (0.05–0.10 mm) at build and after endurance [8].
4. **Stiffness and dynamics:** shaft-line/modal deflection small enough to preserve contact stability and timing accuracy over the speed range (validated by FEA and modal checks) [3, 4].
5. **Thermal and lubrication margins:** stable oil supply/temperatures with no evidence of starvation or abnormal heating at the carrier, covers or thrust faces [4, 8].

Meeting these criteria—together with the geometric and manufacturing feasibility established in this chapter—constitutes a practical definition of viability for the proposed springless valvetrain on the 9A1 head.

3.4 Conclusions

This Chapter sets a clear technical brief and boundary conditions for the project: integrate a springless, desmodromic valve actuation concept into the Porsche 9A1 (992 GT3 RS) cylinder head with the explicit performance target of raising the safe maximum speed from 9,000 rpm to 10,500 rpm, while preserving the production envelope and interacting correctly with the original valvetrain architecture. The chapter anchored this ambition in two pillars: (i) a concrete, component-level nomenclature for the desmodromic assembly (cams, small and long rockers, shafts, valve stack and auxiliary elements), ensuring unambiguous communication in later design and metrology tasks; and (ii) a candid statement of implications—head and cam profile redesign to package the mechanism, robust lubrication and thermal management, and structural verification of any new carrier—framing the risks that subsequent chapters would need to retire. Collectively, these elements

converted a high-level aspiration (springless operation to eliminate valve-spring limitations and valve float at high speed) into a measurable engineering problem with traceable requirements and success criteria, consistent with accepted practice in valvetrain development.

Chapter 4

Measurements

This section gathers the key results from the head and donor hardware in one place and explains how they are used. For the Porsche head, the measurements define the geometry that everything must respect; for the desmo parts, they reveal what needs adapting and how much room there is to do it. Together they form the practical baseline for CAD, interference checks and the later skeleton studies.

4.1 Measurements Taken on Engine's Head

The first stage of testing involves precision geometric measurement of the cylinder head using a ZEISS coordinate measuring machine (CMM) with Calypso software. This step ensure the critical features such as valve seat alignment, camshaft positioning, and machined tolerances conform to design requirements. The collected data serves as a baseline for subsequent assembly and performance validation.

For the next sections regarding the measurements, the following nomenclature will be taken:

4.1.1 First Setup

The first setup of measurements was performed with the Porsche head on a normal layout positioned in the ZEISS machine. The main targets for this setup were:

- Define basic measurements needed to define the geometry of the Porsche engine - valve angles;
- Define camshafts' distances;
- Define the height of the head of the engine;
- Define the diameters of the supports of the camshafts;
- Define valve basis offset;

The setup of this measurement step was performed with an aid of the 10 mm blocks to support correctly the engine on the surface. The first step was to measure the "table" plan, which would serve as an reference of the system and to which all measurements will be performed to.

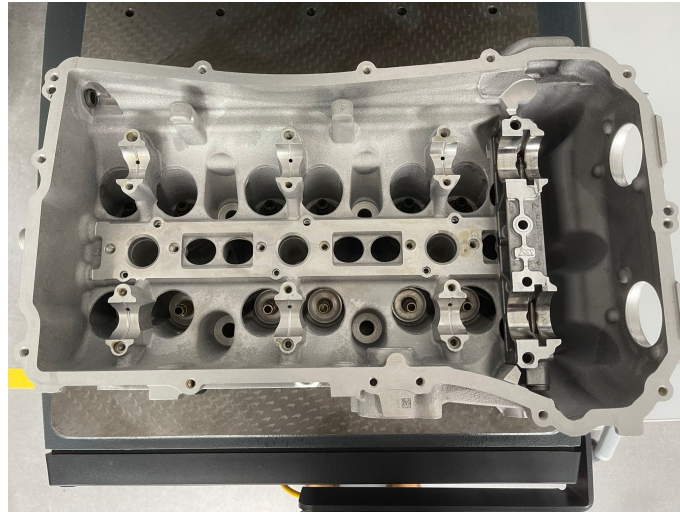


Figure 4.1: First Setup of Porsche measurements

The methodology of this setup was to perform basic geometries - like plans and circumferences and then perform the measurements between these geometries directly in the software. This setup led to the following measurements:

Table 4.1: Measurements from Setup 1

Left	Center
Camshafts Distance [mm]	118,00
Intake Valve Angle [°]	11,10
Exhaust Valve Angle [°]	19,11
Distance from the bottom of the head to the camshaft's plane [mm]	150,60
Height of the Head [mm]	177,60
Diameter of supports for the camshaft [mm]	35,00
Offset on the bottom plane for the Intake valve [mm]	18,00
Offset on the bottom plane for the Exhaust valve [mm]	18,00

4.1.2 Second Setup

In the second setup, the Porsche head was turned upside down in order to measure the perpendicular distance between the plan of the valve and the plan of the bottom of the head. This was performed considering the height difference between the valve seat planes and the bottom plan of the engine head. The setup can be seen below on Figure 4.2.

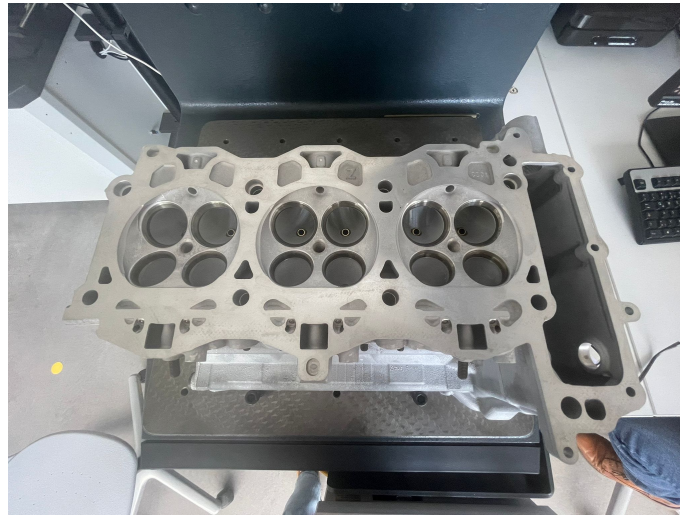


Figure 4.2: Second Setup

With this setup the goal was to gather the measurements available on the lower part of the motor head. The measurements taken are presented on the Table below:

Table 4.2: Measurements from Setup 1

Left	Center
Distance between cylinders [mm]	118,00
First Diameter of support [mm]	22,50
Second Diameter of support [mm]	12,50

Regarding the distance between cylinders, a different method was taken for measuring. As previously mentioned, the machine only works on machined surfaces, however, the definition of contour of the cylinders is defined by mostly casted surfaces. However, there are two small surfaces on each side of the bore that are machined.

Therefore the machine was moved horizontally from one side of the bore to the other, passing through the theoretical centre of the cylinder until the other side, simulating a close approach to the real cylinder bore.

A second part on this second setup was to use shafts of 6 mm to measure the bottom plan of the valve, in a setup like the picture below:



Figure 4.3: Second part of Second Setup

With this second addition it was decided to take initial measurements regarding the placement of the different radii for the different valves. Based on said measurements the results achieved can be seen below. All measurements represented from now on are all in milimeters:

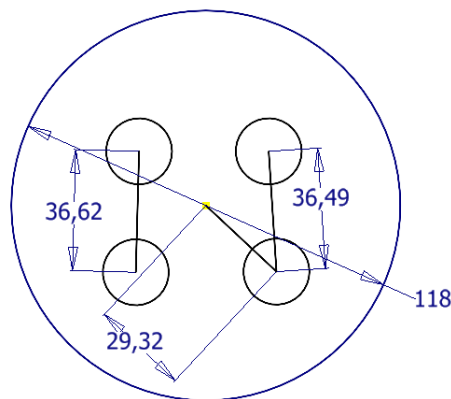


Figure 4.4: Second Set of Measurements

Besides, using the shafts, especially the machined surface, an approximation of how deep the valve enters when closed was also measured. This measurement is important because it provides the information of height to which the valve starts when it is completely closed. Also, the inclination of the valves was once again measured to make sure the data previously obtained was correct.

4.1.3 Third Setup

For the third setup, the main goal was to measure the different fixation points existing on the Porsche head, by measuring the radii of the fixation points and the distance between them. This process started with measuring the coordinates of each point (in relation to a software defined axis system), and then to implement triangles to measure the different distances between these points.

For this setup the orientation of the Head was equal to the previously defined, however the machined shafts had to be removed.

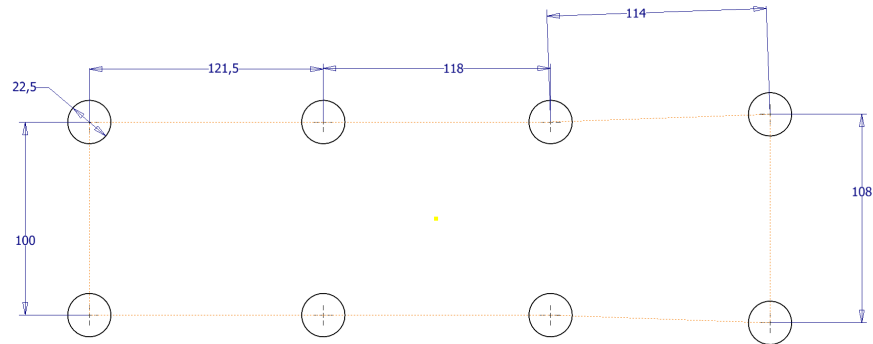


Figure 4.5: Parallel and Perpendicular distances between Fixation Points

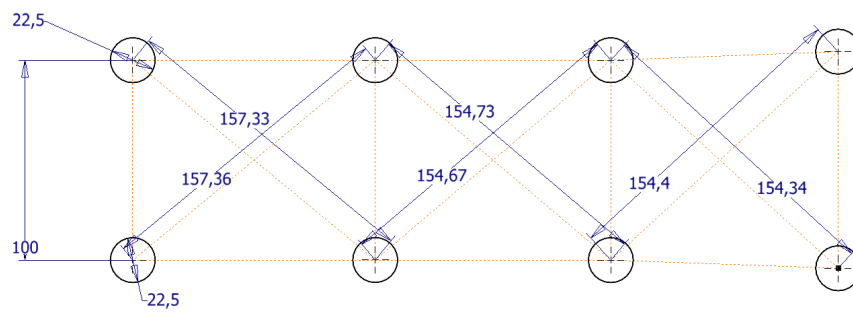


Figure 4.6: Diagonal distances between Fixation Points

The next step was to take the define the measurements related to the centers of the cylinders. These measurements can be seen below:

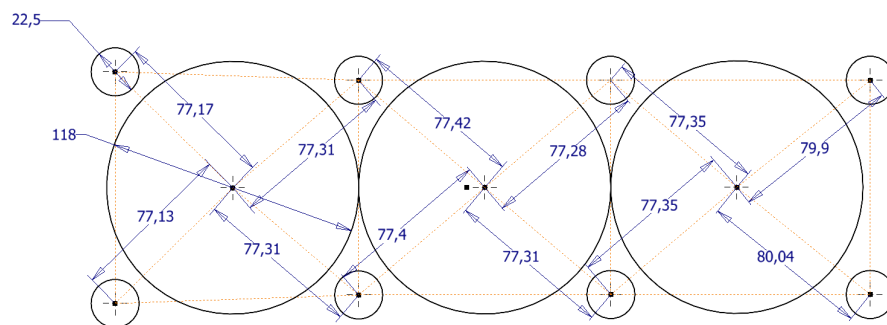


Figure 4.7: Distances related to the center of cylinders

4.2 Measurements taken to the Desmo's valvetrain

4.2.1 First Setup

For the first setup of the desmodromic component, the Desmo Head was positioned with the side of the end of the valves facing up, and with the goal of taking basic measurements:



Figure 4.8: First Setup on the Desmodromic

The main measurements are presented in Table 4.3:

Table 4.3: Measurements from Setup 1 of Desmodromic

Left	Center
Diameter of supports [mm]	10,50
Inclination of top the Desmodromic head [°]	5,00
Inclination of Exhaust Valves [°]	13,63
Inclination of Intake Valves [°]	12,04
Diameter of Camshaft supports [mm]	24,00
Distance between Camshafts [mm]	85,50

It is relevant to mention that the shafts used for support of the Desmo head were standard squared shafts to ensure a regular support (in case of irregularities on the surface of the machine).

4.2.2 Second Setup

On the second setup, the Desmodromic component was placed on this regular position, with the camshaft support facing up. The main target of this setup was to measure the distances between the different camshafts in the relation to the bottom plan of the Desmo head.

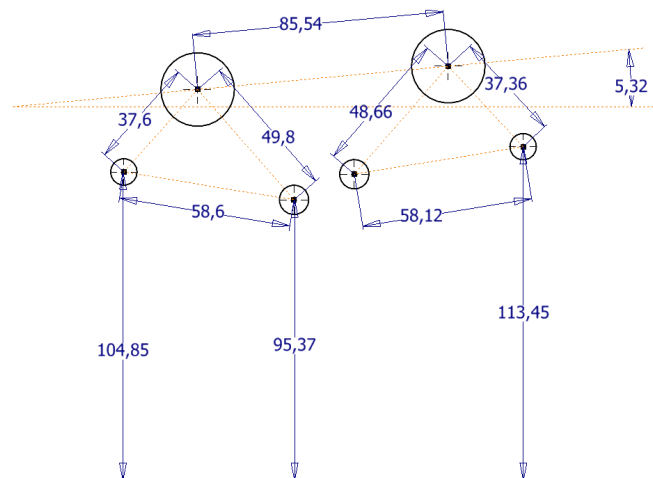


Figure 4.9: Measurements taken for the Second Setup

Once again the support of the Desmo head was performed with the aid of square shafts.

4.3 Consequential Measurements

Next up was measuring the Desmo and relevant Porsche components for the final assembly. This was needed to mimic as real as possible the original components in the final assembly.

The first component measured consequently was the Small Rocker (Component 2 - 3.2). The sketch performed resulting from the measurements is presented on Figure 4.10.

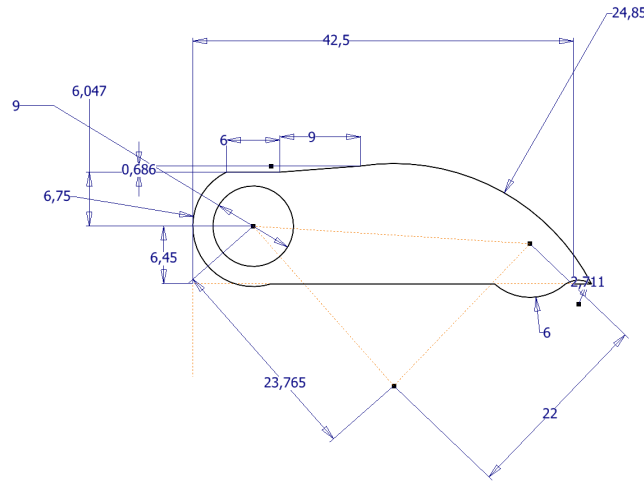


Figure 4.10: Sketch of Component 2 - 3.2

Given this geometry, an extrusion was performed of 9 millimeters, resulting in the following piece:

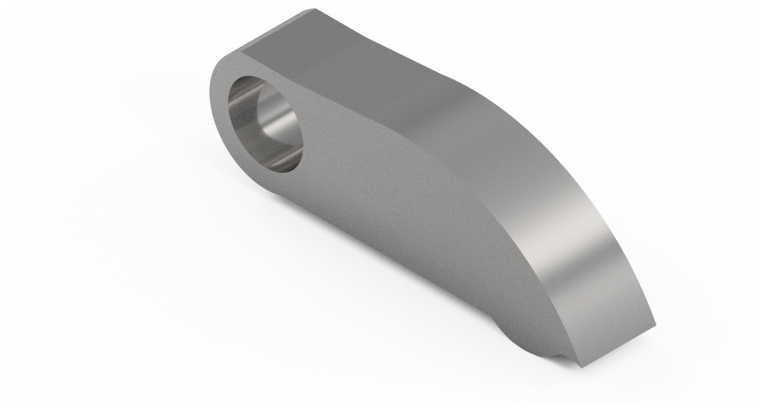


Figure 4.11: Extrusion of Component 2 - 3.2

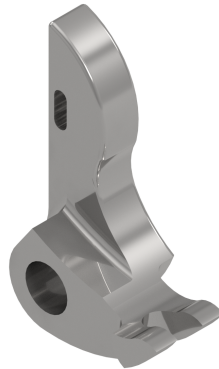


Figure 4.14: Extrusion of Component 7 - 3.2, for the intake side

Then, the designs for the Bushing were made using a caliper. The final CAD is presented:

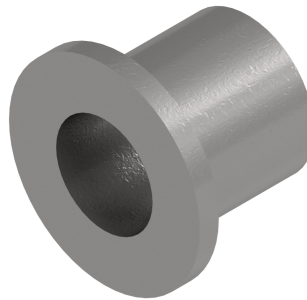


Figure 4.15: Extrusion of Component 12 - 3.2

Another component measured in a similar way was the Beating component on the top of the valve.



Figure 4.16: Cross-Section of Component 6 - 3.2



Figure 4.17: Extrusion of Component 6 - 3.2

The final measurements were to define the valves. It was decided to draw both the Desmo Valves and the Porsche Valves. The final components are presented all below.

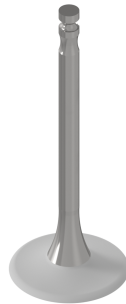


Figure 4.18: Extrusion of Component 10 - 3.2

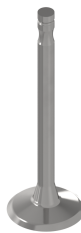


Figure 4.19: Extrusion of Component 8 - 3.2

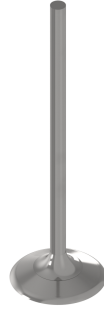


Figure 4.20: Extrusion of the Exhaust Porsche Valves

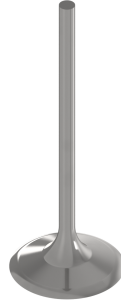


Figure 4.21: Extrusion of the Intake Porsche Valves

Then, the masses measured on a bench balance are compared directly with the CAD-derived values (“Mass Properties”) for the same parts, using the component names adopted elsewhere in the thesis (e.g., finger follower, rocker springs and shafts, valve retainer, and Porsche intake/exhaust valves). All parts were weighed clean and dry; software values were computed on the final models with nominal material densities. The error reported in the last two columns is defined as the Equation below:

$$E_g = m_{\text{soft}} - m_{\text{meas}}, \quad E_{\%} = \frac{m_{\text{soft}} - m_{\text{meas}}}{m_{\text{meas}}} \times 100. \quad (4.1)$$

A positive value therefore indicates that the software prediction is heavier than the balance reading, and a negative value the opposite. This compact check is used simply to confirm that the digital models are consistent with physical hardware for subsequent mass and inertia accounting in the valvetrain analyses. The values are present below:

Table 4.4: Mass comparison between measured parts (balance) and software-estimated masses. Positive error means the software value is heavier than the measured value.

Component	Measured [g]	Software [g]	Error [g]	Error [%]
Valve stopper	6.051	6.169	+0.118	+1.95
Rocker bush	11.372	11.422	+0.050	+0.44
Finger follower (Porsche)	29.759	30.893	+1.134	+3.81
Long rocker spring (intake)	75.435	76.997	+1.562	+2.07
Long rocker spring (exhaust)	75.421	76.563	+1.142	+1.51
Finger-follower shaft	24.500	24.505	+0.005	+0.02
Spring shaft	32.500	32.847	+0.347	+1.07
Small rocker spring (intake)	11.720	11.295	-0.425	-3.63
Small rocker spring (exhaust)	11.720	11.305	-0.415	-3.54
Valve retainer (top)	2.502	2.526	+0.024	+0.96
Exhaust valvetrain assembly	28.238	27.663	-0.575	-2.04
Intake valvetrain assembly	37.669	36.328	-1.341	-3.56
Exhaust valve (Porsche)	32.792	31.632	-1.160	-3.54
Intake valve (Porsche)	38.994	37.873	-1.121	-2.87

The deviations in Table 4.4 show a mixed pattern. Larger machined parts sit on the positive side, up to +3.81% for the Porsche finger follower. Small springs sit on the negative side, down to 3.63% (in module) for the intake rocker spring. This asymmetry aligns with incomplete CAD features such as threads, fillets, and small cavities, and with density scatter from alloys after heat treatment or coating. Trapped oil, adhesive residue, and surface wear also add grams on measured parts. Finally, the Scale resolution introduce small spread, consistent with the low end of the range.

The magnitude remains modest for system-level predictions. A mass bias within $\pm 4\%$ translates into a similar bias on inertia for parts with fixed geometry. Natural frequencies scale with the square root of mass, so a +4% mass lowers frequency by roughly 2%. Contact load peaks depend more on kinematics and stiffness than on these mass offsets. The figures therefore support use of CAD masses for comparative design and for order-of-magnitude dynamic checks, with local uncertainty bands carried into results.

4.4 Development of skeletons

To compare architectures rigorously and to drive a clean, top-down design, skeleton parts for both platforms were created — the Porsche 911 GT3 RS bank and the Ducati desmodromic head — and used them as the single source of truth for datums, axes, control curves and envelope surfaces. In Creo Parametric, a skeleton centralises these master references and makes it simpler to analysis. This workflow also keeps manufacturability and inspection in view, because the skeleton can host the drawing datums, gauge coordinate systems and “no-go” volumes. Each skeleton was built in

millimetres with parameters recorded to two decimal places (e.g., $Cylpitch = 118.00$ mm).

It is important to mention that one of the targets with this development was to study the position of the camshafts, considering not only the target of changing the least amount of the original material, but also considering the team did not have access to the original chains and gears of the Porsche and the original chain of the Desmo component.

The first skeleton performed was the **Porsche GT3 RS** one. That can be seen below in the Figure 4.22. Both the Porsche and Ducati skeletons were produced with the same disciplined steps. First, the three principal datum planes were fixed — deck, bank mid-plane and a transverse plane through the reference bore centre—and a bank coordinate system (CSYS) with X along the bank, Y across the bank and Z normal to the deck. Second, the bore circles were sketched and set to 118.00 mm for the Porsche diameter; each bore CSYS carries named axes for later patterning. Third, the valve-axis curves were added (two intakes, two exhausts) as datum lines that pierce the chamber envelope; their angles and offsets are known parameters. Fourth, the actuation axes were added (upper and lower shaft lines) and the compact rocker keep-out as 2D features with driving parameters.

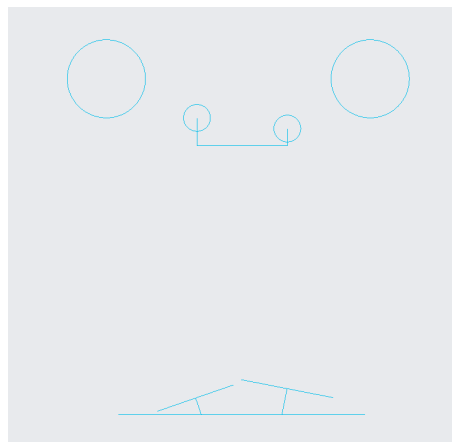


Figure 4.22: Porsche Head Skeleton

Then, the Valve Train parameters were defined. Starting with the planes perpendicular to the Valve Axis, 6 planes were defined - 2 planes for the ends of the Porsche Valves; 2 planes for the end of the spring on the Porsche Valve; 2 planes for the placement of the Spring on the engine. Next, a sketch was performed including the positioning of the camshafts, as well as the positioning of the original rockers. All of these components had specific axis designed for them as well. The final step was to define planes for these components to certify if the inclination and distances were all truthfull to the measurements.

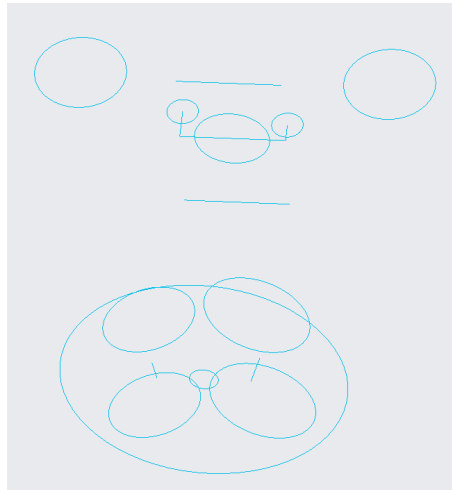


Figure 4.23: Porsche Head Detailed Skeleton

The second skeleton performed was the Desmo one. This skeleton required much more detail considering the different components associated with this system. The initial steps were similar to the previously defined system, only with different values. The differences began in the Valve Position Group.

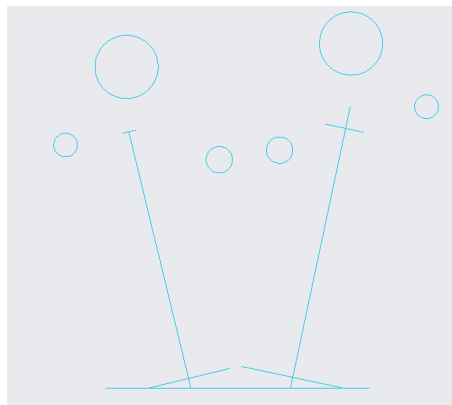


Figure 4.24: Desmodromic Skeleton

Firstly, the planes related to the Valve were defined. Once again, there were 6 planes defined - 2 planes for the end of the valves; 2 planes for the distance where there is the upper movement; 2 planes for the distance where there is the down movement. Afterward the camshaft and associated shafts measurements were integrated, always respecting the inclination that the camshafts have on the case of Desmo system. All of the respecting axis and planes were then created.

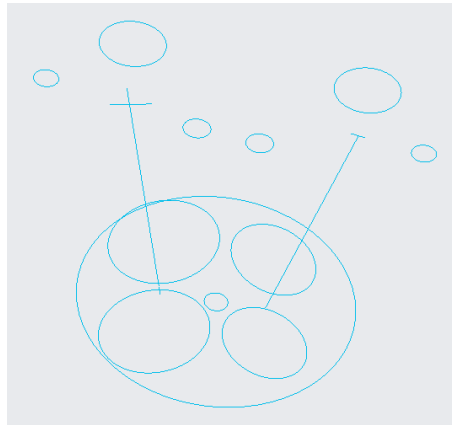


Figure 4.25: Desmodromic Angular Skeleton

The next step was to divide these skeletons into Exhaust and Intake and study if, given the same valve inclination, where would the camshafts' axis and the shafts' axis would sit in relation to the valves. One assumption that was made at this point of the project were that new valves would have to be made during the manufacturing of the project. The other condition for the study was to make the set of axis of each skeleton coincident.

This has to happen not only because of the liberty on length that this provides but also because of the difference on mechanism of movement from the Porsche Valves to the Desmo Valves. This should not bring questions on the durability of the valves considering the material would be the same as the original and the diameter will be the same as the original one.

This study was then performed with the aid of the Creo Parametric software, and the final results are present below:

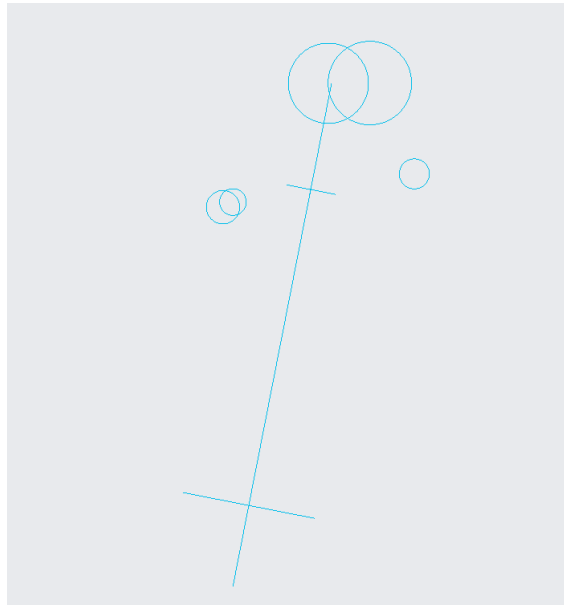


Figure 4.26: Study for the Intake Side

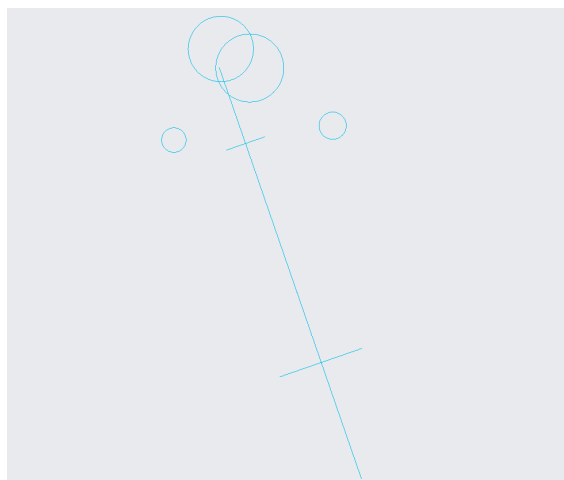


Figure 4.27: Study for the Exhaust Side

As one can see, the distances between the Porsche camshaft and the Desmo camshaft are considerable small when joining the two axis. Using the Measure tool from the Creo software, the linear distances between the camshafts were measured. The following results were obtained:

Table 4.5: Distances between camshafts after study

Side	Distance [mm]
Intake	12.67
Exhaust	12.41

Joining both comparisons in an only sketch, the conclusion was that the placement of the Desmo camshafts can be adapted for the Porsche placement. The distances concluded were of about 10% of the total distance between camshafts (118.00 mm).

4.5 Scanning and digitalisation of the Desmo and Porsche components

This subsection documents the scan-to-CAD work undertaken on the desmodromic donor parts and on selected Porsche 9A1 head components. Optical scans were aligned to the CMM datum scheme and exported as triangulated meshes (STL/OBJ). Meshes were cleaned and reduced, then taken through reconstruction attempts in Autodesk Fusion 360, Autodesk Inventor, PTC Creo, SOLIDWORKS, Blender, MeshMixer and 3ds Max. The objective was to obtain an editable parametric CAD master that preserved critical geometry while keeping file size and feature count within practical limits for downstream design and tolerance analysis [79, 80].

Fusion-centric workflow

Most effort, and the most productive intermediate results, were achieved in Autodesk Fusion 360. Following mesh hygiene (outlier removal, hole filling and uniform decimation), the dataset was sectioned on functional planes (deck, bearing spans, representative port chords) and sketches were created to anchor later surfaces. Conversion focused on Fusion's Mesh to BRep tool ("Converter malha" in the Portuguese interface), exercising its three modes: Facet, Prismatic and Organic. The practical approach was:

- begin from a reduced but disciplined mesh, with chordal error constrained in critical regions;
- apply Facet mode where planar and cylindrical regions dominated, to create lightweight bodies quickly;
- use Prismatic on local prismatic features (bosses, ribs, bores) to recover editable faces from sections;
- reserve Organic for free-form cast zones and knit patches back to section curves;
- iterate between local re-meshing and additional decimation whenever face-count limits or kernel errors were reached.

This Fusion-first path allowed rapid section-driven iteration and selective conversion of local features into manageable BReps while keeping the remainder of each part as mesh for context. Whole-part conversions remained impractical on the heaviest datasets, but repeated, stable feature-level conversions were obtained around bearing seats, covers and datum faces [115].

Workflow and operations

The following steps were applied iteratively across the tools listed.

- **Mesh clean-up and reduction.** We stripped noise and spikes from the raw meshes, closed small holes and applied only light smoothing. Decimation was driven by a target chordal error so that critical regions stayed within the intended tolerance band while non-critical cast areas were simplified [80].
- **Registration and sectioning.** All scans were registered to the A–B–C datum frame used in metrology. We then cut cross-sections at functional planes (deck, bearing spans, port sections) to anchor later surface fitting and to cross-check against the CMM dimensions [79].
- **Feature extraction.** Where possible the software auto-detected planes, cylinders and cones; in free-form zones we traced section curves by hand to avoid artefacts when automatic tools introduced noise.
- **Surfacing and solid trials.** Using mesh-to-BRep and reverse-engineering tools, we laid out NURBS patches and trimmed surfaces, then attempted to knit them into watertight solids suitable for STEP/IGES export.

Software used and observed behaviour

Result and constraints

Despite systematic mesh refinement, disciplined sectioning and targeted reduction of non-critical detail, a robust full-solid CAD master was not achieved. The principal constraints were: (i) multi-million facet counts in the raw scans driving memory and kernel limits during Mesh to BRep or surface knitting; (ii) local non-manifold edges and small gaps after patch fitting, preventing watertight solids; and (iii) the combination of tight-tolerance features (bearing seats, deck references) with highly free-form cast surfaces, which demands patch quality and continuity (G1/G2) beyond what could be achieved within the available tool settings and compute budget. In practice, only local, feature-level conversions were stable; whole-part conversions either failed or produced bodies too heavy to edit reliably [115, 117, 118].

For the purposes of this thesis, the cleaned and aligned meshes remain the authoritative geometry for qualitative interference checks and context, while CMM-derived features provide the numeric reference for tight tolerances; despite several stable, local Mesh-to-BRep conversions in Fusion around critical features, a single editable solid of the whole assembly could not be produced within practical limits, reflecting the central trade-off between file size and geometric fidelity. The next iteration will therefore segment the assembly into functional zones, rebuild each with tolerance-driven surface fitting, cap mesh density per zone before any BRep attempt, and evaluate a specialised reverse-engineering package with stronger regioning and patch workflows

Table 4.6: Scan-to-CAD trials: role, key functions, outcome and limits.

Software	Role in pipeline	Key functions used	Outcome and limits
Fusion 360	Mesh clean-up and CAD conversion	Reduce mesh; Remesh; Section Sketch; Fit Curves; Mesh to BRep	Best intermediate results; reliable on local features; whole-part conversions hit kernel and face-count limits on dense meshes [115].
Autodesk Inventor	Mesh import and sectioning	Mesh handling; Section views; attempted BRep via add-in	Heavy meshes required aggressive decimation; parametric conversion unstable on complex areas [116].
PTC Creo	Reverse engineering and surfacing	Import Data Doctor; curve networks; REX surfacing	Good patch control; performance dropped with multi-million facets; solids knitting incomplete [117].
SOLIDWORKS	Scan-to-CAD and primitives	ScanTo3D; Mesh Prep Wizard; Surface/Loft	Useful for primitives and small free-form zones; global conversion limited by file size [118].
Blender	Retopology and mesh editing	Shrinkwrap; Quad Remesh; manual retopo	Effective for topology simplification; requires hand-off to CAD for NURBS/solids [119].
MeshMixer	Fast reduction and repair	Reduce; Make Solid; Analysis/Inspector	Quick decimation and fixing; not intended for precise NURBS creation [120].
3ds Max	Retopology and segmentation	Retopology; Optimize; Slice	Helped segment regions and lighten meshes; export to CAD still needed for parametric features [121].

for large free-form parts, with all reconstructed surfaces verified back against the scan in the established datum frame [79, 80].

4.6 Measurement of Desmo and Porsche camshafts' lift curve

This section describes the bench used to acquire valve-lift profiles from the desmodromic and Porsche camshafts and the processing route followed to generate lift-versus-angle curves.

Measurement rig (as-built)

The camshaft is supported between two rigid aluminium pedestals bolted to an extruded base, with the journals located on cylindrical rests to set height and axis straightness. The right-hand end is clamped to a knurled hub and coupled coaxially to an incremental encoder configured at 256 pulses per revolution. With x4 decoding the counter yields 1024 counts per cam revolution, i.e. an angular resolution of about 0.35° per count. A vertical slide carries a contact displacement probe (spring-loaded, LVDT-type) that bears on each lobe through a small hardened roller/pin to minimise side load. A National Instruments multifunction USB DAQ is used: the encoder channels are read on

a counter input and the probe signal on an analogue input. The whole assembly is hand-driven at low speed for quasi-static capture, avoiding inertial effects and allowing repeatable phasing [3, 6, 122, 123].



Figure 4.28: Setup for measuring the camshafts' lift curves

Signal path and acquisition

Encoder A/B (and index, when available) provide absolute cam angle over each revolution; the probe delivers displacement in volts. The probe is zeroed on the base circle region with a small preload, then a slow, continuous rotation is made while logging angle and displacement synchronously. For each lobe, three to five revolutions are captured and averaged to reduce random noise and operator variability; a brief repeatability check is made at the same setup before swapping camshafts [81, 83]. Sampling is event-based on the encoder counter but the data are resampled onto a uniform angle grid (typically 0.1°) by interpolation to simplify differentiation and timing extraction.

Post-processing

The displacement channel is converted to lift by subtracting the minimum value over the base circle:

$$h(\theta) = x(\theta) - \min\{x(\theta)\}.$$

When a roller/pin is used, the small kinematic offset is corrected by subtracting the roller radius from the effective base-circle radius; for these quasi-static measurements this correction is small but included for consistency [6]. Opening and closing timings are reported at a 1.0 mm lift threshold; peak lift, duration, and simple acceleration/jerk envelopes are obtained by numerical differentiation after light smoothing. All datasets are exported as CSV and plotted in **UniPlot** for curve comparison and report figures.

Procedure (both camshafts)

1. Mount and align the camshaft on the pedestals; set axial position against a hard stop.
2. Set the probe on the target lobe with a small preload; zero the displacement channel on the base circle.
3. Rotate the camshaft slowly and continuously; acquire angle and displacement for at least three full revolutions.
4. Repeat for all lobes of interest; repeat the sequence after swapping to the second camshaft so both datasets are directly comparable.
5. Process and export the resulting lift curves to UniPlot; archive raw time-angle data alongside processed angle-lift files.

Cross-read for the 9A1 adaptation

- **Amplitude compatibility.** Desmodromic peak lifts (about 13–14 mm intake; 9–10 mm exhaust) align with 9A1 levels, avoiding a lift-limited breathing penalty.
- **Low seating speed.** Both desmodromic closing profiles reduce velocity consistently near zero lift, supporting controlled seating at target speed.
- **Smooth derivatives.** Velocity and acceleration traces are continuous (no cusps), anticipating moderated jerk and reduced risk of structural excitation in the carrier/shafts.
- **Angular normalisation.** For direct overlays later, normalise crank angle to a common reference (e.g., 0° at opening @ 1 mm lift) and resample to uniform angular resolution.

Limitations and data handling

Peak readings above are read off the plotted scales. For quantitative use (multibody/FEA excitation), export lift, velocity and acceleration as numeric arrays at uniform crank resolution; apply only minimal smoothing if necessary and retain true peak values.

4.7 Lift curves: results and interpretation

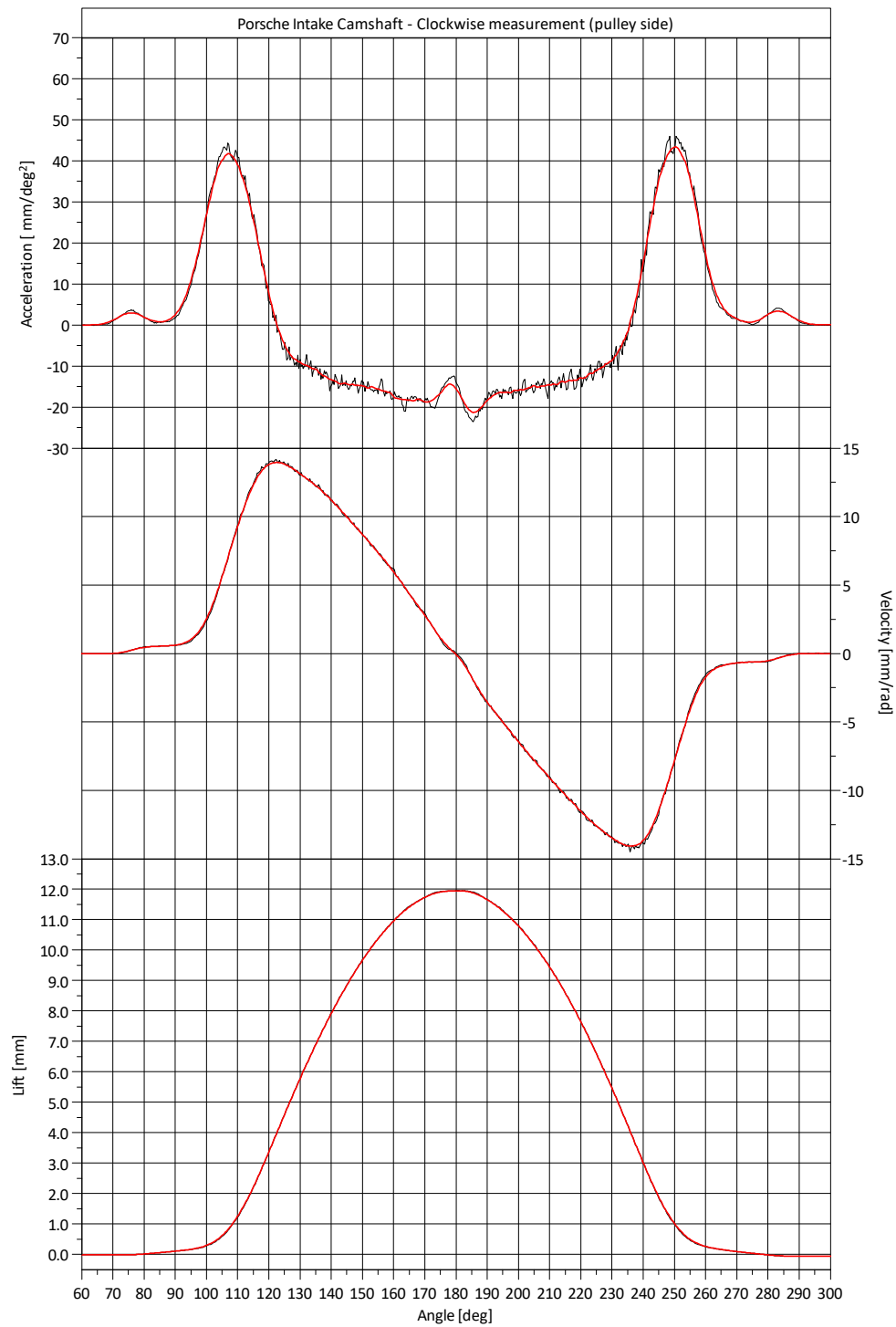
Conventions and dataset

All profiles were measured on the pulley side with the indicated rotation sense: CW (clockwise) or CCW (counter-clockwise). For the desmodromic train, the opening and closing cams were measured separately and also overlaid “in phase”. In plots of the desmodromic closing lobe, the lift axis may appear with negative sign only to improve visual separation; absolute magnitude matches opening.

Porsche 9A1 baseline cams

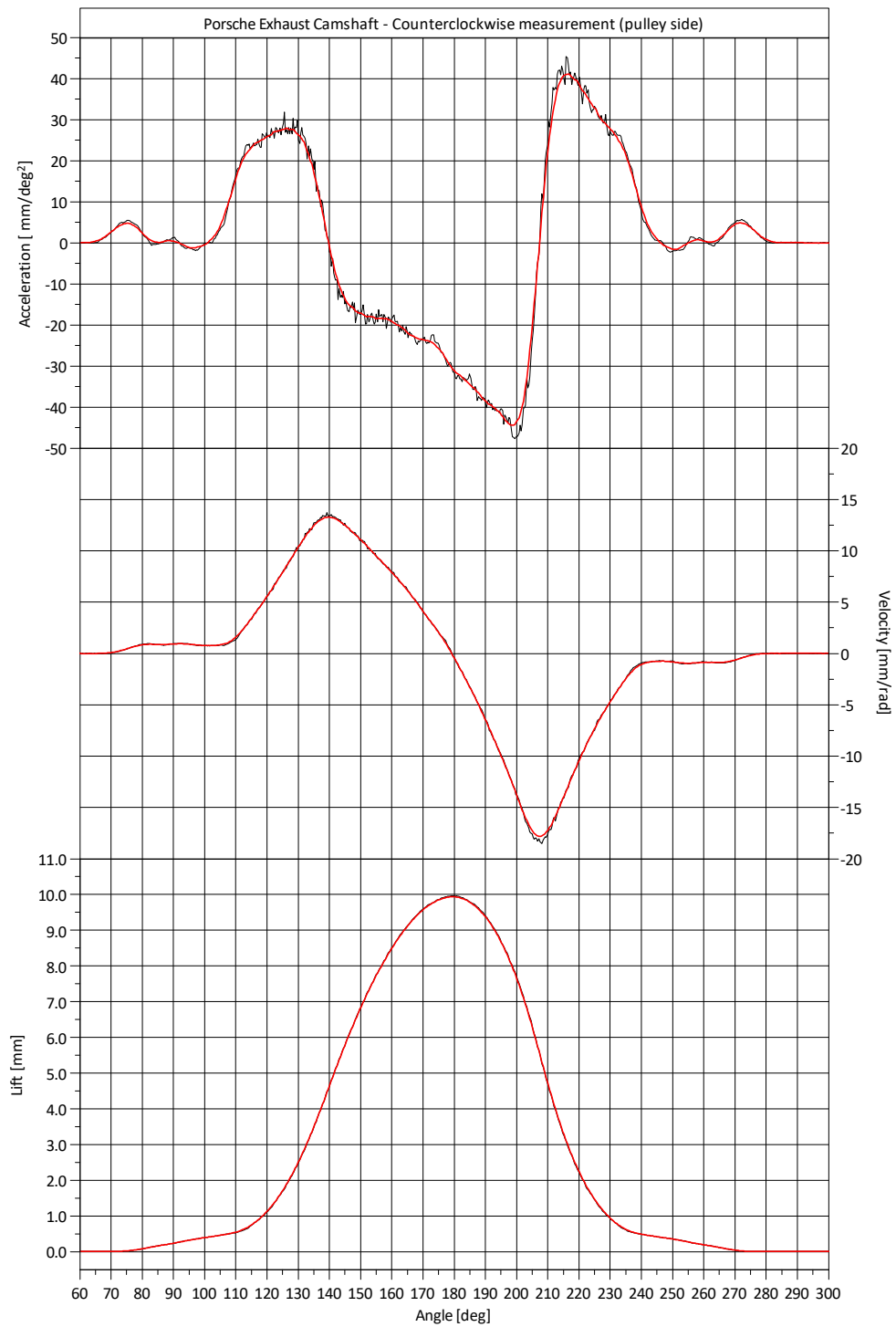
Intake (CW). The intake lift peaks around 12–13 mm, with a brisk opening flank, a short plateau near maximum lift, and a softened closing flank approaching seat (velocity tending to zero). Acceleration is continuous and smooth.

Figure 4.29: Porsche 9A1 intake cam (CW): lift-velocity-acceleration measured on the pulley side.



Exhaust (CCW). Exhaust peak lift is slightly lower (10–11 mm). Peak velocities are somewhat higher than on intake, but the closing flank again slows clearly near zero lift; acceleration extrema are more pronounced yet smooth.

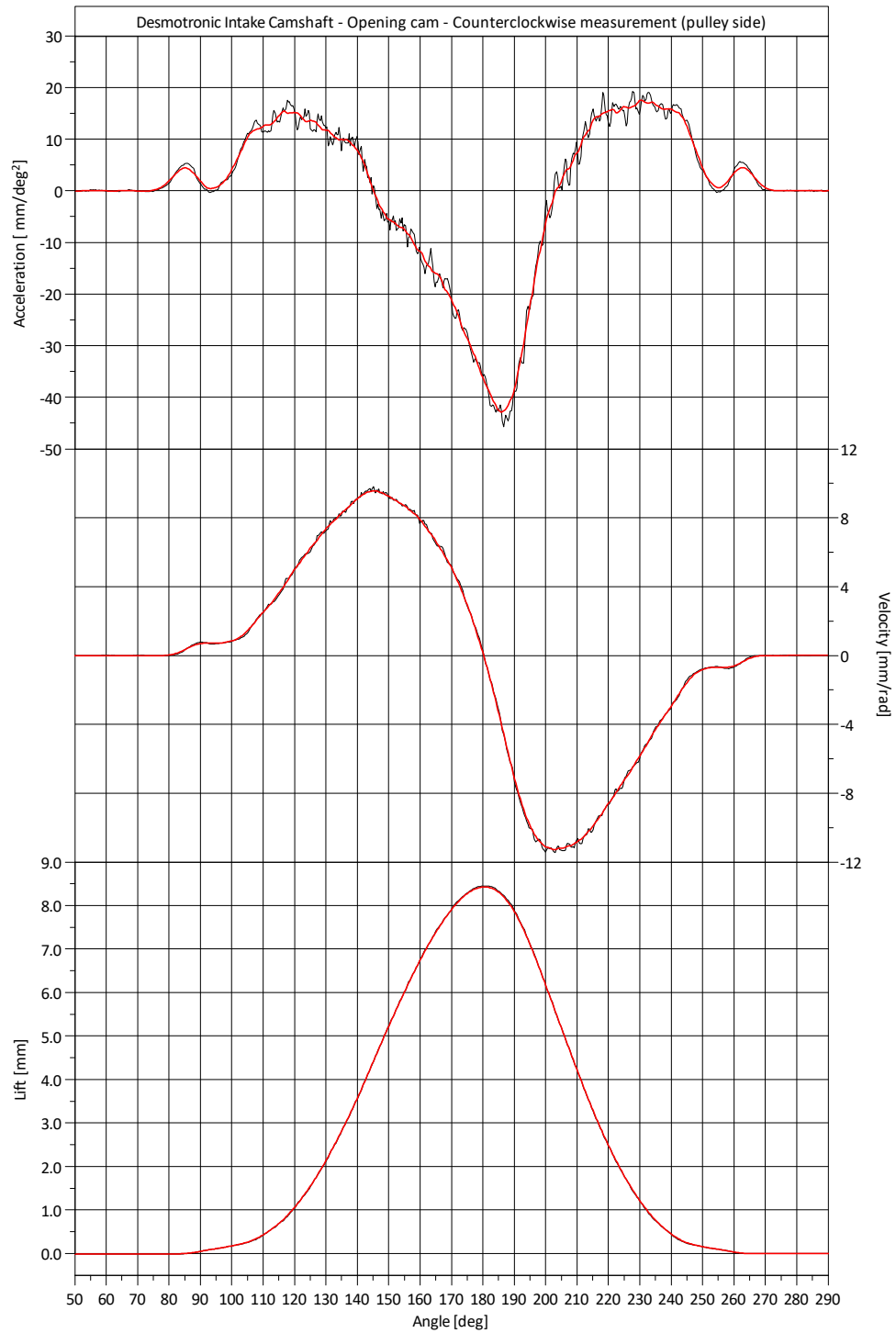
Figure 4.30: Porsche 9A1 intake cam (CW): lift-velocity-acceleration measured on the pulley side.



Desmodromic intake

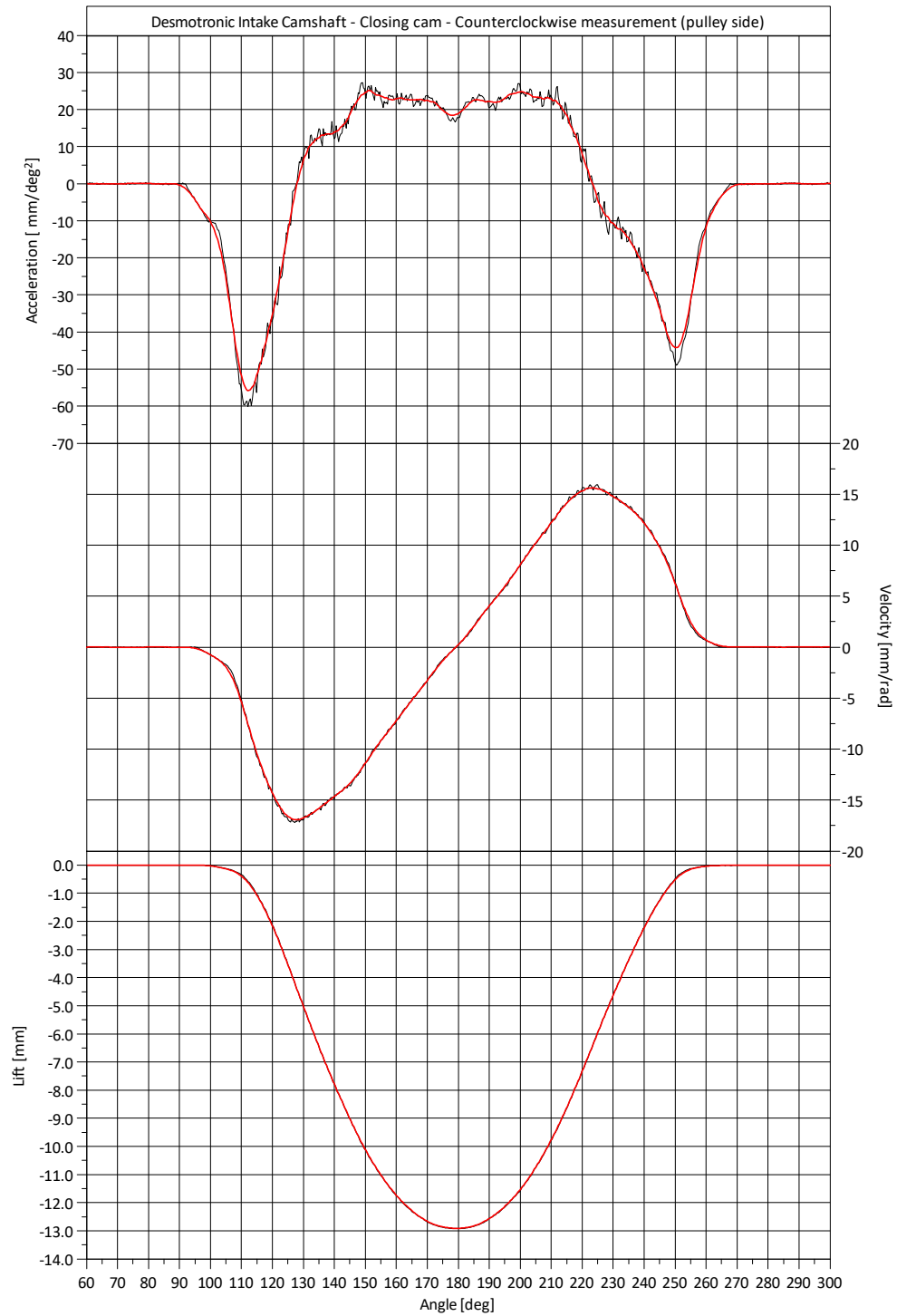
Opening (CCW). Opening amplitude is about 13–14 mm with clean, monotonic flanks and peak velocity in the low tens of mm rad^{-1} ; acceleration remains continuous.

Figure 4.31: Desmodromic intake — opening cam (CCW).



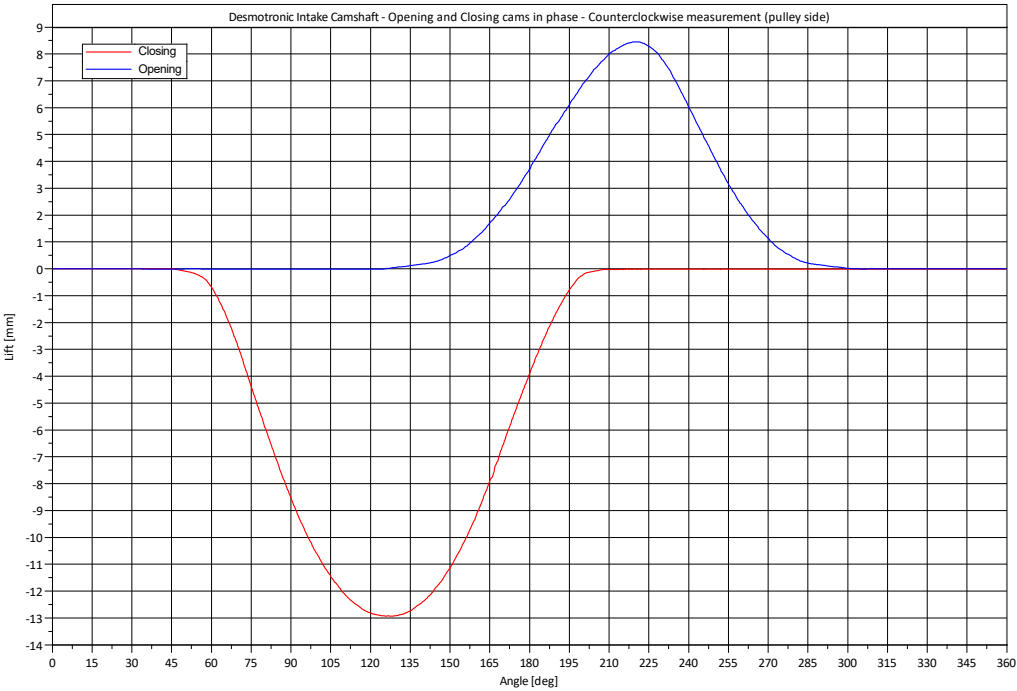
Closing (CCW). Closing shows decisive mid-stroke deceleration and a clear reduction of velocity near seat (tending to zero at small lifts), consistent with positive mechanical closure.

Figure 4.32: Desmodromic intake — closing cam (CCW).



Opening and closing in phase. The overlay demonstrates continuity with no dead zone at hand-over and a low-velocity approach to seat.

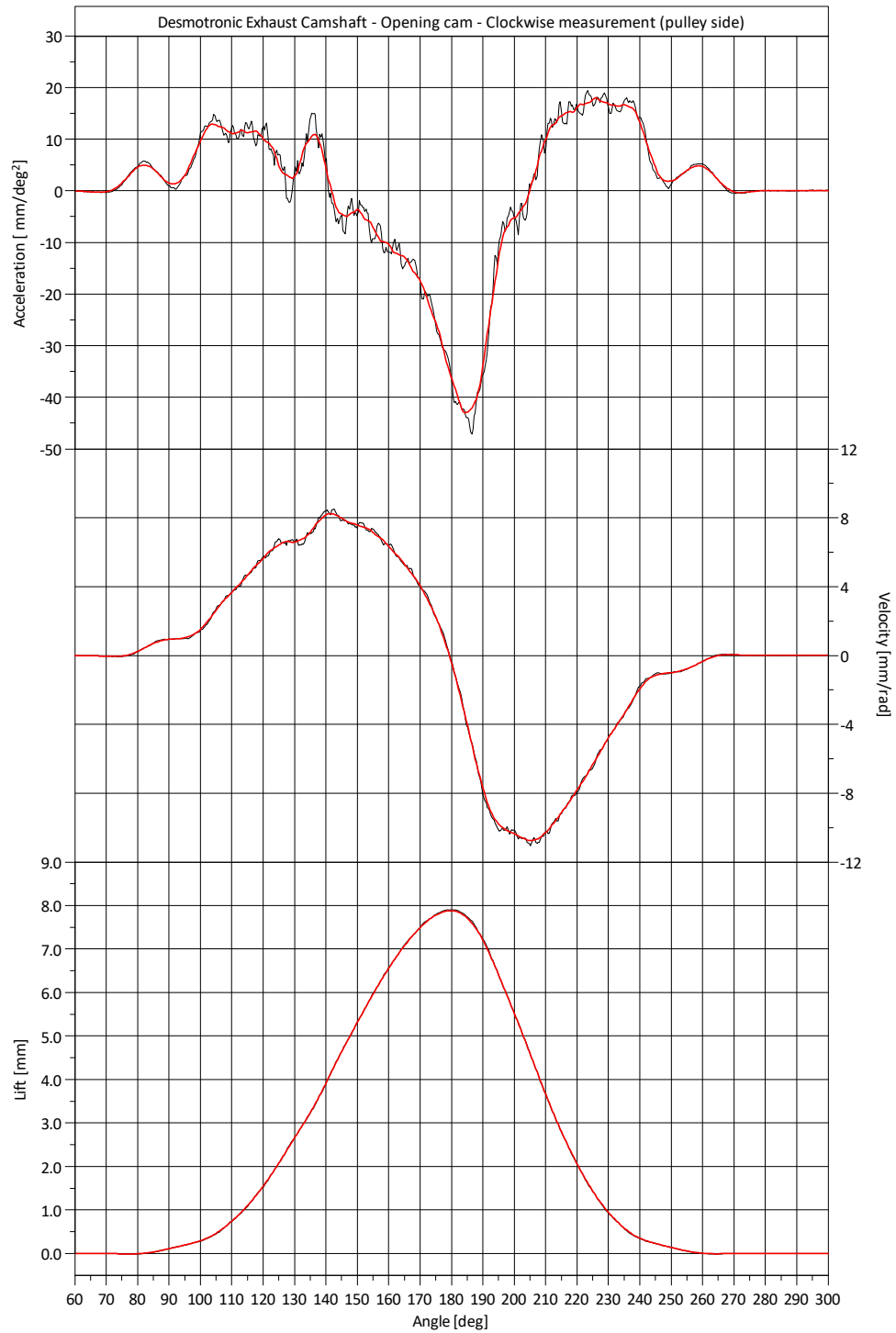
Figure 4.33: Desmodromic intake — opening and closing in phase (overlay)



Desmodromic exhaust

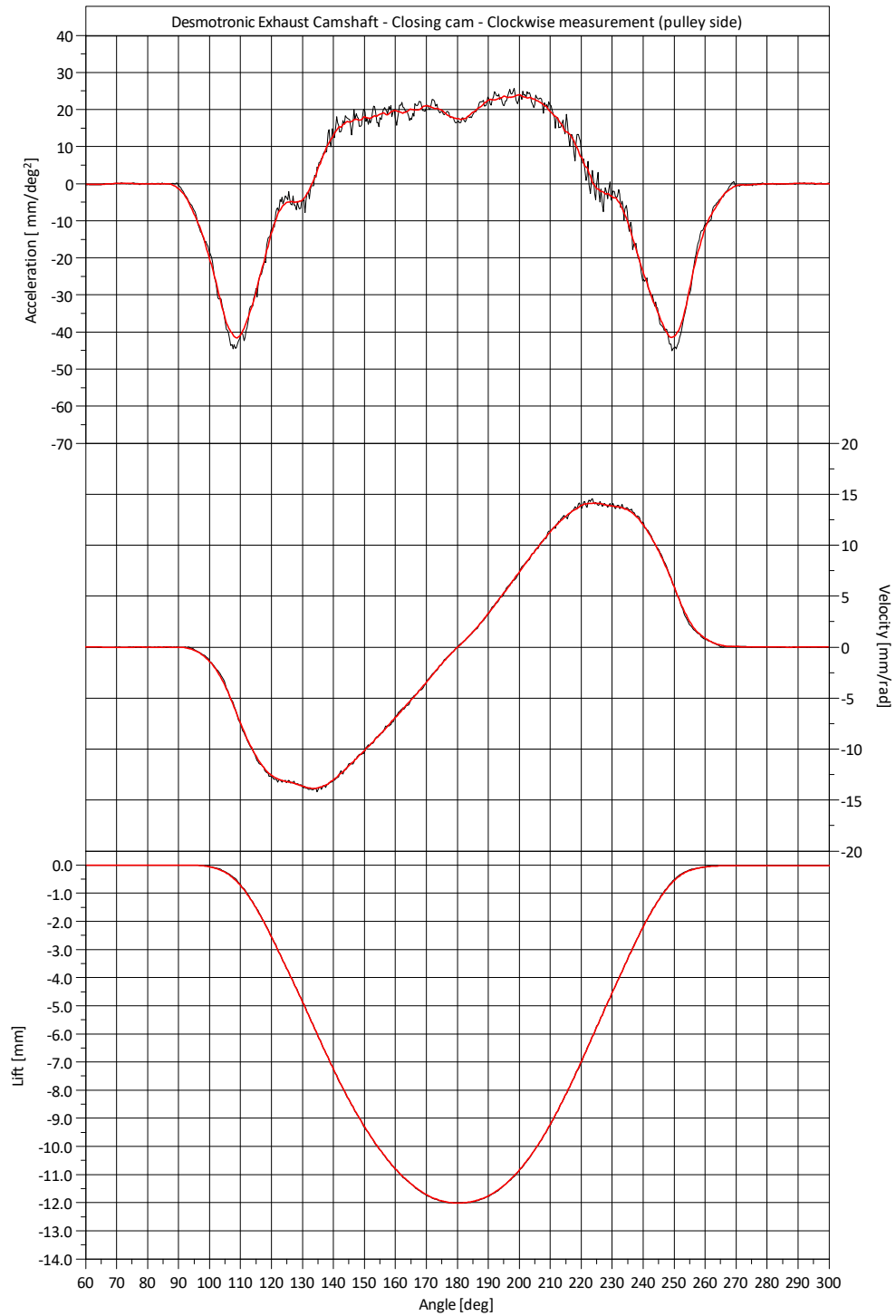
Opening (CW). Useful lift is around 9–10 mm with a clean profile and peak velocities in the low tens of mm rad^{-1} .

Figure 4.34: Desmodromic exhaust — opening cam (CW).



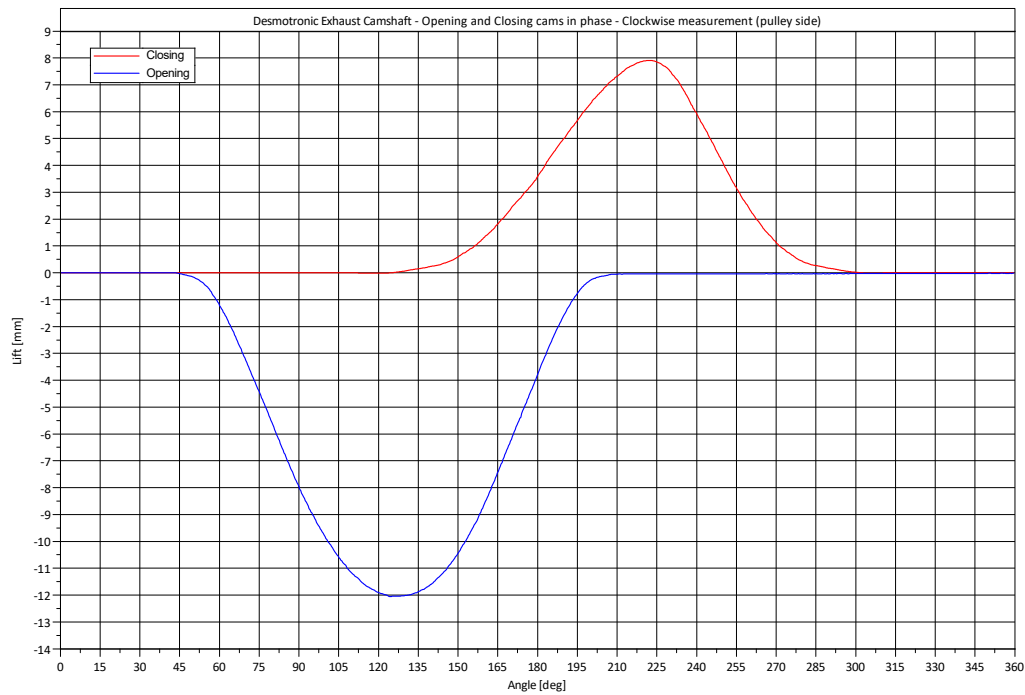
Closing (CW). Closing again reduces velocity clearly near seat, indicating controlled landing; velocity/acceleration magnitudes are comparable to desmodromic intake.

Figure 4.35: Desmodromic exhaust — closing cam (CW).



Opening and closing in phase. The overlay shows continuous control and a near-zero closing velocity at seating.

Figure 4.36: Desmodromic exhaust — opening and closing in phase (overlay).



Cross-read for the 9A1 adaptation

- **Amplitude compatibility.** Desmodromic peak lifts (13–14 mm intake; 9–10 mm exhaust) align with 9A1 levels, avoiding a lift-limited breathing penalty.
- **Low seating speed.** Both desmodromic closing profiles slow consistently near zero lift, supporting controlled seating at the target speed.
- **Smooth derivatives.** Velocity and acceleration traces remain continuous (no cusps), implying moderated jerk and a lower risk of structural excitation in the carrier/shafts.
- **Angular normalisation.** For direct overlays later, normalise crank angle to a common reference (e.g., 0° at opening @ 1 mm lift) and resample at uniform angular resolution.

Limitations and data handling

Peaks above are read from plotted scales. For quantitative use in dynamics/FEA, export lift, velocity and acceleration as arrays at uniform crank resolution; apply only minimal smoothing if needed and preserve true extrema.

4.7.1 Results

Notes on accuracy and consistency

Angle resolution is approximately 0.35° per count (1 024 counts per cam rev with quadrature decoding). The displacement scale is set from the probe calibration (mm per volt) and checked with gauge blocks over a short range. Because the rig is quasi-static and contact-based, repeatability is governed mainly by probe seating, preload and angle interpolation; averaging and consistent datum setting keep the spread small for the purposes of comparative lift profiling [81, 83]. The same rig, sensor settings and processing steps are used for both the desmodromic and Porsche camshafts so that differences in opening phasing, peak lift and closing flank can be attributed to the hardware rather than the method [3, 6].

4.8 Conclusions

Chapter 4 established the measurement and digitisation foundations used in the remainder of the thesis. First, the head and donor components were captured with a dual route—tactile CMM for controlling features and optical scanning for global context—and tied to a common datum frame. The resulting capability and uncertainty envelope were stated explicitly: CMM routines deliver order-of-magnitude ± 0.008 mm (95 %) on linear KPIs, while large best-fit distances from the scanner sit around ± 0.14 mm (95 %). This supports the decision to verify critical, machined geometry on the CMM and to reserve the mesh for clearance mapping and qualitative surfaces, in line with ISO 10360-2, VDI/VDE 2634-2 and the ISO “GUM” approach to uncertainty [79–81].

Second, a scan-to-CAD pathway was exercised across several platforms, with most progress achieved in Autodesk Fusion 360 using the *Mesh Converter* in *Facet*, *Prismatic* and *Organic* modes. Feature-level conversions around bearing seats, datum faces and covers were reliable, but a single, watertight solid for the full assembly proved impractical within available face-count and memory limits; this reflects the inherent trade-off between file size and geometric fidelity for large, mixed free-form and tight-tolerance parts [115, 117, 118]. The meshes therefore remain the authoritative reference for context, with CMM features providing numeric control.

Third, a bench was built to acquire quasi-static lift profiles for both camshafts. The cam is supported on rigid pedestals, driven by hand, and instrumented with a quadrature encoder for absolute cam angle and a contact displacement probe for lift; angle–lift data are synchronised, resampled to a uniform grid and exported to **UniPlot** for comparison and figures. The method yields repeatable opening/closing phasing and peak-lift identification at fine angular resolution, adequate for the intended geometric comparison of profiles [3, 6, 122, 123].

The chapter has, therefore, delivered: (i) a traceable measurement framework with stated uncertainty; (ii) cleaned, aligned and sectioned meshes plus partial, local BRep reconstructions; and (iii) lift curves for the desmodromic and Porsche camshafts on a common processing route. Its principal limitations are also clear: no full-part parametric solid has yet been produced from the mesh, and the lift curves are quasi-static (no dynamic inertial or oil-film effects are inferred from these data). These limits are acceptable at this stage but motivate the next steps—segmented, tolerance-driven reverse engineering, short GR&R and SPC checks on the controlling CMM routine, and progression to structural/tribological validation of the desmodromic layout in Chapter 5 and the test plan outlined later [3, 6, 83].

Chapter 5

Concept Development

Having validated the basic concept of the springless desmodromic valve actuation system, the design process entered an iterative refinement phase. In this chapter, six iterations of the desmodromic housing and associated components are presented. The term "housing" here refers to the entire assembly adapting the desmodromic mechanism to the Porsche GT3 RS engine - including the support structure and the integrated rocker arms. Each iteration focused on specific improvements or tests: addressing the shortcomings identified in the initial prototype, optimizing certain parameters, and enhancing overall performance and reliability. For each iteration, we describe what was changed or tested, the engineering rationale behind those changes, and the outcomes or lessons learned.

5.1 First Iteration

The first design shows the first embodiment of the concept exercised on a purpose-built bench rig. It comprises a prismatic aluminium cradle with a rectangular inspection window, two removable side-cheeks carrying cam templates, a central cartridge that hosts the paired rockers, and two 12 mm locating pillars used to clamp and datum the assembly on the original Porsche head. The closing and opening rockers are mounted on parallel shafts captured in the cartridge.

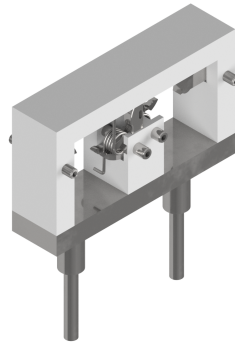


Figure 5.1: Iteration 1 – Single-Bay Bench Mock-up.

Only elements visible in the render are described here. No valve models are shown. This omission is intentional: the valves are the original engine components and the measured geometry adopted in the mock-up preserves the stock valve axis and tip height, so the original valves and related hardware can be obtained without redesign.

At this stage, the mock-up demonstrates: (i) a compact bridge housing concept with a rectangular window that provides tool access to pivots and fixings; (ii) a central cartridge that constrains the rocker pair on parallel pins; (iii) a modular 50.00 mm layout to expedite repeatable adjustments; and (iv) a base supported by two pillars to clamp and datum the unit on a plate or surrogate head surface. No quantitative claims are made here about forces, contact timing or dynamic behaviour; the figure simply documents the baseline physical arrangement and the adjustment scheme observable in the model.

Engineering Rationale: This iteration served as a proof of concept. The rationale was to change as little as possible from known references: it used the unmodified measured geometry of the head and largely mirrored Ducati's desmodromic geometry scaled to the Porsche valve. By doing so, we established a starting point whose performance could be evaluated relative to expectations from existing systems.

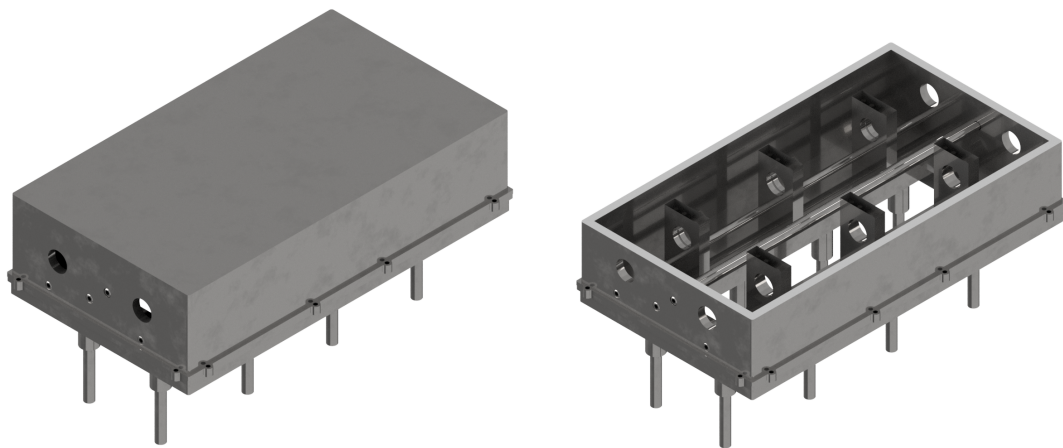
Lessons Learned:

- This iteration's bridge window and central cartridge were effective for visibility and quick kinematic adjustments. The next step for it is to perform a sealed, full-length housing with a cover and perimeter flange;
- This module works for a single pair of valves, however, the full engine is a 3 cylinder engine with 12 valves, so introducing repeatability is crucial to simplify machining, inspection and assembly jigs that will give a more realistic look into a full installation;

- In this iteration the valves are not shown because they remain the original parts, adopted measurements allow their direct reuse. Keeping them implicit at this stage keeps attention on the housing and shaft supports without conflating the work with components that are unchanged.

5.2 Second Iteration

The second iteration evolved from a single-bay bench rig to a closed, bank-length housing made from a baseplate and a deep cover. With the cover fitted (Figure 5.2), the assembly is a rectangular enclosure fixed along a continuous perimeter flange using an even pattern of small fasteners. Two large circular through-holes and several smaller holes are visible on the end face; these align with internal features shown in the uncovered views and indicate provision for two parallel rotating/axial elements to pass through the housing, while the smaller holes appear to serve for ancillary fixings or dowels. A row of vertical mounting pillars/studs projects below the baseplate along both long edges, providing a straightforward means to support or clamp the unit to a fixture or surrogate head surface.



(a) Cover fitted – end-face pass-throughs and flange pattern. (b) Cover removed – three bays with split saddles for twin shafts.

Figure 5.2: Iteration 2 – transition from single-bay rig to an enclosed bank-length carrier with repeatable split supports.

With the cover removed, the baseplate presents three repeated bays. Each bay contains two opposed half-cylindrical saddles arranged across the window—giving six saddles in total along the length. The saddles are mirrored about the central opening and are machined with a regular

scalloped relief, consistent with split-support seats for two parallel shafts that would run the full housing length. The internal geometry is deliberately simple and symmetric: upright blocks carry the saddles; the web between bays stays planar; and the perimeter flange contains clearance holes at a steady pitch, which will aid sealing and assembly. Compared with Iteration 1, the internal cartridge and 10.00 mm grid are no longer present; the adjustment concept is replaced here by fixed seats and a repeatable bay pattern.

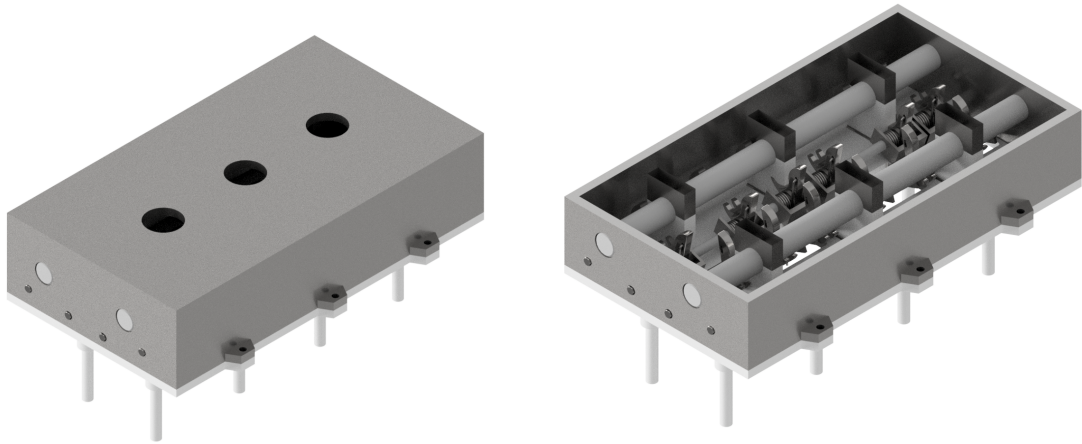
Engineering Rationale: The second iteration consolidates the open, single-bay mock-up into an enclosed, bank-length carrier with a baseplate, a deep cover on a continuous perimeter flange, and three repeated bays each containing a pair of half-cylindrical saddles; this fixes two straight, parallel shaft axes across the full span, replaces ad-hoc pivot adjustment with a measurable alignment scheme, and prepares realistic load paths. The cover establishes a closed section for stiffness and a practical bolt pitch for future gasket design, while the end-face through-holes provide coaxial access for shafts, mandrels and any end-mounted timing hardware, simplifying metrology (run-out, concentricity) and assembly sequencing in a way consistent with production valvetrain carriers. Retained underside pillars keep the prototype bench-mountable for independent straightness checks, and the valves remain intentionally unmodelled because the adopted geometry preserves the original valve axes and tip heights, allowing continued use of OEM valves and hardware without redesign.

Lessons Learned:

- First, shifting from an adjustable cartridge to fixed, repeated saddles reframes the problem from “where to place pivots” to “how to hold two precise, parallel bearing lines”. This makes the tolerance strategy explicit: baseplate flatness, bore concentricity in each split saddle, bay-to-bay parallelism and positive re-location of the cover (dowels or controlled fastener pairs) now dominate. In short, adjustability has been purposefully traded for bank-level repeatability and inspectability;
- Second, enclosing the structure clarifies stiffness and sealing. The perimeter-flange/cover creates a closed section that is inherently more resistant to out-of-plane bending than the open window of Iteration 1, and it reveals where local ribs or cross-ties between opposed saddles would most effectively reduce deflection under rocker loads;
- Third, the three-bay repetition demonstrates manufacturability and assembly discipline: a single bay cell can be tooled, gauged and then pattern-copied along the bank, simplifying machining programs, CMM routines and torque sequences; however, losing the modular grid of Iteration 1 means the datum scheme, inspection gauges and reassembly controls must now carry the tuning burden.

5.3 Third Iteration

Iteration 3 advances the enclosed housing into a fully populated carrier with optimised dimensions: the cover is retained (now with 3 circular access ports on the roof) and the baseplate carries two continuous, parallel shafts per bank (supposed to represent the camshafts), each seated in the split saddles introduced in the previous iteration and clamped by simple caps. Repeating rocker modules are now installed between the camshafts along the full length; each module is built around two through-shafts and a compact link set. The underside pillars keep the unit bench-mountable so the two shaft axes can be qualified independently of the head before any engine trials, which is standard practice for long, split-bearing carriers.



(a) Bank-level overview with twin continuous shafts and (b) Roof with access ports – closed section and service access modules installed.



(c) Close-up of rocker module showing integrated helical return spring.

Figure 5.3: Iteration 3 – populated carrier: repeating rocker modules between twin shafts and integrated return springs; roof ports for access and verification.

This iteration introduces a helical return spring within each rocker module. As shown in the close-up, the spring is captured between cheek plates and reacts against a collar on the rocker core; it returns the rocker to the closed position and, consequently, returns the valve. The use of such light auxiliary springs at module level is a common prototyping aid: it stabilises components during assembly, reduces the risk of mis-seating when the cover is lowered, and allows repeatable, low-force motions while gauging clearances to surrogate “cam” tubes and strap caps.

Engineering Rationale: This iteration populates the bank-length carrier with two continuous, parallel shaft lines and repeated rocker modules, each module incorporating a helical return spring that restores the rocker to the closed position and, consequently, returns the valve. The closed cover (with three roof access ports) establishes a realistic sealing boundary and a stiffer closed section than the open cradle used earlier, keeping the assembly bench-mountable so straightness and parallelism of the shaft pair can be qualified with mandrels before any head-mounted trials. Cam-envelope proxies (the long cylindrical dummies) are retained to verify clearances to straps, cheeks and cover without committing to lobe geometry. The result is a bank-level alignment and packaging platform on which spring pre-load, installed length and rate can be specified to guarantee seating.

Lessons learned:

- Moving to continuous shafts with strap caps made the assembly sequence and metrology unambiguous: run-out can be checked end-to-end with mandrels, concentricity can be checked at each split saddle, and the roof ports give repeatable torque access along the bank.
- Embedding the helical spring within each module proved valuable for deterministic return and stable bench handling, but it introduces the problem of fixing in place this small assembly, which could be problematic especially facing high vibrations;
- The enclosed architecture clarified gasket land width and confirmed that a closed section markedly improves bending stiffness compared with the open rig, which will aid contact stability at speed. At the same time, keeping the original valves out of scope remained helpful: with valve geometry unchanged, effort stays focused on the modular structure.

5.4 Fourth Iteration

The fourth iteration keeps the third iteration bank-length carrier, cover and twin continuous shafts but changes the camshaft supports to angled blocks machined into the lid. The supports are now neglected, focusing on only improving the camshafts supports, while all other features and clearances remain as before. The rocker-shaft arrangement is unchanged; valves are still not modelled.

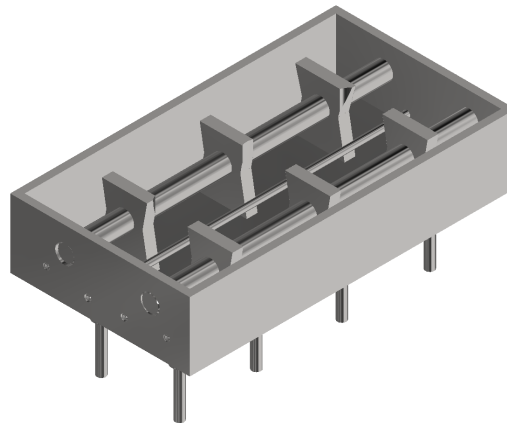


Figure 5.4: Iteration 4 – cover-integrated (wedge) supports for the shaft line, keeping the established carrier envelope.

Engineering Rationale: Moving to angled (wedge) supports on the cover improves the local load path from cam/shaft reaction into the lid ribs, increases tool clearance around the seats, and reduces mass compared with tall rectangular uprights, while maintaining the centre distance and the previously verified roof clearances. The inclined seats also facilitate oil shedding away from the bearing split and allow a generous approach for mandrels through the roof apertures during bench alignment. No other geometry is altered in this iteration, so manufacturability and the assembly sequence remain as proven in Iteration 3.

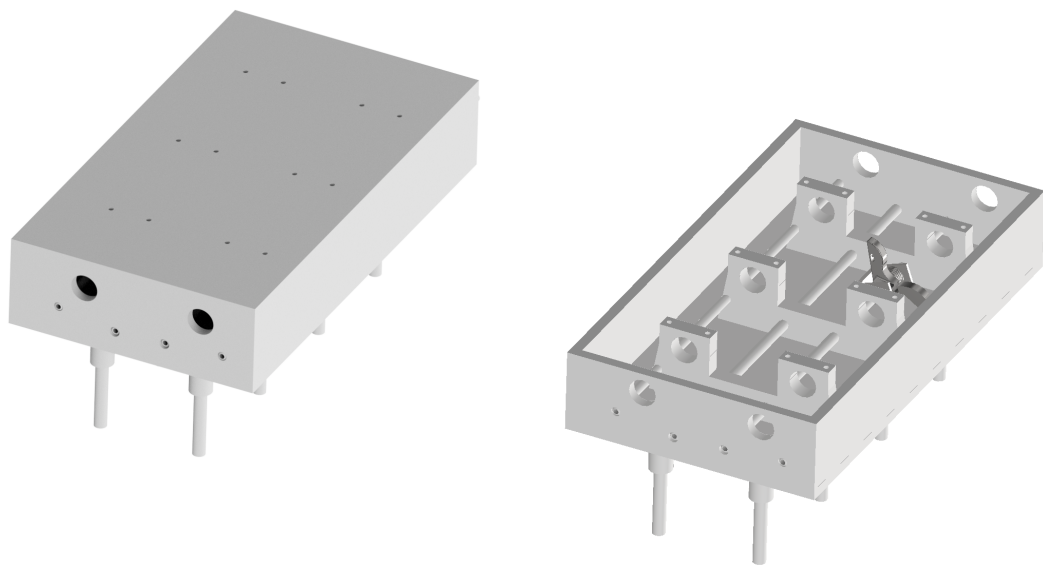
Lessons Learned:

- Although the angled seats behave well as individual supports, alignment of the rocker shafts remains a concern: at present the rocker shafts are located only by the lid, so any off-axis assembly force or operational side-load from the rocker/spring module can introduce shaft bow across the span. With long, split-bearing carriers this risk is non-negligible; without a base support or dowelled, ladder-type clamping, the shaft straightness depends too heavily on lid stiffness and fastener torque patterns;
- The angled cam supports validate the packaging choice but emphasise the need to couple stiffness with sealing: gasket land width and bolt pitch around each roof opening must be confirmed so that the inclined geometry does not compromise the future oil control strategy. These checks can be completed in parallel with the shaft-support redesign into a ladder arrangement.

5.5 Fifth Iteration

In this iteration the carrier is reworked internally to support the two rocker camshafts on a ladder-frame: each shaft now passes through a row of opposed pedestals and through-tubes that tie the side walls, creating continuous, straight bearing lines across the bank (visible as the long bars

through the bosses). The aim is to stiffen the span, reduce shaft bow and make alignment measurable with end mandrels after assembly, in line with good practice for long, split-bearing carriers. A second change is the introduction of bolt-on “small-rocker” cartridges: each secondary rocker is mounted to a compact block that locates on the mid rail and is retained by screws; this allows pre-assembled modules to be placed between the shaft lines before the cover is fitted. The remainder of the envelope—end pass-throughs, cover, and bench pillars—follows Iteration 3.



(a) Open view – opposed bosses/cross-tubes closing the shaft line through depth.

(b) Cover fitted – closed section and perimeter sealing rails visible.

Figure 5.5: Iteration 5 – Ladder-Frame Carrier.

Engineering Rationale: The ladder-frame concept answers a weakness exposed by lid-only supports: with long shafts carrying rockers and return springs, lateral components of load can introduce shaft bow if the bearing line is not closed through the depth of the carrier. By taking each shaft through opposed bosses and cross-tubes, reaction forces are carried directly to both side walls, increasing the effective second moment of area of the assembly. This makes alignment measurable as well as stronger. The bolt-on small-rocker cartridges aim to decouple delicate assembly tasks from the main build. In principle, each cartridge can be fixtured on the bench, its helical return spring set to the specified installed length and pre-load, and the module gauged before being inserted into the housing. That approach promises repeatability and reduced handling risk, and it aligns with modular assembly methods used when complex valvetrain sub-assemblies must be installed in a constrained envelope.

Lessons learned:

- The bolt-on small-rocker cartridges ease bench assembly but are too bulky once the twin shafts and cover are installed; fasteners/adjusters are obscured and routine tasks would force partial strip-down. A slimmer retention concept (e.g., pinned shoes, keyed dovetail or true split-shoes removable with shafts in situ) is required;
- The present cross-tubes and deep bosses make the housing difficult to machine (trapped bores, poor tool access, problematic metrology). Repartition into machine-friendly elements—open saddles plus separate caps, dowelled and bolted—so bores can be line-bored/honed and inspected in open setups.

5.6 Transition to the Final Design

With the proof of concept completed and the architecture established, the design was reframed from a manufacturing-first perspective by starting with a solid billet that matched the maximum permissible envelope and removing only the material strictly required for future components and clearances. A rectangular block was wrapped around the measured master geometry and global datums were fixed on the base and end faces so every removal referenced a stable coordinate frame. Material was then subtracted as prismatic, tool-friendly cut-outs: longitudinal galleries for the camshafts, a matrix of bay apertures for rocker modules and service access, and end-face pass-throughs for shafts and fixings, with complementary recesses safeguarding the combustion-chamber roof and valve-guide zones; fillets and blends were intentionally deferred until ribs and sealing lands were validated.

This subtractive “block-out” method preserved stiffness and sealing by default—continuous perimeter flanges, cover rails and deep side walls remain intact while voids are carved only where motion is needed—whilst keeping manufacturability visible by restricting removals to operations executable from sensible setups. Because the billet is wrapped around the measured master, integration with the existing head is guaranteed and the original valves and hardware are retained; only the superstructure (cams, rocker–spring modules, carrier and cover) is shaped within a stiff, closed section.

5.6.1 Converting the block-out to machinable features

Following the subtractive “block-out” study, the reserved volumes were converted into machinable features by extrusion along sensible tool axes. In practice, each prismatic pocket that will become a through-hole, gallery or window was extended normal to the relevant datum face (roof, base or end face) until it intersected the exterior surface, creating open features that are drillable or millable. This step transforms abstract keep-outs into a process-capable set of operations (e.g., drilling, counter-boring, interpolated pockets), ensures chip evacuation and metrology access [124].

In parallel, a cylinder-bore extrusion was retained in the head underside to represent the true volumetric claim of the combustion chambers. Keeping these three large cylindrical placeholders visible forces all subsequent carrier features to be validated against realistic chamber keep-outs. The images show this intermediate outcome clearly: roof-side galleries and windows extended to open faces for manufacturability, and the underside populated with three cylindrical extrusions that convey the occupied envelope and datum positions of the bores.

5.6.2 Final Assembly

After the block-out and extrusion stages, a full bank assembly was created that brings together the desmodromic superstructure and the original valves within the verified envelope. The rectangular roof windows and end-face pass-throughs remain, so assembly and inspection access are preserved while the closed section provides bending stiffness consistent with a production-style carrier. Packaging is therefore proven: the components coexist without clashes to the chamber keep-outs and the service apertures appear workable.

In plan view, the circular outlines of the three bores are visible beneath the housing and each bay shows four rocker stations arranged symmetrically about the valve axes. The rockers, spring packs and cheeks sit on the lower shaft, while the upper shaft line carries the mating members; both shafts run through the bay on straight axes with strap-cap rhythms. The arrangement demonstrates repeatability: the same clearances are maintained at each bay, and there is margin to the nearby bore volumes and sidewalls, suggesting the earlier subtractive sizing was appropriately conservative.

The single unresolved point is the fixation of each rocker on its shaft. As modelled, the rockers appear to pivot on plain bushes with minimal axial definition; at high speed this risks axial walk, rattle or fretting of thrust faces, and complicates repeatable re-assembly after service. The design therefore needs an explicit axial-retention and end-float scheme per rocker compatible with oil strategy.

Axial retention of the rockers: conceptual options and verification

With geometry and kinematics proven, the remaining practical gap is the *positive axial retention and location* of each rocker on its shaft. Without a defined scheme, axial walk and micro-slip at thrust faces can lead to rattle and fretting damage at high speed, compromising repeatability after service. Fretting and false brinelling under small-amplitude oscillation are well documented in oscillating bearing/thrust interfaces, and dynamic load ripple from the valvetrain can exacerbate these mechanisms [12, 125–127].

Option A — Grooved shaft with retaining rings (double circlips), hardened thrust washers, light axial preload A widely used architecture is a shaft with precision grooves, each rocker axially bracketed by *two* retaining rings and ground thrust washers; a light spring or compliant spacer at bay level removes rattle while preserving a measured end-play window. Design attention must focus on groove root geometry and web spacing, because snap-ring grooves act as fatigue stress concentrators; increasing root radius and adopting stress-relief features reduces K_t and improves life [128]. Washer materials/coatings (e.g., nitrocarburised steels with low-friction top-layers) can lower friction and wear in thrust interfaces [129]. This concept keeps full serviceability and individual rocker replacement, at the cost of careful groove design and inspection. [128, 129].

Option B — Shouldered shaft with fixed spacers (no grooves), end capture Where shaft bending strength and fatigue are paramount, a turned shoulder locates each rocker stack with ground spacers and thrust washers; the assembly is captured at the ends by a cap or split-collar. Eliminating snap-ring grooves removes the dominant geometric stress raiser, trading easy re-spacing for deterministic axial locations—well suited to a frozen layout or “ladder” carriers. Dynamic loads from the desmodromic train still require stiffness and end-play control, but the shaft’s fatigue risk is reduced relative to grooved designs [12, 128].

Option C — Flanged (or thrust-faced) bush solution with axial spacers A third path integrates the thrust function into the bushing (flanged bush or bonded thrust-face) so axial reaction occurs on engineered bearing surfaces, with axial spacers to set end-play. This concentrates tribology on a purpose-designed sliding pair and avoids discrete washers. Oscillating bronze/steel journal and thrust contacts at small amplitudes are tractable with suitable lubrication and finishes, but require film-robustness checks and monitoring of friction hysteresis under oscillation [130, 131].

Verification protocol (summary) For any selected concept: (i) set and verify a measured end-play window per rocker (e.g., 0.05–0.10 mm) at build and post-endurance; (ii) blue and inspect thrust faces after representative oscillatory endurance to screen for fretting scars/false brinelling; (iii) modal/stiffness checks on the carrier/shaft line to limit dynamic deflection; and (iv) confirm that thrust and pivot interfaces remain within acceptable contact-stress/film conditions across the 10 500 rpm duty [12, 126, 127].

Recommendation Prototype **Option A** for maximum serviceability, implementing generous groove radii and hardened/low-friction thrust washers, and down-select to **Option B** if shaft fatigue margins prove tight. **Option C** is attractive where integrated bushes align with the manufacturing plan, provided oscillatory lubrication performance is validated *in rig* [128, 130, 131].

Expected structural hotspots (for FEA) Hotspots are anticipated at: (i) shaft spans at mid-bay (bending under lobe torque ripple), (ii) rocker shoe tips and pad/roller contacts (local Hertz

stress), (iii) rocker thrust faces (axial load transfer), and (iv) carrier bolt lands and bridge sections (local stiffness). Targets are: peak stress below fatigue limits with suitable safety factor; shaft-line deflection < 0.02 mm across the span to preserve contact stability; mode placement to avoid excitation by the lobe-torque spectrum [12, 104].

Chapter 6

Conclusions

This thesis demonstrates the adaptation of a springless, desmodromic valvetrain to the Porsche 9A1 cylinder head while retaining the original valves and a production-style carrier. A measurement → parametric modelling → concept iteration → manufacturability workflow delivered a geometric proof of concept with coherent service access and a machinable carrier (“block-out” with extrusions). Envelope checks and cam-to-cam separations confirm workable packaging. The main open item is rocker-on-shaft axial retention and end-float control.

6.1 System-level implications

Raising the safe engine speed target from 9 000 to 10 500 rpm provides headroom for higher specific power where breathing supports it. Eliminating valve springs reduces valvetrain parasitic losses and decouples seating control from spring dynamics, improving closing authority at high speed. NVH and durability outcomes depend on end-play stability, contact integrity, lubrication and shaft-line stiffness, which are now clearly framed for verification.

Measured BSFC deltas of order 1 % are plausible when spring work is removed and contact/lubrication losses remain controlled, in line with reported gains for flexible or pneumatic return systems [4, 10, 11].

6.2 Limitations

The present work establishes geometric and process feasibility only. Functional validation at 10 500 rpm has not yet been demonstrated. Reliability hinges on a positively located axial retention scheme, controlled end-play, and verified contact, lubrication and stiffness margins.

6.3 Future work

Near-term (component and rig level)

1. **Axial retention design freeze.** Select and dimension the concept; specify materials/coatings; define a measured end-play window per rocker.
2. **Structural and modal analysis (FEA).** Verify stress and deflection limits for carrier, covers and shafts; confirm mode separation across the operating range.
3. **Tribology and contacts.** Define closing-path loads, target pressures and film parameters; finalise surface finishes and oil-supply strategy.
4. **Kinematics and friction rig.** Map closure integrity, seating behaviour and parasitic losses versus speed; establish a baseline for projected BSFC effects.

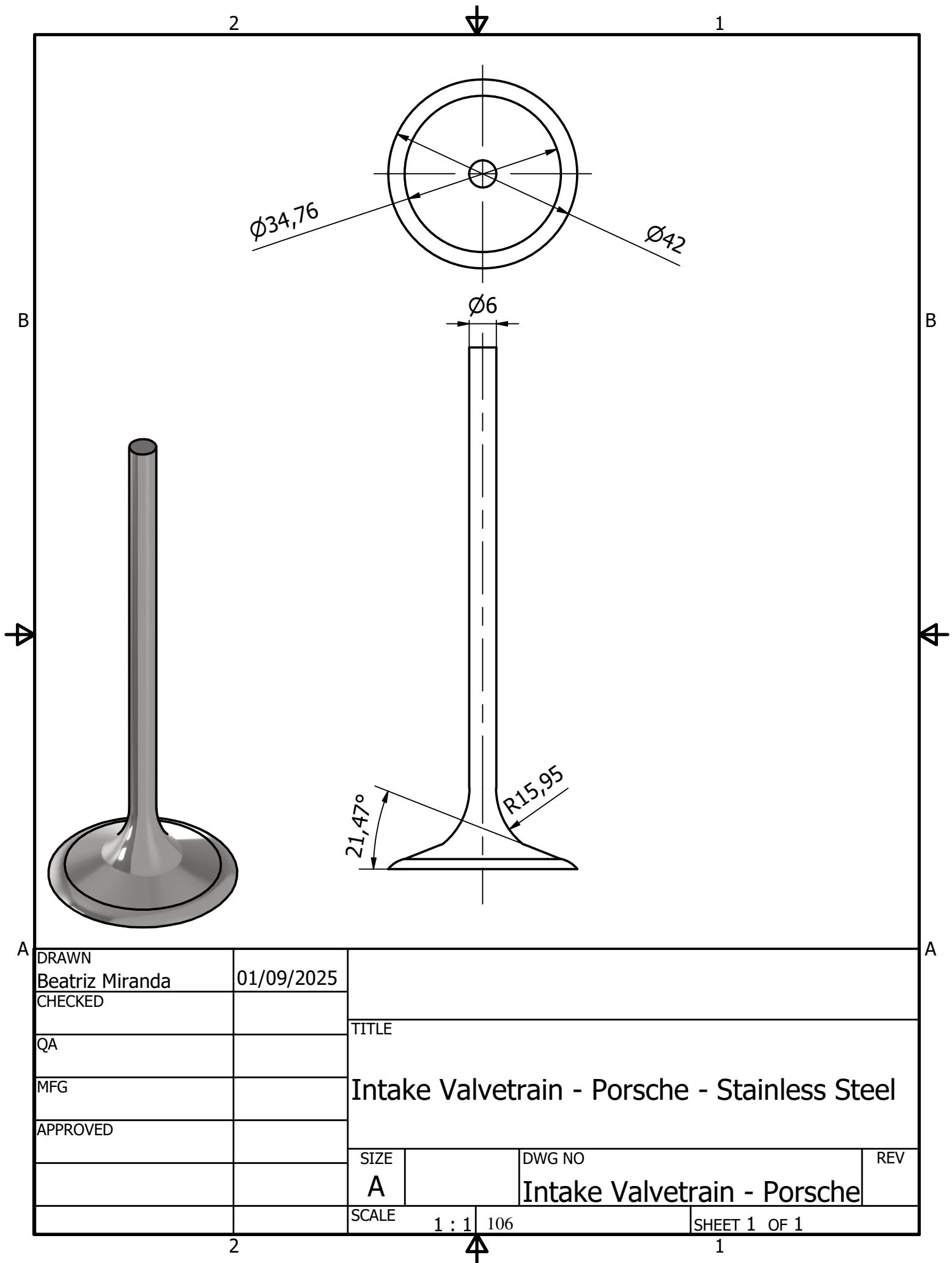
Medium-term (engine level)

1. **Engine dynamometer validation.** Hot start, speed sweep to 10 500 rpm, and steady points for fuel use, timing stability, NVH and leakage.
2. **Endurance and teardown.** High-speed cycling, inspection of thrust faces, shafts and bushes, and verification of end-play stability and rebuild repeatability.
3. **System metrics.** Quantify power, efficiency and NVH deltas versus baseline, and update the sustainability narrative with measured data.

Closing statement. A functional ambition has been converted into a reproducible engineering route. The next stage will close rocker retention and end-play, and progress through analysis, rig and engine testing to turn projections into evidence. This defines a clear pathway from geometric feasibility to functional validation at 10 500 rpm. The study paves the way for the first demonstrated automotive application of a desmodromic system in a production Porsche architecture, with potential impact on performance and sustainability.

Appendix A

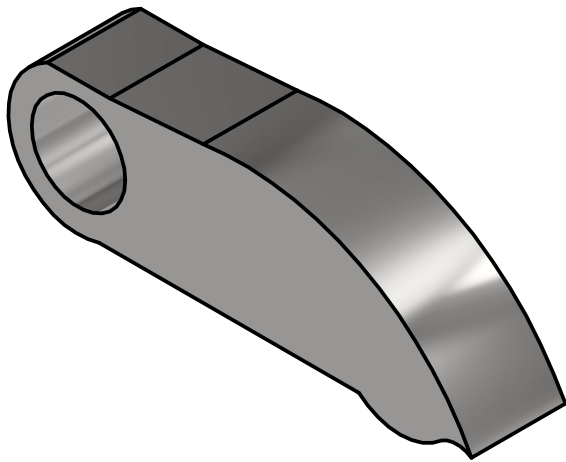
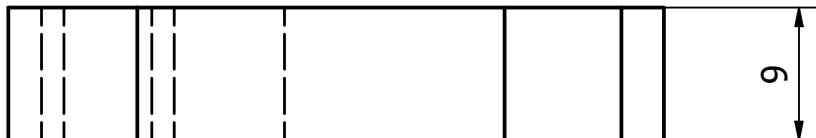
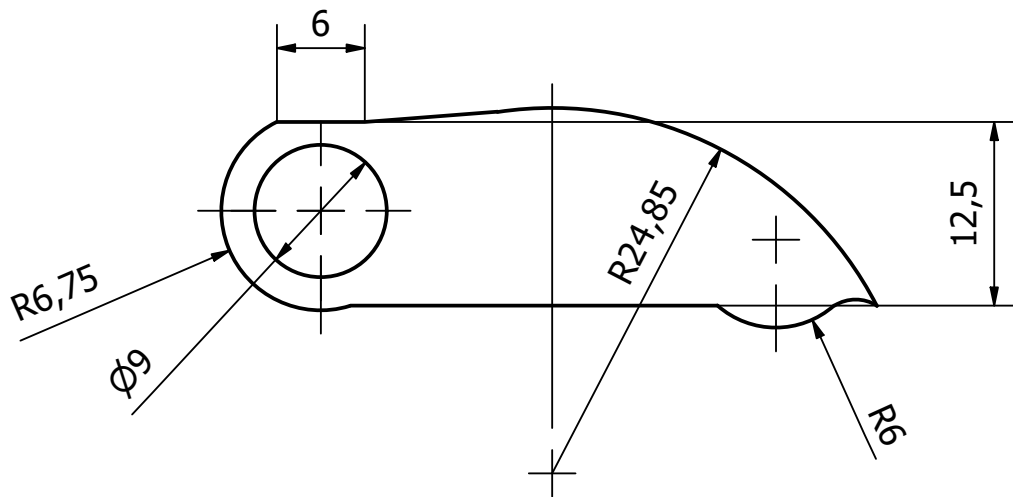
Technical Drawings (PDF)



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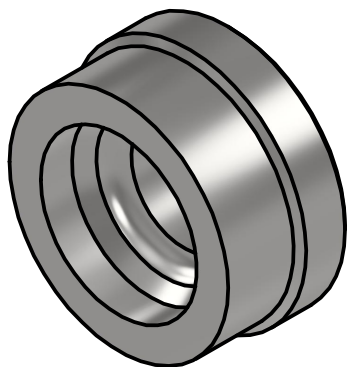
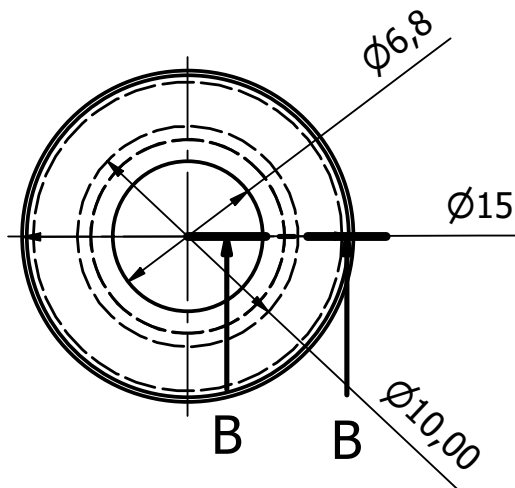
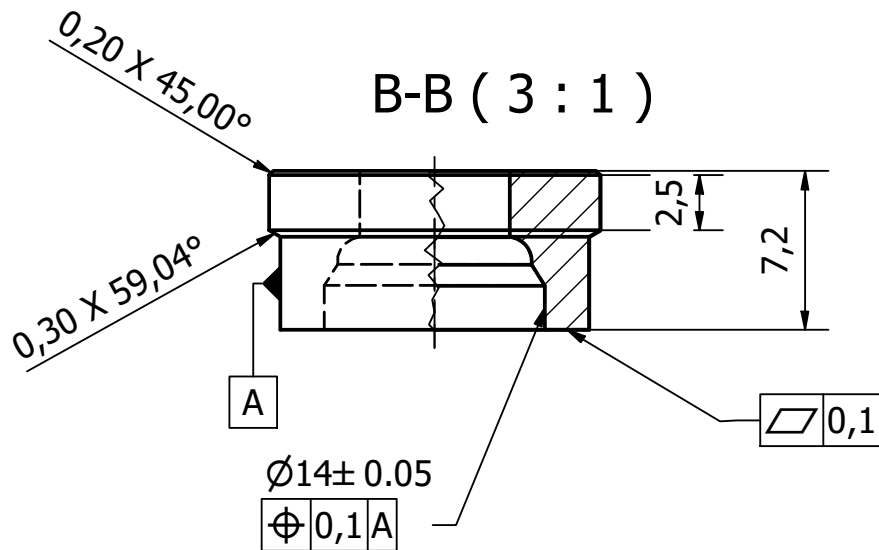
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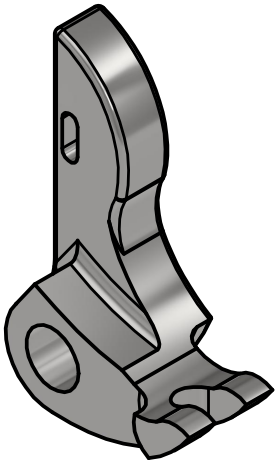
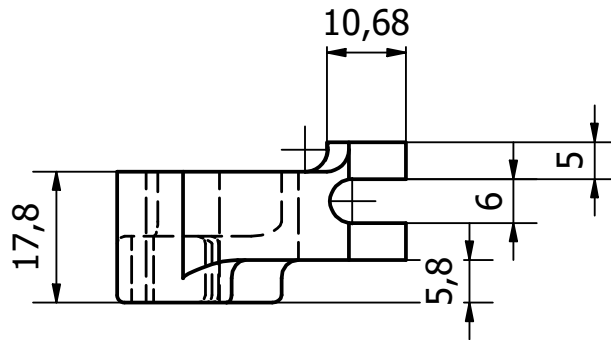
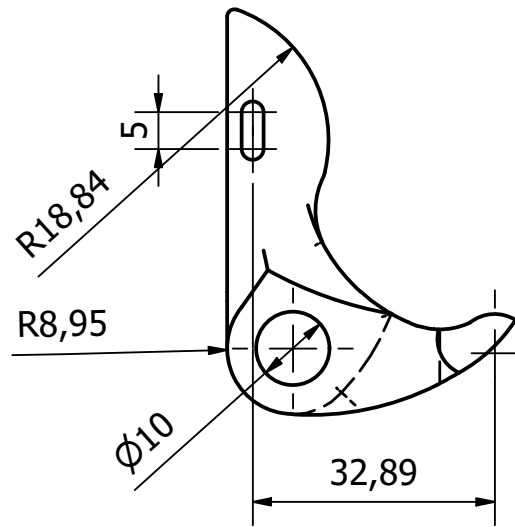
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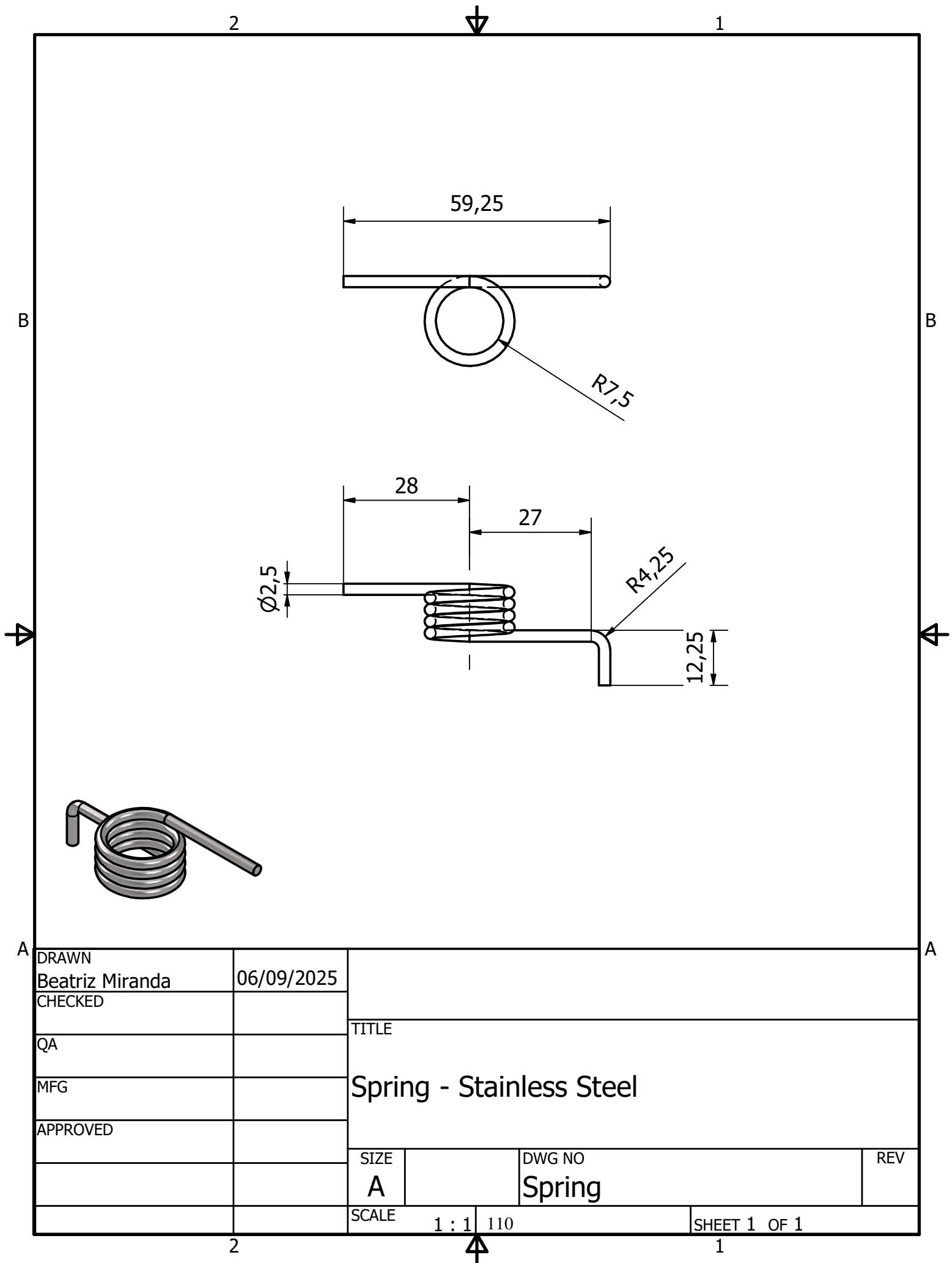
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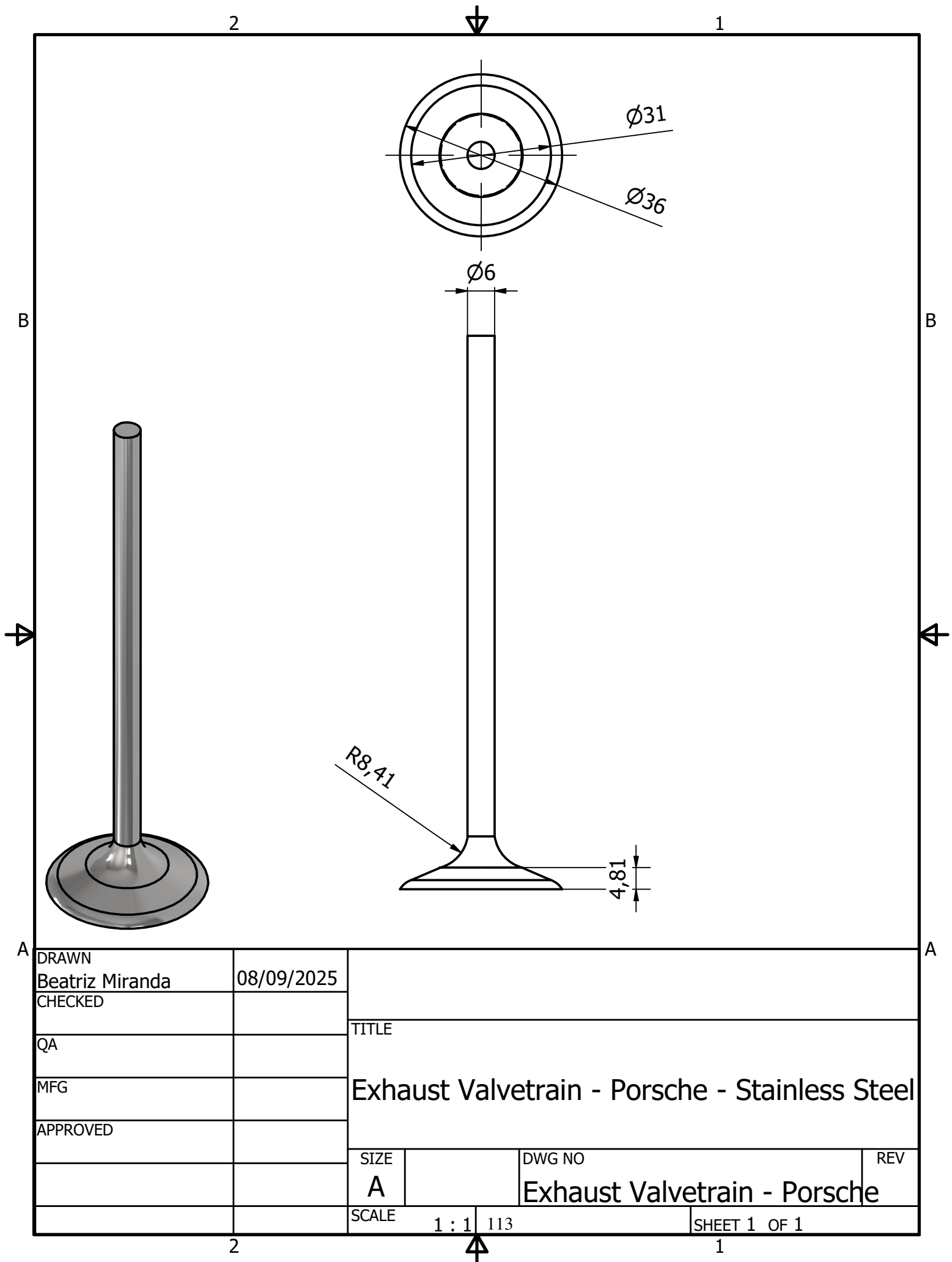
DRAWN	Beatriz Miranda	08/09/2025	TITLE Closing Rocker - Stainless Steel			
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			SIZE A	DWG NO Closing Rocker	REV	
			SCALE 1 : 1	109	SHEET 1 OF 1	

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