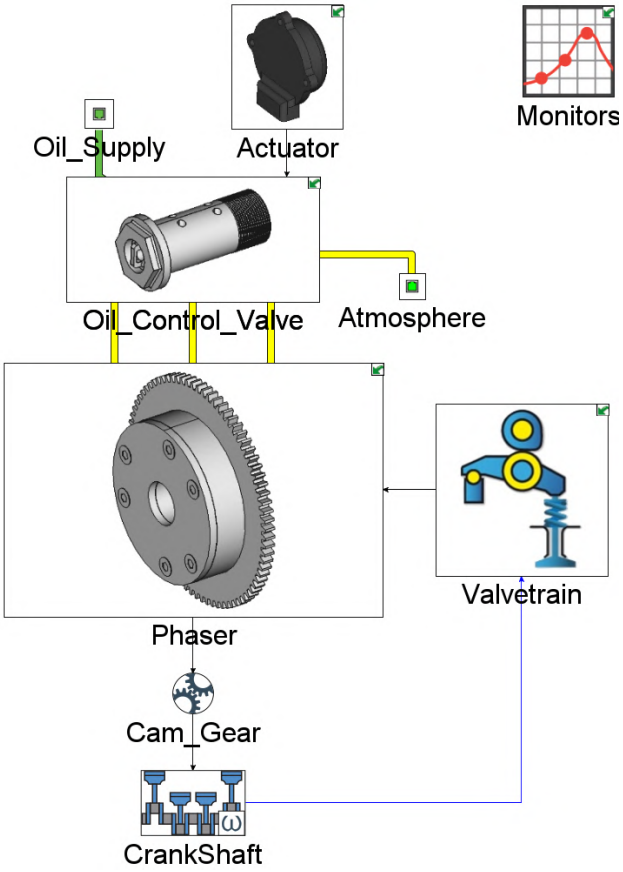
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Cam Phaser Simulation in GT-Suite

Master's Thesis in Mobility Engineering

SARATH SANKAR

DEPARTMENT OF MECHANICS AND MARITIME SCIENCES

CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2026
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Master's thesis 2026

Cam Phaser Simulation in GT-Suite

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UNIVERSITY OF TECHNOLOGY

Department of Mechanics and Maritime Sciences
Division of Transport, Energy and Environment
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Gothenburg, Sweden 2026

Cam Phaser Simulation in GT-Suite
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Cover: GT-SUITE model of the Cam Torque Actuated phaser system developed in this thesis, showing its hydraulic, mechanical, and electrical components.

Typeset in LaTeX
Printed by Chalmers Reproservice
Gothenburg, Sweden 2026

Cam Phaser Simulation in GT-Suite
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Abstract

Cam phasers enable variable valve timing by changing the angular position of the camshaft relative to the crankshaft. In heavy-duty diesel engines, cam phasing can support functions such as thermal management, engine braking, and potential replacement of certain conventional engine components. Understanding the internal behaviour of a cam phaser is therefore important for further system development. However, existing supplier models provide limited access to the internal hydraulic and mechanical behaviour of the system.

This thesis develops an in-house system-level model of a hydraulic Cam Torque Actuated vane phaser in GT-SUITE. The model combines the hydraulic, mechanical, and electrical domains of the system, with particular focus on the Oil Control Valve and the internal oil flow paths governing phaser operation.

The model is calibrated against test-cell measurements. The results show that the model captures the actuator and cam phasing behaviour observed in the measurements, although some deviation remain. Beyond reproducing measurable system responses, the physics-based model enables investigation of internal quantities and oil flow behaviour that cannot be directly measured in the test setup. The developed model therefore provides a basis for improved understanding and further development of the cam phaser system.

Keywords: Cam phaser, GT-SUITE, Variable Valve Timing, Oil Control Valve, System simulation.

Acknowledgements

Acknowledgements
I would like to express my sincere gratitude to everyone who contributed to this thesis and made the whole experience soo memorable. First, I would like to thank my supervisor, Nishchay Sharma, for the incredible amount of time and effort he invested in this work. From our countless morning meetings to the numerous discussions throughout the project, he was almost like a thesis partner to me. His guidance, encouragement, and constant support were invaluable, and I genuinely could not have completed this thesis without him. I am grateful to my examiner, Petter Dahlander, for taking the time to review my work at different stages of the thesis and for providing valuable feedback. Special thanks go to my co-supervisors, Reza Babaahmadi and Hampus Lindholm Bergman, for always being happy to help and for sharing their knowledge of the cam phaser system.
I would like to thank Markus Hellberg Mediaa, who conducted the tests that provided the measurement data for this thesis. Beyond the test data, his knowledge and perspectives contributed greatly to my understanding of the project, including the Black, Grey, and White Box modelling concept, among others. I am grateful to Johan Germundsson, whose previous thesis work provided a valuable foundation for me to build upon.
My sincere thanks go to Tomas Björnelund for providing me with this opportunity and along with all my colleagues at TGRPCEV, for creating such a welcoming and enjoyable working environment. I have learned a great deal from working alongside them, both technically and personally. I would like to thank Andrew Spencer, whose initial conversation following my enquiry about thesis opportunities, helped connect me with the engine valvetrain team and eventually led to this thesis.
Finally, I am deeply grateful to my family and friends for their endless encouragement and support. The countless phone calls with my family and conversations with my friends helped me through the challenging days and made the entire journey much easier. Looking back, having the opportunity to do my thesis in my dream field of core engine development has been one of the best decisions of my life. I have genuinely enjoyed every bit of the experience, and it is something I will always remember fondly.

Sarath Sankar, Södertälje, 2026

Disclosure of AI Use

AI tools were used to support the writing process, particularly to improve structure and grammar. The ideas, analysis, and conclusions presented in this thesis are the author's own. All AI-assisted material was reviewed by the author, who assumes full responsibility for the final content of the thesis.

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1

Introduction

This chapter provides a short background to the thesis, presents its aim, and explains the motivation behind developing the cam phaser model.

1.1 Background

How does a four-stroke diesel engine work?
To create fire, three elements are required: oxygen, heat, and fuel. A diesel engine creates fire, technically referred to as combustion, through its operating cycle:

1. **Intake:** Air is drawn into the engine, providing oxygen.
2. **Compression:** The air is compressed, increasing its pressure and temperature and providing the heat required for ignition.
3. **Power:** Fuel is injected into the hot compressed air, causing combustion. The expanding gases drive the piston downward and produce power.
4. **Exhaust:** The combustion gases are expelled from the cylinder, preparing it for the next cycle. The four-stroke cycle then repeats.

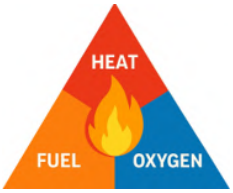


Figure 1.1: Fire Triangle [1]

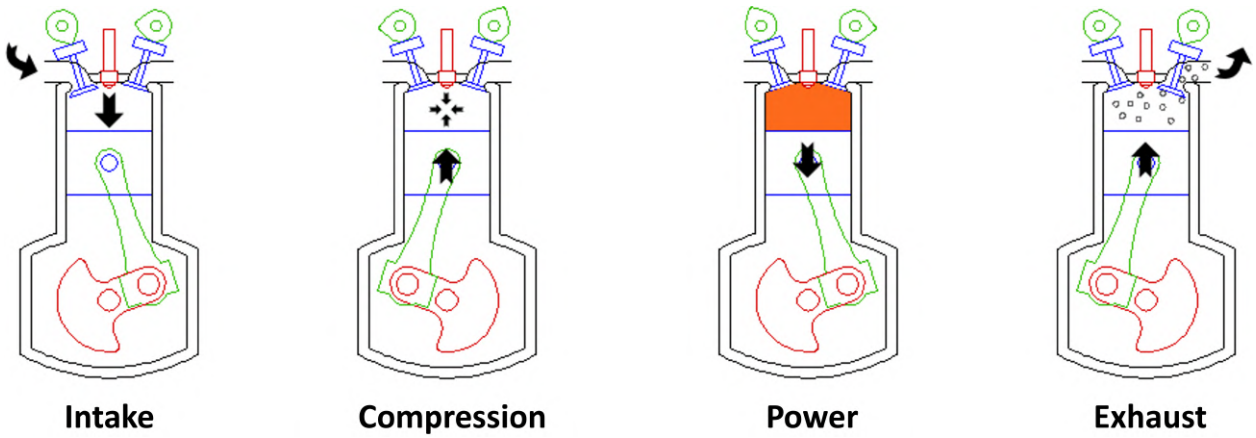


Figure 1.2: Diesel Engine [2]

What are camshaft and crankshaft in an engine?
Figure 1.3 shows a simplified single cylinder engine. At the top, the cam lobes sits on the camshaft, which actuates the valves below. At the bottom, the piston assembly and the crankshaft. If we draw an analogy between a human body and an internal combustion engine. Camshaft is the mechanical brain of the engine, which controls the inhalation and exhalation of the engine or technically called as gas exchange process by opening and closing the intake and exhaust valves. Crankshaft is the heart of the engine, which converts the reciprocating motion of the piston (combustion pressure) into rotational motion (engine output).

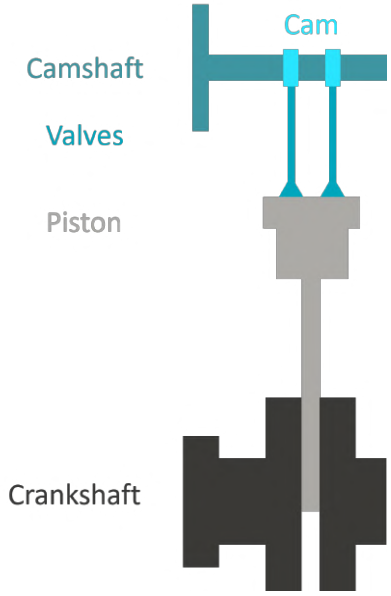


Figure 1.3: Single Cylinder Engine Schematic

What is a Valve Event?
Each inhalation and exhalation or which is technically called as a Valve event is described by Lift, Duration and Phase. Note, The profile of the cam is fixed, to add flexibility to the engine under different operating conditions, we use VVA (Variable Valve Actuation).

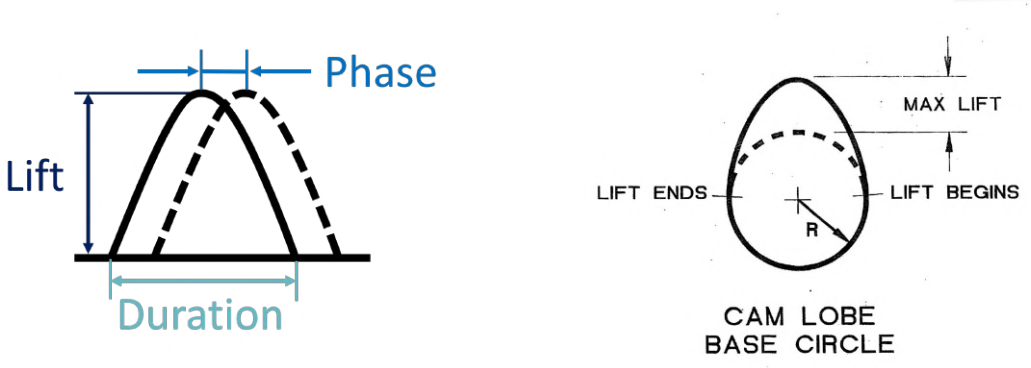


Figure 1.4: Valve event characteristics and cam lobe profile [4]

What is Variable Valve Actuation?

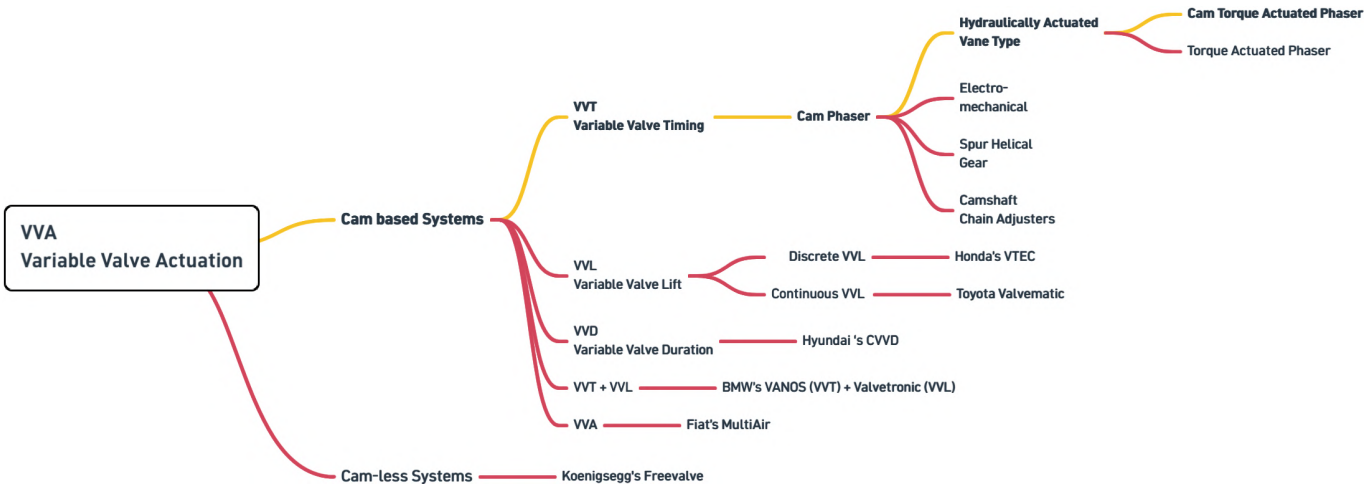
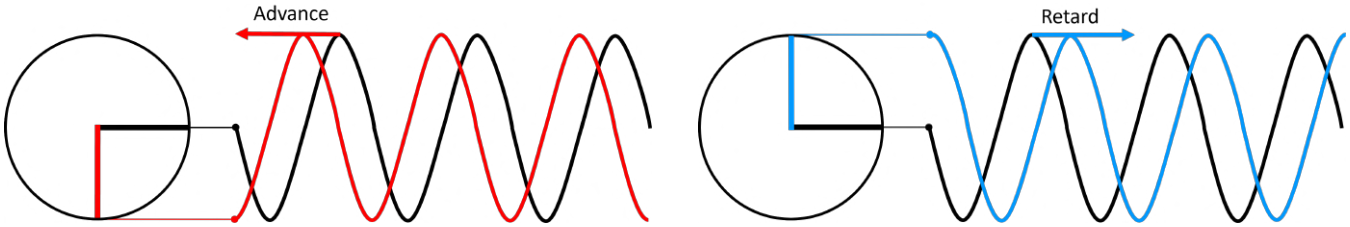


Figure 1.5: Classification of VVA technology (reproduced from DieselNet [3])

A fixed camshaft profile is a compromise across the engine's load and speed range. Variable Valve Actuation (VVA) technologies introduce flexibility by enabling variation of valve timing, duration, and/or lift [3]. While variable valve timing is widely used in spark-ignition engines, its role in diesel engines is more application-driven and closely linked to emission control and thermal management. Figure 1.5 illustrates the classification of Variable Valve Actuation technologies. VVA systems can be classified as camshaft based systems or camless systems. Camshaft based systems can be further classified according to their function: Variable valve timing, Variable valve lift, Variable valve duration and combinations of these. [3] [10].

What is a Cam Phaser?

As the name suggests, cam refers to the camshaft and phase refers to angular position. A cam phaser therefore changes the angular position (phasing) of the camshaft relative to the crankshaft. [3]. Phasing the camshaft relative to the crankshaft changes when the valves open and close with respect to piston position, thereby changing the valve timing.



(a) Advance **(b)** Retard

Figure 1.6: Camshaft and crankshaft angular positions represented as sinusoidal waves, with phase shift (phasing), either forward (advance) or backward (retard)

Why are cam phasers needed in a diesel engine?
Cam phaser serves three primary functions [7]

1. **Real−Time Thermal Management** Cam phasers regulate airflow through the engine by continuously and variably adjusting valve timing on the intake and exhaust camshafts. This allows the engine to achieve optimal airflow for performance and aftertreatment efficiency. For example, during low-load operation, the cam phaser can adjust the valve timing to reduce airflow and maintain higher exhaust temperatures, improving aftertreatment performance and fuel economy.
2. **Simpler Engine Design** Cam phasers can eliminate the need for components such as the intake throttle, exhaust brake and external EGR system, resulting in reduced engine complexity and enhanced durability.
3. **Engine Braking** When combined with a Compression Release Brake (CRB), which releases compressed air during the compression stroke to generate braking torque, cam phasers enable Variable Valve Braking (VVB), allowing for higher brake power and smooth brake response. VVB can also be used to reduce engine speed during gear shifting, enabling smoother upshifts.

These advantages become particularly relevant in heavy-duty diesel applications, where durability, braking capability, and aftertreatment efficiency are critical.

What need brought cam phaser to the HD engines now than before?
Conventional heavy-duty diesel engines have historically met emission regulations using high-pressure fuel injection (eg. Scania XPI system designed for maximum injection pressures of up to 2,400 bar [9]), turbocharging with EGR, and exhaust after-treatment systems (DOC/DPF/SCR)[8]. With VVA the dependence on some hardware (EGR, throttling, exhaust brake) can be reduced. The integration of cam phasers in heavy-duty diesel engines is therefore driven more by optimization and future tightening emission standards than by basic regulatory compliance.

Why is hydraulic cam phaser the choice of HD Diesel Engines?
Hydraulic cam phasers are suitable for heavy-duty diesel applications due to their compatibility with existing valvetrain architectures and lubrication systems. They use proven hydraulic technology, tolerate high loads and engine braking conditions in heavy-duty application, and offer a compact and cost-effective solution compared to fully variable or camless systems.

How can addition of cam phaser allow removal of intake throttle, EGR and Exhaust Brake?
Through manipulation of valve events, cam phasers can functionally replace or reduce reliance on certain conventional hardware components:

1. Intake throttle: Retarding intake closing reduces effective compression and fresh air → throttling effect [11].
2. EGR: Replacing external or cold EGR with internal or hot EGR by increasing valve overlap [11].
3. Exhaust brake: Optimising the engine braking with cam phaser enabling Variable Valve Braking providing superior braking performance with smooth response[11].

To better position hydraulic cam phasers within the broader context of valve train technologies, it is useful to briefly review the classification of variable valve actuation systems.

1.2 Aim

Develop an in-house GT-Suite model to improve understanding and development of the Cam Torque Actuated phaser system.

1.3 Why

The boxes in Figure 1.7 represents the cam phaser system. The figure shows the model from the supplier, which is a black box model, Input-output model with no knowledge of the internal system. With such a model, there is no possibility to investigate problems or to optimise the design, which as resulted in lot of issues with unknown root causes. The present understanding can be called as a gray box, partial knowledge of the internal system combined with mathematical system representation, as represented by Figure 1.9(b). The Goal of the thesis is to develop a white box model, physics-based model with detailed knowledge of the internal system.

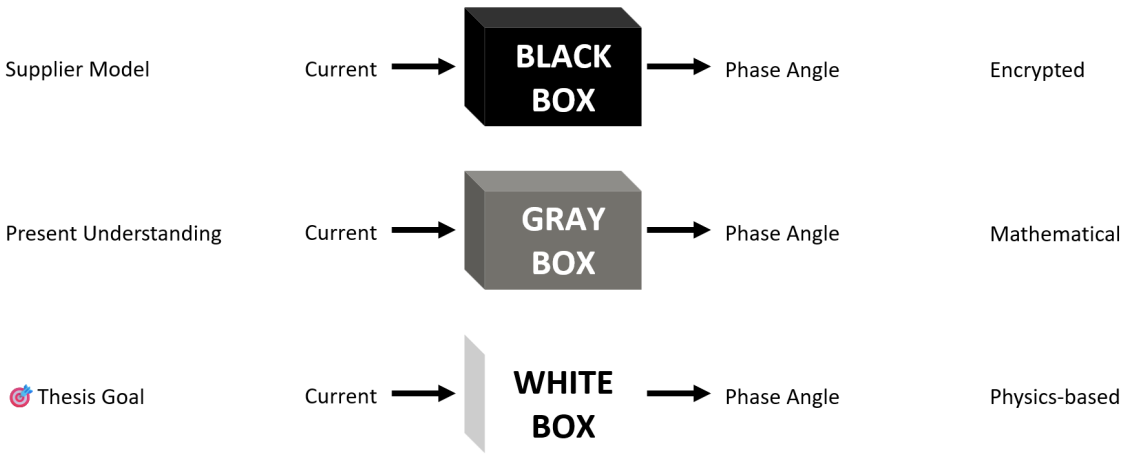


Figure 1.7: Black, Grey, and White Box Modelling Approach

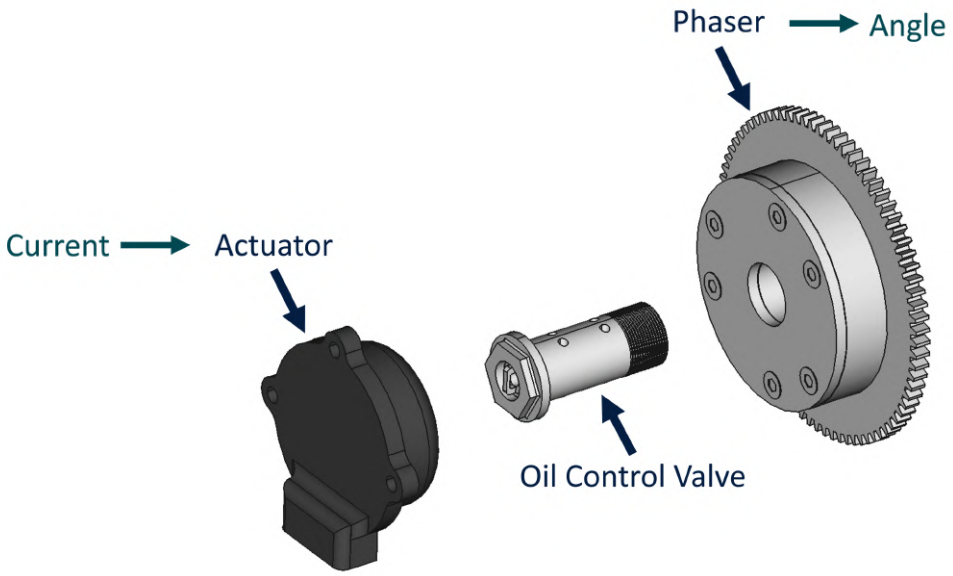
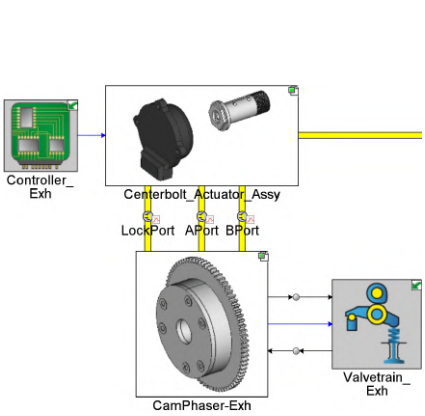
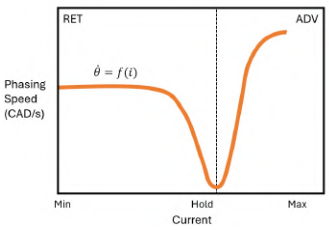


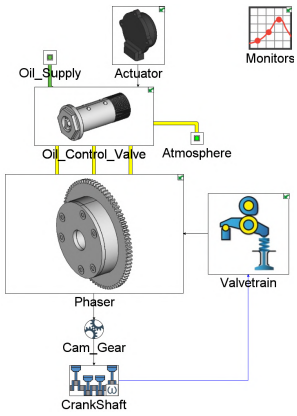
Figure 1.8: Cam Phaser System



(a) Black Box: Encrypted Model



(b) Grey Box: Mathematical Model



(c) White Box: Physics-based Model

Figure 1.9: Black, Grey, White Box Modelling Approach: Models

1.4 Goals

The main goals of this thesis are:

1. **Visualization:** Visualize the CTA phaser operation and internal oil flow.
2. **Modelling:** Develop a GT-SUITE model of the CTA phaser system.
3. **Calibration:** Calibrate the developed model against measurement data.

2
Cam Phaser

2.1 Let's Build the Cam Phaser

To explain the cam phaser system, the concept is built up step-by-step in this chapter. To start, a simplified six-cylinder engine is illustrated in Figure 2.1, which extends the single-cylinder representation in Figure 1.3 to six cylinders. A six-cylinder engine is chosen as the basis for the explanation, as it is the most commonly used engine configuration in the truck industry [12].

2.1.1 Six Cylinder Engine

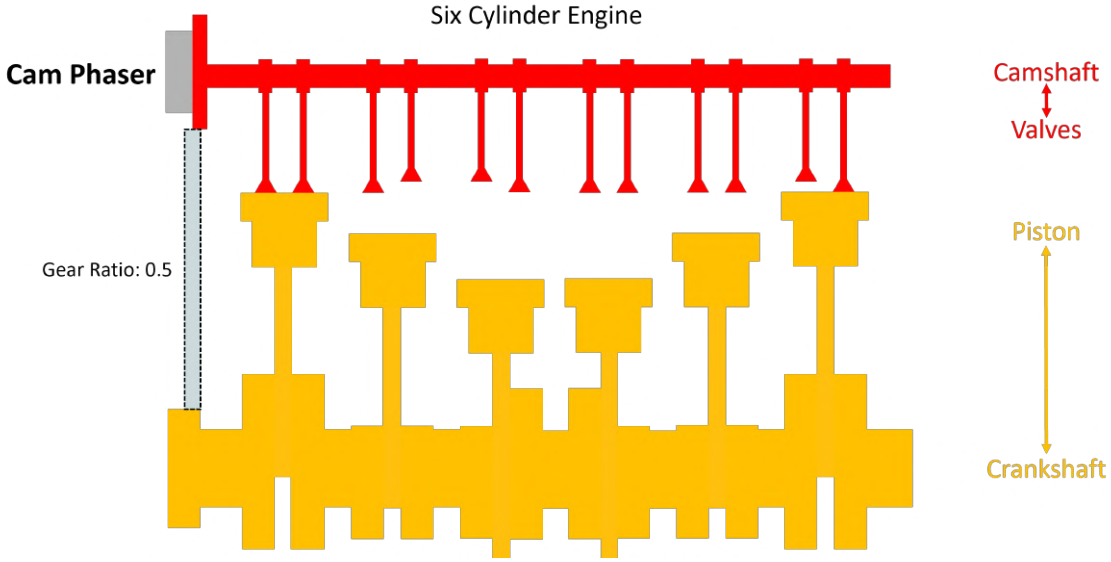


Figure 2.1: Simplified Six Cylinder Engine

In the Figure 2.1, the combustion happens between the valves and the piston. The valves are connected to the camshaft and the crankshaft to the pistons. So, phasing the camshaft and crank shaft will give control over the combustion. This is achieved using a cam phaser, which is generally mounted at the end of the camshaft.

2.1.2 Cam Phaser

The cam phaser is a hydraulic coupling consisting of a rotor and stator, as shown in Figure 2.2. The rotor, highlighted in red, is connected to the camshaft and valves, while the stator, highlighted in yellow, is connected to the crankshaft and piston. Relative motion between the rotor and stator enables phasing.

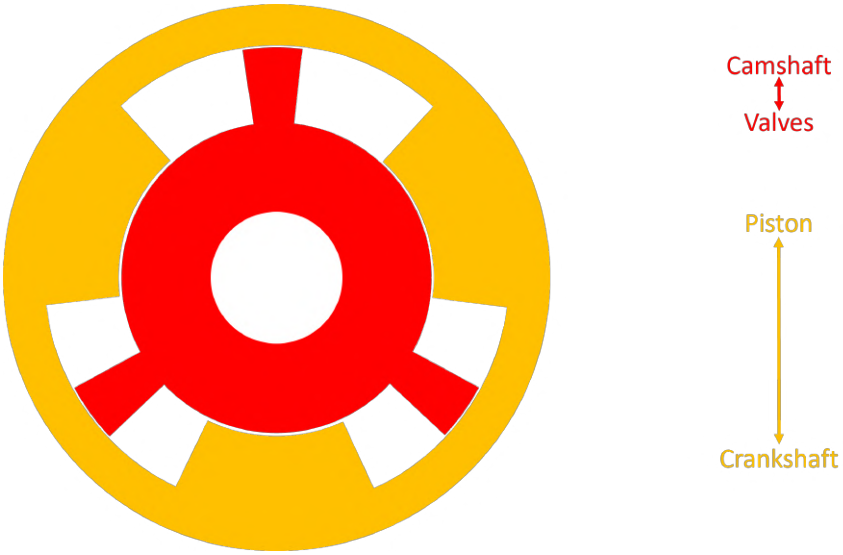


Figure 2.2: Cam Phaser: Rotor & Stator

As shown in Figure 2.3, the cam phaser assembly consists of two chambers, aptly named as the Advance and Retard chambers according to the phasing direction associated with each chamber. Vane seals separate these chambers, while internal channels in the rotor connect them to the Oil Control Valve in the center, which controls the oil flow to and from the chambers.

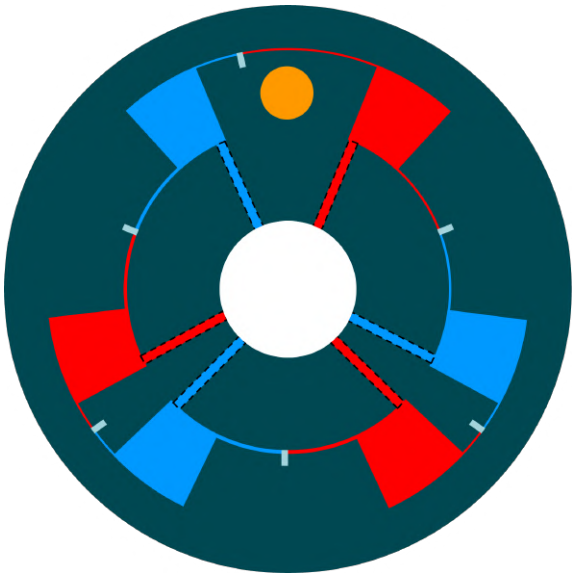


Figure 2.3: Cam Phaser

Figures 2.4 and 2.5 show the effect of rotor/camshaft rotation on the valve event:

- **Advance:** The rotor moves anticlockwise relative to the stator, shifting the valve event to the left and causing the valve to open earlier.
- **Retard:** The rotor moves clockwise relative to the stator, shifting the valve event to the right and causing the valve to open later.

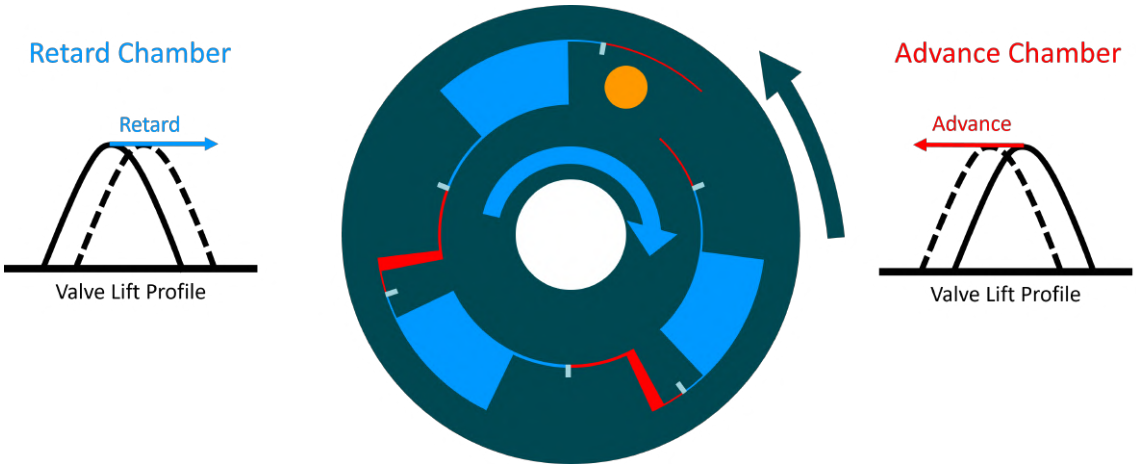


Figure 2.4: Cam Phaser: Retard

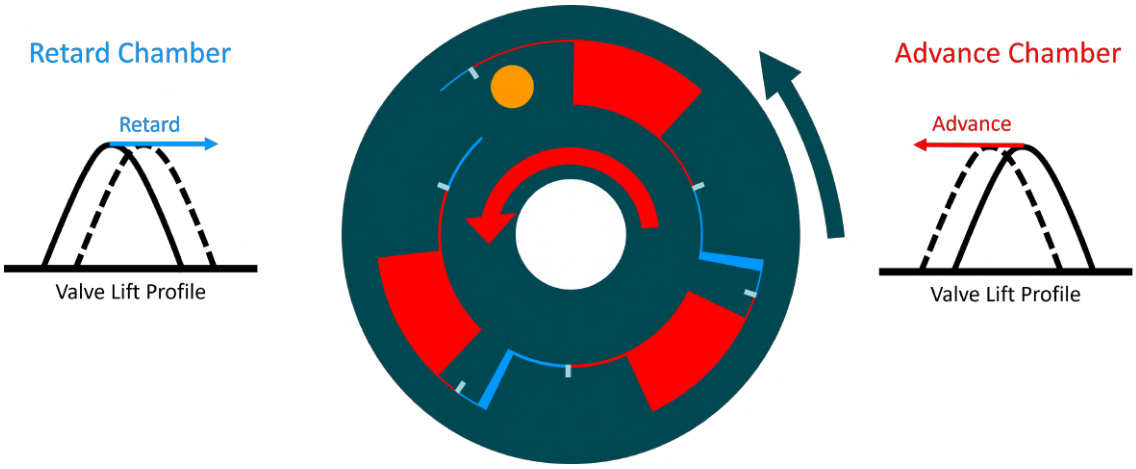


Figure 2.5: Cam Phaser: Advance

2.2 Oil Flow

The oil flow through the cam phaser is controlled by the position of the Oil Control Valve. Depending on the Oil Control Valve position, the supply and vent paths to the Advance and Retard chambers are opened or closed, thereby controlling the direction of phasing. Figures 2.6–2.8 illustrate the oil-flow paths for the three operating positions.

2.2.1 Hold

When the Oil Control Valve is in the Hold position, as shown in Figure 2.6, it blocks the supply and vent paths to both the Advance and Retard chambers. Oil is therefore trapped within the chambers, hydraulically maintaining the relative position between the rotor and stator and allowing the cam phaser to hold at any given position.

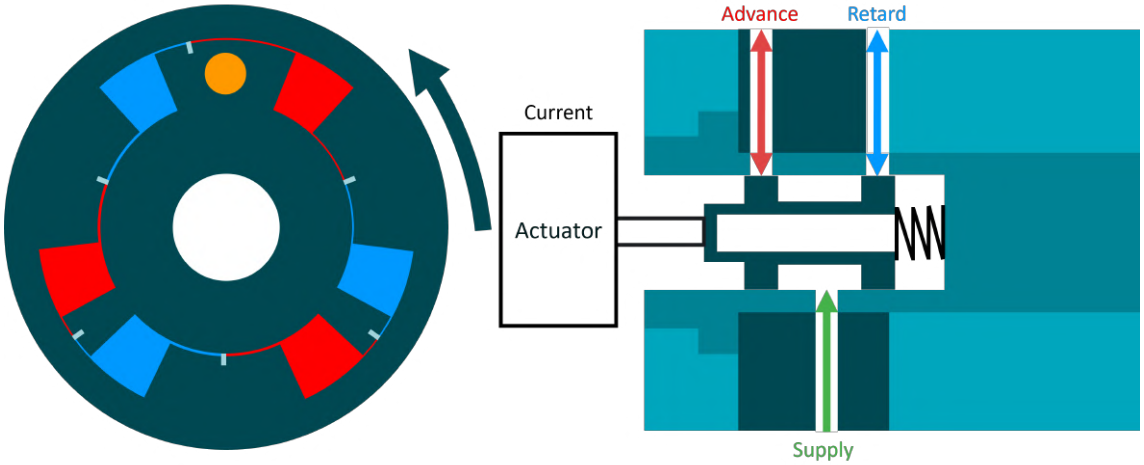


Figure 2.6: Oil flow in Hold position

2.2.2 Retard

When the Oil Control Valve is in the Retard position, as shown in Figure 2.7, it connects the Supply to the Retard chamber and the Advance chamber to the Vent. The resulting pressure difference between the chambers generates a hydraulic torque on the rotor, causing it to rotate relative to the stator in the Retard direction.

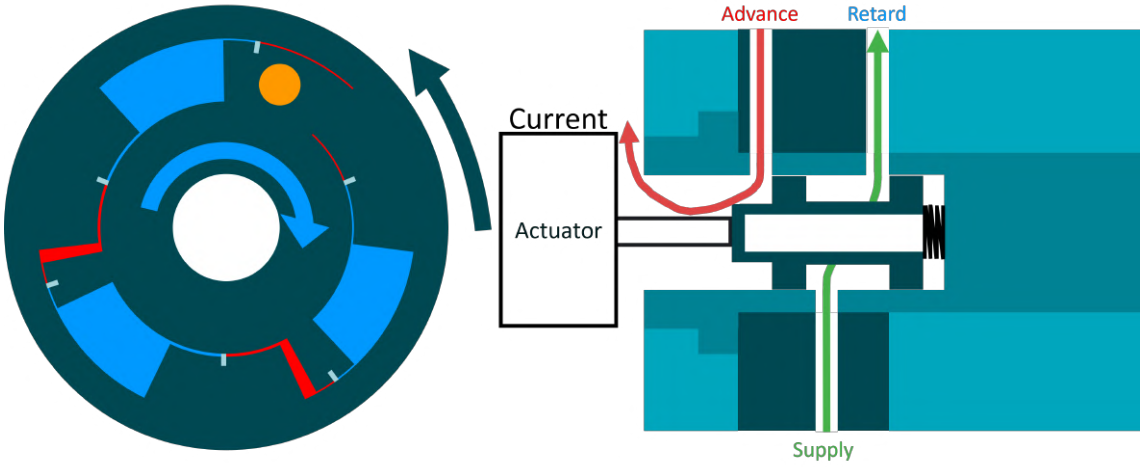


Figure 2.7: Oil flow during Retard phasing

2.2.3 Advance

When the Oil Control Valve is in the Advance position, as shown in Figure 2.8, it connects the Supply to the Advance chamber and the Retard chamber to the Vent. The resulting pressure difference between the chambers generates a hydraulic torque on the rotor, causing it to rotate relative to the stator in the Advance direction.

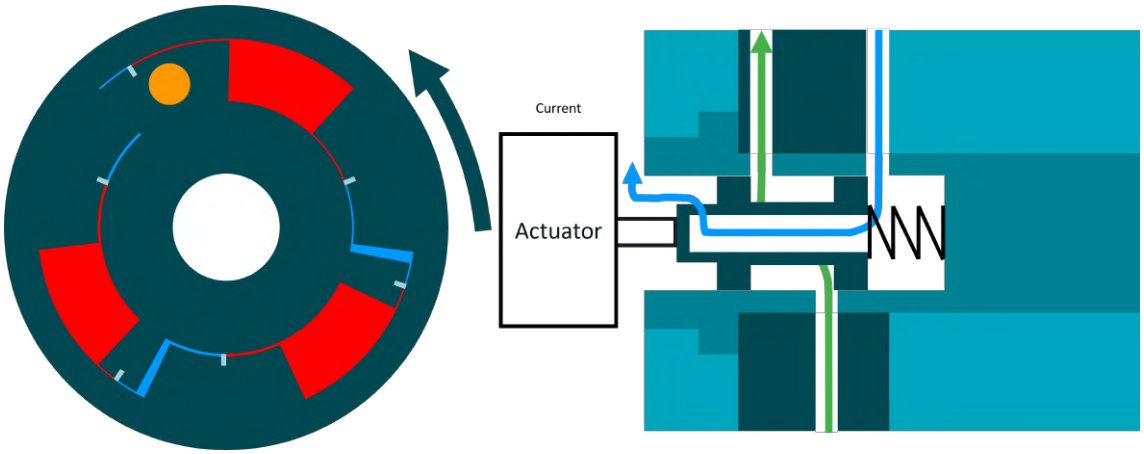


Figure 2.8: Oil flow during Advance phasing

The oil flow paths for the three operating positions are summarized in Figure 2.9.

Phasing	Supply → Retard	Supply → Advance	Retard → Vent	Advance → Vent
Retard	✓	✗	✗	✓
Hold	✗	✗	✗	✗
Advance	✗	✓	✓	✗

Figure 2.9: Torque Assisted Phaser's Oil Flow Table

The flow paths described above correspond to a conventional Torque Assisted phaser, where the primary driving force is provided by the supplied oil pressure. In a Cam Torque Actuated phaser, the camshaft torque energy generated by the valvetrain is instead utilized to pressurize the phaser chambers and drive the phasing motion [5]. To understand how this changes the oil-flow paths, a brief introduction to cam torque is provided before discussing the resulting flow.

2.3 Cam Torque

Cam torque is generated by the interaction between the camshaft and the valvetrain. The sign convention used in this work defines torque acting against the direction of camshaft rotation as positive cam torque, while torque acting in the direction of camshaft rotation is defined as negative cam torque.

The main sources of cam torque are [5]:

- **Valve Spring Force:** Figure 2.10 illustrates cam torque generated during one rotation of the camshaft. As the camshaft rotates clockwise:
  1. During **valve opening**, the cam lobe pushes the valve downward and compresses the valve spring. The spring force opposes the camshaft rotation, producing positive cam torque.
  2. As the cam lobe approaches **maximum valve lift**, the spring force acts increasingly closer to the centre of the camshaft, therefore the torque decreases and approaches zero at maximum lift.
  3. During **valve closing**, the compressed spring pushes the valve back towards its closed position and assists the camshaft rotation, producing negative cam torque.
  4. Once the **valve is closed**, the spring force no longer produces a significant cam torque, while friction continues to oppose the camshaft rotation and provides a positive torque contribution.
- **Combustion Pressure Acting on the Valve:** Combustion pressure acting on the valve can contribute to the load transmitted through the valvetrain to the camshaft.
- **Valvetrain Inertia and Friction:** The acceleration of the valve and other valvetrain components generates inertial forces, which become more significant at higher engine speeds. Friction acts against the direction of camshaft rotation throughout the valve event, producing a positive torque contribution. This is illustrated by the red torque arrows in Figure 2.10, including at the closed-valve position where the valve spring does not contribute to cam torque.

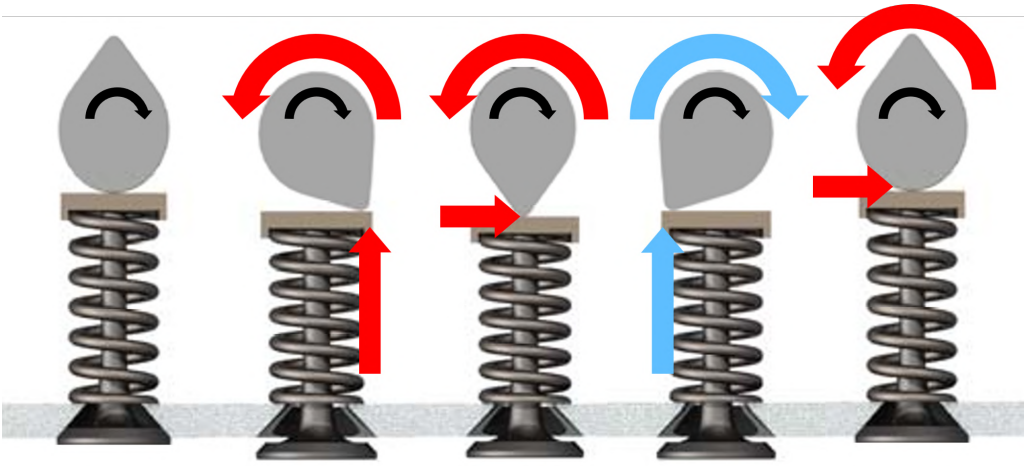


Figure 2.10: How the cam torque direction changes through a valve event (reproduced from Smith and Simpson [5])

Figure 2.11 shows the cam torque over one engine cycle at 1000 RPM, generated by the cam torque model used in this thesis. The profile contains six alternating positive and negative cam torque regions, corresponding to the six-cylinder engine model. The cam torque model only models the contribution from the valve spring force and does not include the effects of combustion pressure or friction. Therefore, the resulting cam torque magnitude does not represent the actual magnitude experienced in a firing engine.

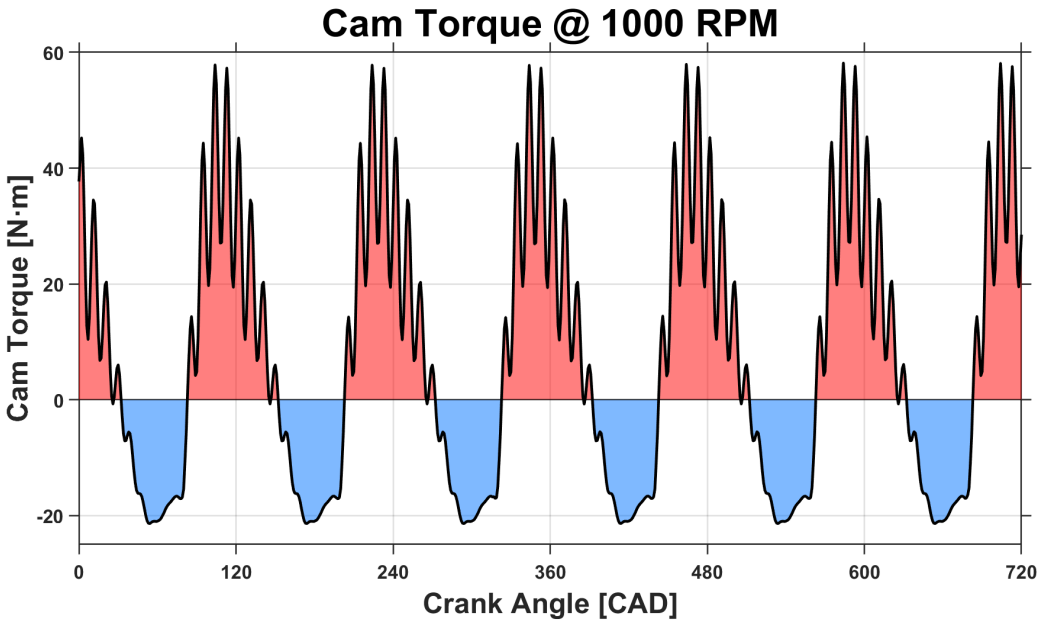


Figure 2.11: Cam torque over one engine cycle at 1000 RPM

These alternating positive and negative torque regions provide energy in both rotational directions, which a Cam Torque Actuated phaser utilizes to pressurize the phaser chambers and recirculate oil between them [5]. Therefore, unlike the Torque Assisted Phaser flow table, the Cam Torque Actuated phaser flow table must consider both positive and negative cam torque, as shown in Figure 2.12.

	Positive Cam Torque: Retards (Pressurises Advance)				Negative Cam Torque: Advances (Pressurises Retard)			
Phasing	Supply → Retard	Supply → Advance	Retard → Advance	Advance → Retard	Supply → Retard	Supply → Advance	Retard → Advance	Advance → Retard
Retard	✗	✗	✗	✓	✗	✓	✗	✗
Hold	✓	✗	✗	✗	✗	✓	✗	✗
Advance	✓	✗	✗	✗	✗	✗	✓	✗

Figure 2.12: Cam Torque Actuated Phaser's Oil Flow Table

Figure 2.13 compares the Oil Control Valves of the Torque Assisted Phaser and Cam Torque Actuated Phaser. The key differences are:

- **Cam Torque:** In the Torque Assisted Phaser, oscillating cam torque acts as a disturbance that can cause unwanted phaser oscillations. In contrast, the Cam Torque Actuated Phaser turns this disturbance into its primary energy source for phasing.
- **Recirculating Flow:** In the Torque Assisted Phaser, phasing is achieved by supplying pressurized oil to the target chamber while oil from the other chamber is vented. In contrast, the Cam Torque Actuated Phaser achieves phasing by transferring oil to the target chamber from the other chamber using the pressure generated by the cam torque. The external oil supply is therefore primarily required for the initial filling of the phaser and to replenish the small amount of oil lost through leakage during normal operation [5].
- **Check Valves:** The Oil Control Valve of the Cam Torque Actuated Phaser incorporates check valves to prevent reverse oil flow during the opposing cam torque. This makes them key components in converting the oscillating cam torque into a net oil transfer between the chambers in the desired phasing direction.

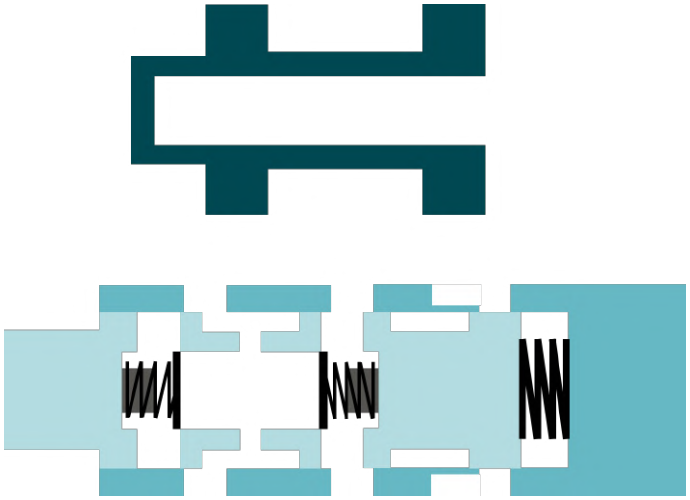


Figure 2.13: Comparison of TA and CTA Oil Control Valve configurations

The flow paths summarized in Figure 2.12 can be physically identified in the sectional views of the Cam Torque Actuated Phaser. Figure 2.14 shows the connections to the phaser chambers, while Figure 2.15 shows the connection to the external oil supply.

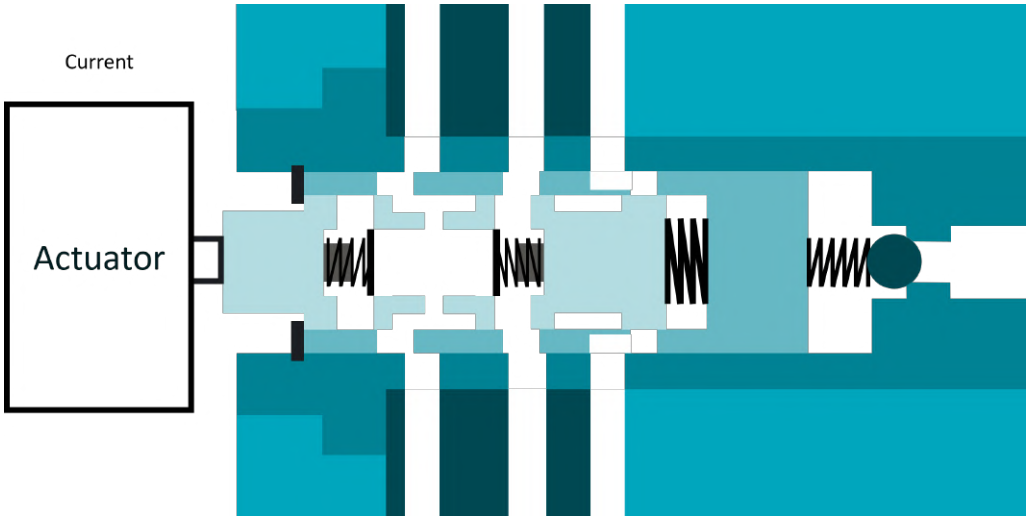


Figure 2.14: CTA Oil Control Valve Chamber Flow Path

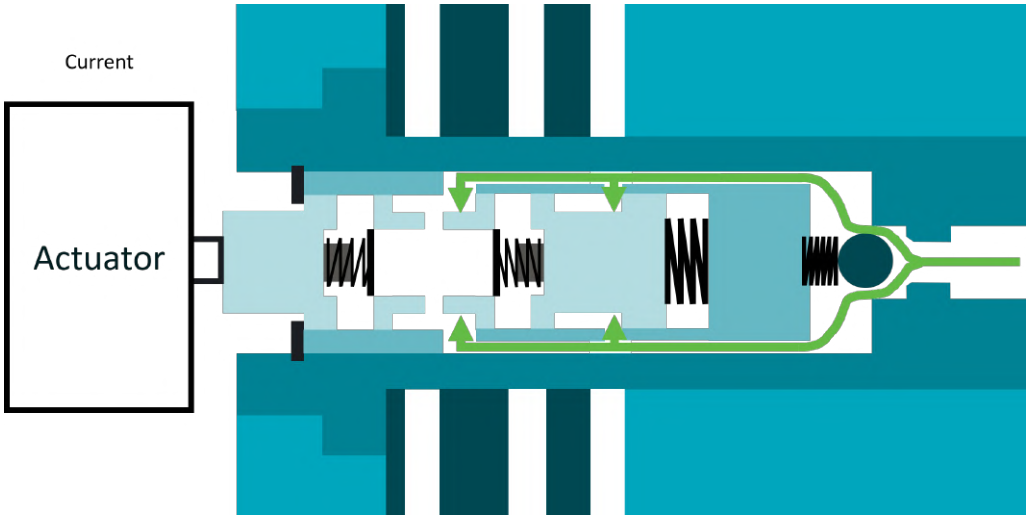


Figure 2.15: CTA Oil Control Valve Oil Supply Flow Path

2.4 Cam Torque Actuated Phaser Oil Flow

Smith and Simpson [5] intuitively describe the Cam Torque Actuated Phaser mechanism as a “Hydraulic Ratchet or One-way Clutch”, with the cam torque energy supplying the power to move the phaser and the check valves creating the one-way ratcheting mechanism.

The Cam Torque Actuated phaser modeled in this thesis is used on the exhaust camshaft, which rotates anticlockwise. Therefore, Figure 2.16 illustrates the cam torque direction under assumptions similar to those used in the model, with anticlockwise camshaft rotation and friction neglected. During valve opening, the valve spring force opposes the camshaft rotation, producing positive cam torque. During valve closing, the spring force assists the camshaft rotation, producing negative cam torque.

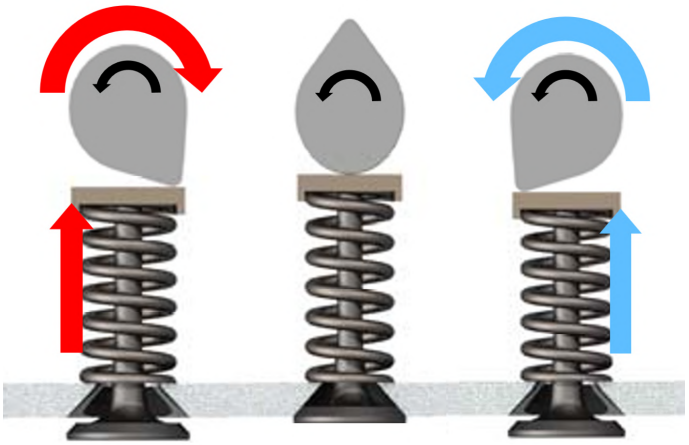


Figure 2.16: Simplified representation of Positive and Negative Cam Torque

2.4.1 Hold

When the Oil Control Valve is in the Hold position, as shown in Figure 2.17, it blocks oil transfer between the Advance and Retard chambers.

- **Positive Cam Torque:** Positive cam torque pressurises the Advance chambers, driving oil towards the Oil Control Valve, where the check valve blocks this flow path. The lower pressure within the Oil Control Valve allows oil from the external supply to enter through the supply check valve.
- **Negative Cam Torque:** Negative cam torque pressurises the Retard chambers, driving oil towards the Oil Control Valve, where the check valve blocks this flow path. The lower pressure within the Oil Control Valve allows oil from the external supply to enter through the supply check valve.
- **Hold:** In both torque directions, the check valves prevent oil from leaving the pressurised chamber, thereby maintaining the relative position between the rotor and stator and allowing the cam phaser to hold at any given position.

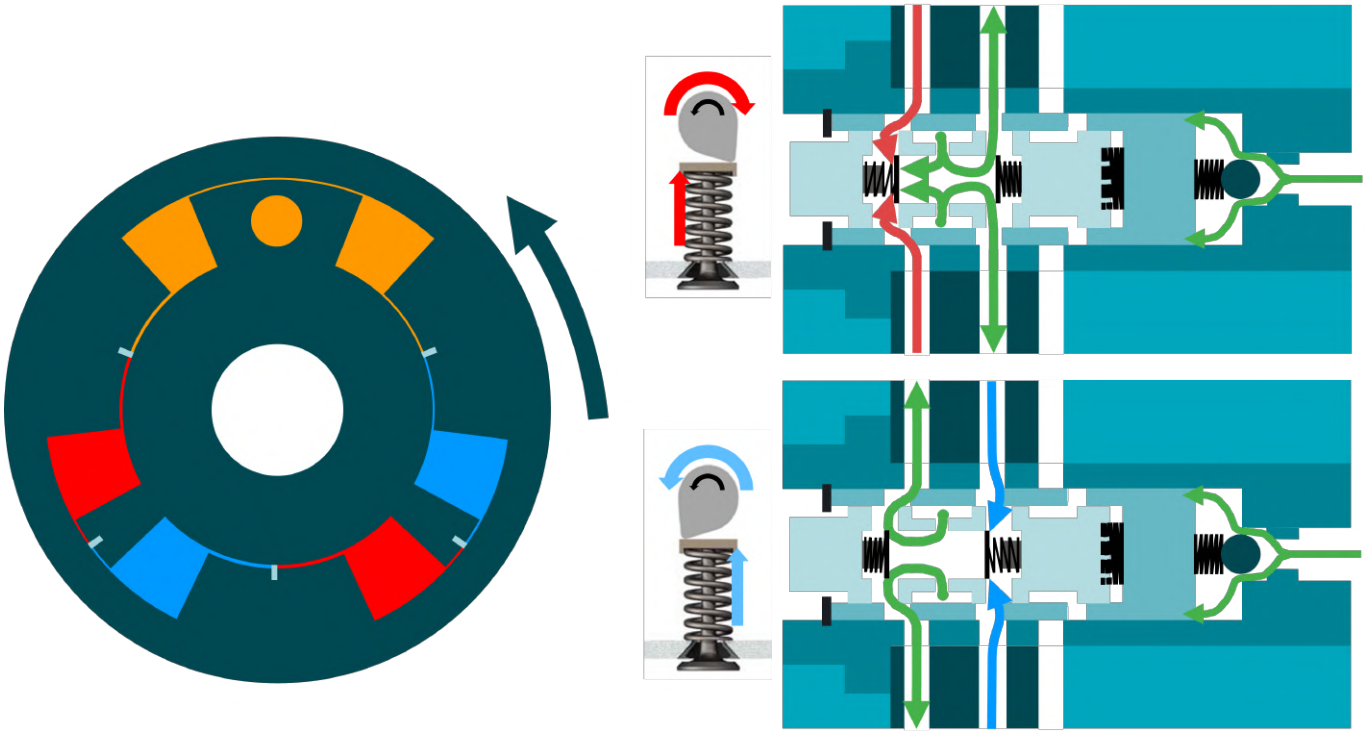


Figure 2.17: Cam Torque Actuated Phaser oil flow in Hold position

2.4.2 Retard

When the Oil Control Valve is in the Retard position, as shown in Figure 2.18, it permits oil transfer from the Advance chambers to the Retard chambers.

- **Positive Cam Torque:** Positive cam torque pressurises the Advance chambers, driving oil towards the Oil Control Valve. The Oil Control Valve permits this flow into the Retard chambers, producing Retard phasing. The resulting high pressure within the Oil Control Valve keeps the supply check valve closed, preventing the lower pressure external oil supply from entering.
- **Negative Cam Torque:** Negative cam torque pressurises the Retard chambers, driving oil towards the Oil Control Valve, where the check valve blocks reverse oil transfer to the Advance chambers. The lower pressure within the Oil Control Valve allows oil from the external supply to enter through the supply check valve.
- **Retard:** Successive positive cam torque pulses transfer oil from the Advance chambers to the Retard chambers, producing progressive phasing in the Retard direction.

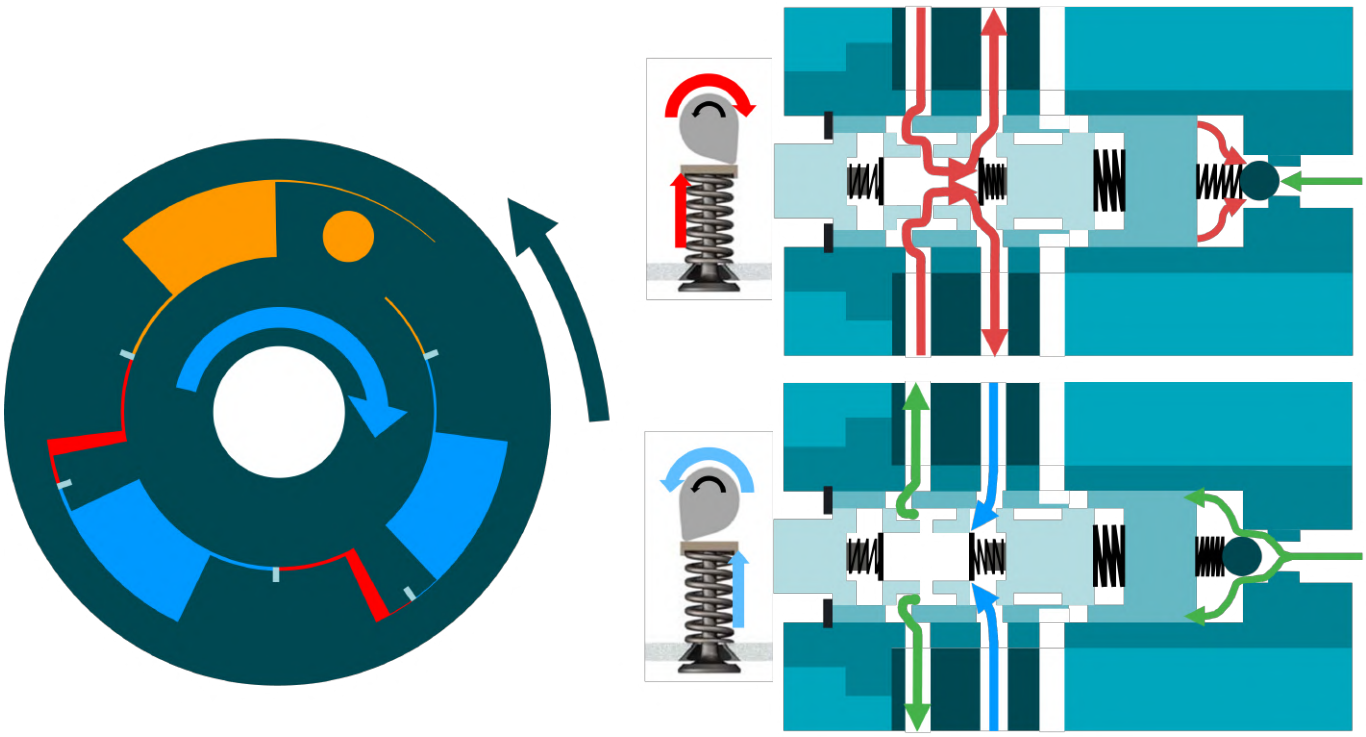


Figure 2.18: Cam Torque Actuated Phaser oil flow during Retard phasing

2.4.3 Advance

When the Oil Control Valve is in the Advance position, as shown in Figure 2.19, it permits oil transfer from the Retard chambers to the Advance chambers.

- **Negative Cam Torque:** Negative cam torque pressurises the Retard chambers, driving oil towards the Oil Control Valve. The Oil Control Valve permits this flow into the Advance chambers, producing Advance phasing. The resulting high pressure within the Oil Control Valve keeps the supply check valve closed, preventing the lower pressure external oil supply from entering.
- **Positive Cam Torque:** Positive cam torque pressurises the Advance chambers, driving oil towards the Oil Control Valve, where the check valve blocks reverse oil transfer to the Retard chambers. The lower pressure within the Oil Control Valve allows oil from the external supply to enter through the supply check valve.
- **Advance:** Successive negative cam torque pulses transfer oil from the Retard chambers to the Advance chambers, producing progressive phasing in the Advance direction.

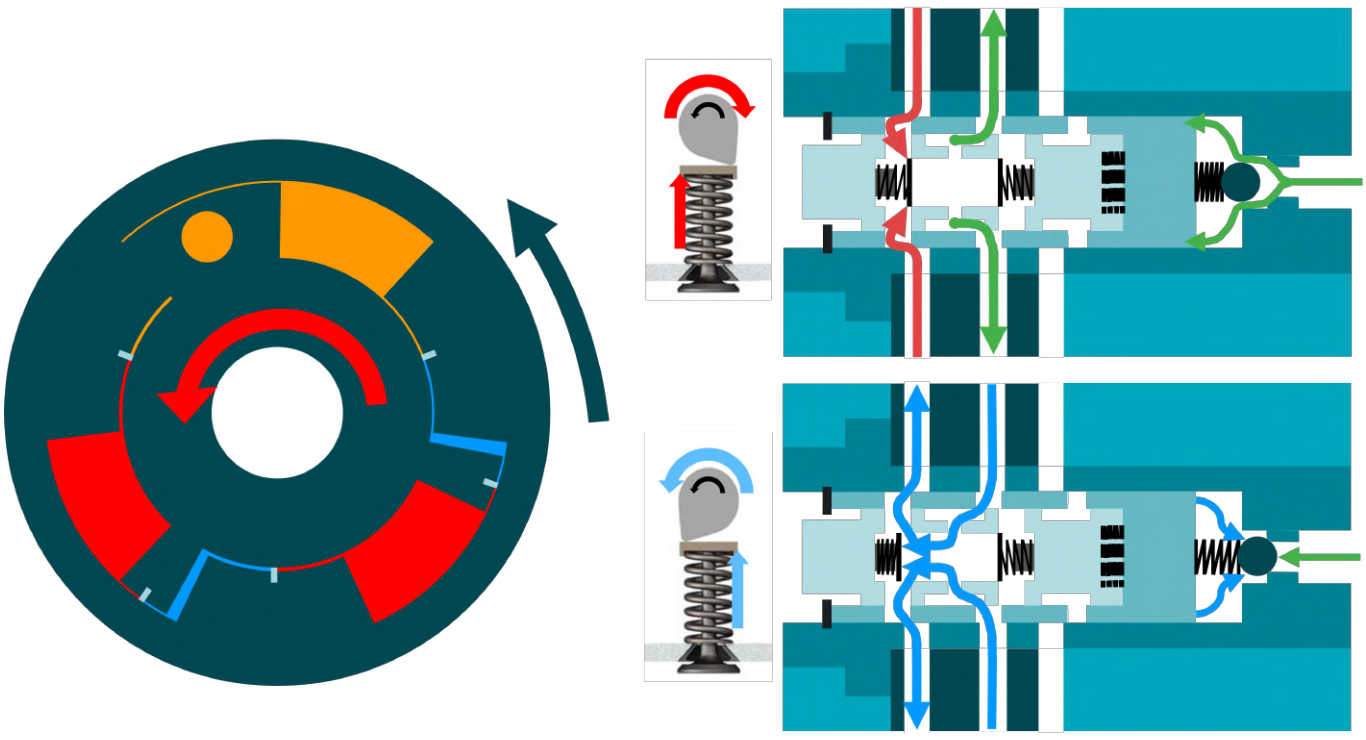


Figure 2.19: Cam Torque Actuated Phaser oil flow during Advance phasing

2.5 Previous Work

Similar works where cam phasers have been modeled where found, *A Camshaft Torque Actuated Vane Style VCT Phaser* [5], *1D Simulation of Hydraulic Cam Phaser System Utilized in Heavy Duty Vehicles* [6]

2.5.1 A Camshaft Torque Actuated Vane Style VCT Phaser

Smith and Simpson (2005) introduced a camshaft torque actuated (CTA) vane-type Variable Cam Timing (VCT) phaser, based on the same fundamental operating principle considered in this thesis. The concept differs from conventional hydraulic oil-pressure-actuated (OPA) systems by utilizing camshaft torque energy rather than relying solely on oil pressure for actuation [5].

The paper highlights several aspects that are particularly relevant to the Cam Torque Actuated phaser considered in this thesis:

- **Cam Torque as Energy Source:** Camshaft torque originates from valve spring forces, inertial effects, and friction during valve events. The resulting alternating positive and negative torque provides the energy required to hydraulically transfer oil between the Advance and Retard chambers [5].
- **Hydraulic Ratchet Mechanism:** The spool and check valve arrangement controls the direction of oil transfer between the chambers. Smith and Simpson describe this mechanism as a hydraulic ratchet, where cam torque provides the energy for actuation while the hydraulic control system determines the direction and rate of phasing [5].
- **Internal Oil Recirculation:** During actuation, oil is transferred from one chamber to the other rather than being continuously supplied and vented. The external oil supply is primarily required for initial filling and to compensate for leakage, resulting in considerably lower oil-flow requirements compared to conventional oil-pressure-actuated phasers [5].
- **Experimental Validation:** The Cam Torque Actuated phaser was experimentally evaluated on a DOHC V6 engine. The testing demonstrated:
  - Fast response rates over a wide engine-speed range,
  - Actuation during engine cranking,
  - Low oil-flow requirements,
  - Small oscillation amplitudes under closed-loop control,
  - Return to base timing at temperatures as low as $-28°C$.

Overall, the work of Smith and Simpson provides the fundamental operating principles and experimental basis for Cam Torque Actuated phasing, which are closely related to the phaser architecture investigated and modelled in this thesis.

2.5.2 1D Simulation of a Hydraulic Cam–Torque Activated Vane Cam Phaser

The previous thesis work in Scania conducted by Johan Germundsson (2023) involved the development of a detailed 1D GT-SUITE model of a hydraulic vane-type cam torque actuated phaser to investigate severe phase oscillations observed in heavy-duty engine applications [6]. The work focused on identifying the dominant physical mechanisms affecting phaser stability.

The thesis was centered around the following key research themes:

- **Root Cause of Phase Oscillations:** Free air inside the advance and retard chambers was identified as the dominant factor reducing hydraulic stiffness. Increased free air lowers torsional stiffness and shifts the resonance frequency toward the cam torque excitation frequency, significantly amplifying oscillations.
- **Resonance Mechanism:** The phaser's resonance frequency was shown to depend on rotor inertia, chamber volume, phaser displacement, and fluid stiffness (pressure and air content). Under ideal conditions the resonance frequency remains above the cam torque frequency; however, increased aeration causes it to drop into the excitation range.
- **Spool–Pressure Interaction:** Spool valve motion directly influences chamber pressures. Small spool-induced pressure reductions can promote air ingestion, reduce stiffness, and trigger instability. PWM-driven spool dithering was found to significantly impact pressure dynamics.
- **Leakage and Oil Properties:** Internal and external leakage strongly affect chamber pressure levels. Reduced oil viscosity increases leakage, lowers chamber pressure, and decreases the phaser's ability to attenuate cam torque disturbances.
- **Air Sources and Aeration:** Three primary air sources were identified: air in supplied oil, air ingress via OCV vent, and air drawn through leakage paths. Pressure drops below atmospheric level were shown to be critical in promoting air ingestion.

Design Improvement Directions:

The study emphasized increasing chamber pressure margins and minimizing free air as primary stability objectives. Key improvement strategies included:

- Increasing phaser displacement to raise stiffness (with resonance trade-offs),
- Reducing external leakage to improve chamber pressure support,
- Optimizing OCV port geometry and spool control strategy,
- Avoiding PWM/dither frequencies close to the phaser resonance,
- Introducing vent back-pressure or oil accumulators to reduce air ingestion,
- Reducing cam torque excitation amplitude.

Model Scope and Demarcations:

The study focused solely on the cam phaser subsystem. Cam torque, oil supply pressure, and engine speed were applied as boundary conditions. The full engine, detailed valvetrain dynamics, and complete oil system were not modeled. A simplified control strategy was implemented, and only selected operating conditions were investigated.

Overall, the thesis established a physics-based explanation for phase instability in hydraulic cam phasers and identified aeration-driven stiffness reduction as the dominant instability mechanism. This previous work provides relevant modelling knowledge and physical understanding that supports the development of the cam phaser model in the present thesis.

3
Numerical Approach

This chapter presents the working principles of the GT-SUITE templates used to develop the cam phaser model. The focus is on understanding how each template represents the corresponding physical behaviour. The implementation of these templates and the model-specific inputs are presented in the next Chapter 4 Modelling.

3.1 GT-Suite

GT-SUITE was selected as the simulation environment for developing the cam phaser model. The main reasons for choosing GT-SUITE are:

- **System-Level Simulation:** GT-SUITE is a system simulation tool that combines 0D lumped-parameter modelling with 1D flow modelling. Hydraulic, mechanical, and control components can be modelled and coupled within a single simulation environment, making it suitable for representing the cam phaser as a complete system and investigating the interaction between its individual components.
- **Established Simulation Tool at Scania:** GT-SUITE is already used within Scania for system-level modelling and simulation. Developing the cam phaser model in the same environment allows it to be integrated with existing engine and subsystem models for further development and investigation.

The following sections briefly introduce the main GT-SUITE templates and solvers used in developing the cam phaser model.

3.2 GT Solver

3.2.1 Flow Solver

In this section, the theory behind GT-Suite flow solvers and the choice of flow solver used in the model is presented. The templates in the GT-Suite's Flow library use one-dimensional Navier-Stokes equations as the flow model. The conservation equations solved by GT-SUITE are shown below.

$$\text{Continuity:} \quad \frac{dm}{dt} = \sum_{\mathrm{B}} \dot{m} \tag{3.1}$$

$$\text{Momentum:} \quad \frac{d\dot{m}}{dt} = \frac{dpA + \sum_{\mathrm{B}} \dot{m}u - 4C_f \frac{\rho u|u|}{2} \frac{dxA}{D_h} - \frac{1}{2}\rho u|u|K_pA}{dx} \tag{3.2}$$

$$\text{Energy:} \quad \frac{d(me)}{dt} = -\rho\frac{dV}{dt} + \sum_{\mathrm{B}} \dot{m}H - hA_s(T_f - T_w) \tag{3.3}$$

where:
B Boundaries of the fluid volume $\dot{m}$ boundary mass flux into volume, $\dot{m} = \rho Au$
m mass of the volume
V volume
p pressure
ρ density
A cross-sectional flow area
A_s heat transfer surface area
e total specific internal energy (internal energy plus kinetic energy per unit mass)
$H = e + \frac{p}{\rho}$ total specific enthalpy
h heat transfer coefficient
T_{f} fluid temperature
T_{w} wall temperature
u velocity at the boundary
C_f Fanning friction factor
K pressure loss coefficient (commonly due to bend, taper or restriction)
D_h equivalent hydraulic diameter
dx length of mass element in the flow direction (discretization length)
$\frac{dp}{dx}$ pressure differential acting across dx

3.2.1.1 Explicit Method

GT-SUITE offers both explicit and implicit time integration methods for solving the flow equations. In the explicit method, the solution at the current time step is calculated using values from the previous time step. This makes the method well suited for capturing fast transient phenomena such as pressure pulsations and high-frequency hydraulic dynamics [17].

The explicit solver requires sufficiently small time steps to maintain numerical stability. The time step is limited by the Courant condition, which ensures that a pressure wave does not travel through more than one discretized flow volume within a single time step:

$$\frac{\Delta t}{\Delta x}\left(|u| + c\right) \leq 0.8M_t \tag{3.4}$$

where Δt is the time step, Δx is the discretization length, u is the fluid velocity, c is the speed of sound in the fluid, and M_t is the time-step multiplier.

The corresponding stable time step is calculated as

$$\Delta t = \frac{0.8 M_t \Delta x}{|u| + c} \tag{3.5}$$

Because the achievable time step depends directly on the discretization length and fluid wave speed, the explicit solver can require very small time steps for hydraulic simulations [17].

3.2.2 Numerical Solver

While the flow solver calculates the fluid flow by solving the conservation equations, the numerical solver handles the time-dependent behaviour. The numerical solver calculates how the system states evolve over time by solving the corresponding differential equations. The solvers interact during the simulation, allowing the hydraulic and mechanical behaviour of the cam phaser to be coupled.

A general system of ordinary differential equations (ODEs) can be written as

$$\dot{x} = f(x, t) \tag{3.6}$$

where x represents the system states, $\dot{x}$ their time derivatives, and f describes how the states change over time.

When a system includes algebraic relationships or constraints, its governing equations can be expressed as differential-algebraic equations (DAEs):

$$F(x, \dot{x}, z, t) = 0 \tag{3.7}$$

where x represents the differential states, $\dot{x}$ their time derivatives, and z the algebraic variables.

Unlike an ODE formulation, this representation does not require the derivatives to be isolated on one side of the equations, instead, the differential equations and algebraic constraints are considered together.The DAE formulation is suitable for the cam phaser model because the equations of motion in multibody dynamic systems can be nonlinear, stiff and expressed as differential-algebraic equations (DAEs) [18].

3.3 GT-SUITE Template Theory

3.3.1 Oil Control Valve

3.3.1.1 Spool Cover Hole

The *SpoolCoverHole* template is used to model the interaction between a moving spool section and a circular port in the stationary sleeve. As the spool moves, it progressively covers or uncovers the port, and the template calculates the resulting change in flow area as a function of spool displacement [19].

Flow Area
The *SpoolCoverHole* template calculates the available geometrical flow area as a function of spool position. The curtain area represents the opening area available as the spool moves, while the hole area is the maximum flow area through the hole. The throat area represents the maximum flow area around the spool passage. The geometrical area is determined by the smallest of these three areas, which acts as the bottleneck for the flow. The leakage area defines the minimum flow area [19].

$$A_{\text{geometric}} = \max\left[\min\left(A_{\text{curtain}},\ A_{\text{throat}},\ A_{\text{hole}}\right), A_{\text{leak}}\right] \tag{3.8}$$

The effective area available for flow is then calculated using the discharge coefficient C_d:

$$A_{\text{flow}} = C_d A_{\text{geometric}} \tag{3.9}$$

This template is suitable for modelling the Oil Control Valve (OCV), as the flow paths are controlled by the translation of the spool relative to the circular holes. In addition to calculating the flow area, the template accounts for pressure forces acting on the spool, volume changes caused by spool displacement, and leakage when the spool is in an overlap or closed position [19].

3.3.1.2 Volume Flowsplit

The FlowSplit template models an arbitrarily shaped flow volume as a single lumped fluid volume with multiple boundaries. The fluid state is calculated within this volume, while the flow through each boundary is calculated separately. The mass within the volume follows the continuity balance [17]:

$$\frac{dm}{dt} = \sum_{\text{boundaries}} \dot{m} \tag{3.10}$$

where m is the mass of fluid within the FlowSplit volume, and $\dot{m}$ represents the mass flow rate across each boundary.

The geometry of a FlowSplit is mainly defined by the following parameters [17]:

- **Volume**
- **Boundary Angle:** Defines the orientation of each boundary and thereby the direction of flow entering or leaving the volume.
- **Expansion Diameter:** Defines the maximum diameter into which the flow can expand after entering a boundary, influencing contraction/expansion losses and pressure recovery.
- **Characteristic Length:** Defines the distance the fluid can travel within the FlowSplit before crossing another boundary or impacting a surface.

The *FlowSplitGeneral* template is suitable for geometries that cannot be adequately represented by the predefined FlowSplit shapes [17].

3.3.1.3 Check Valve

The *CheckValve* template models a check valve using performance data typically available from supplier data sheets or physical testing, without requiring a detailed representation of the valve geometry, mass, and spring characteristics. This provides a simplified representation of the valve when its performance characteristics are known [20].

A check valve allows flow in only one direction, preventing backflow. It begins to open when the pressure difference across the valve exceeds the specified cracking pressure. The valve opening and resulting flow can then be defined using either an effective diameter or a pressure drop–flow rate relationship.

The dynamic response of the valve is governed by the *Valve Lift Time Constant*. A smaller time constant corresponds to a faster valve response, while a larger value results in slower opening and closing. The time constant should not be smaller than the simulation time step. [20]

3.3.2 Phaser

3.3.2.1 Lumped Rotational Inertia: Stator & Rotor

The *Lumped Rotational Inertia* template represents a rotating component as a single lumped moment of inertia, combining the rotational inertia of its subcomponents instead of modelling their individual geometries and mass distributions [21]. This allows the rotor and stator assemblies to be represented by their equivalent lumped inertias.

The rotational motion is described by Newton's second law for rotation:

$$\sum T = J\alpha \tag{3.11}$$

where T is the applied torque, J is the moment of inertia, and α is the angular acceleration.

3.3.2.2 Stator & Rotor Mechanical & Hydraulic Connections

The connection between the stator and rotor is modelled using the mechanical and hydraulic connections listed below to represent the physical interactions between the two assemblies.

Phaser Friction
The *FrictionConnRot* template represents friction between two rotating bodies by accounting for both dry and viscous friction and converting the resulting friction force into a resisting torque, representing the frictional effects between the stator and rotor [22].

Phaser Damping
The *DamperTorsion* template represents torsional damping between two rotating bodies by applying a damping torque based on their relative angular velocity, capturing the damping effect between the stator and rotor [23].

Stator–Rotor Contact
The *ContactRot* template defines the angular limits of relative motion between the rotor and stator by applying a contact torque when rotational limit is reached, representing the mechanical end stops of the phaser [24].

Bias Spring
The cam torque profile in Figure 2.11 indicates that the positive cam torque magnitude exceeds the negative cam torque, creating the need for a bias spring to compensate for this imbalance. The *SpringTorsion* template represents the bias spring connecting the rotor and stator by generating a torque based on their relative angular displacement, the spring's torsional stiffness, and its preload [25].

$$T_2 = k(\theta_1 - \theta_2) - T_p \tag{3.12}$$

where k is the torsional stiffness, $\theta_1 - \theta_2$ is the relative angular displacement, and T_p is the spring pretension.

Vane Interaction
The *MechFlowConnRot* template couples the hydraulic chambers with the rotational motion of the rotor and stator. Chamber pressure acts over an effective vane area to generate a force, which is converted into torque using an effective radius [26]:

$$T = m\, P\, A_{\text{eff}} R_{\text{eff}} \tag{3.13}$$

The change in volume of the chambers due to the relative angular displacement between the rotor and stator is given by [26]:

$$\Delta V = A_{\text{eff}} R_{\text{eff}} (\theta - \theta_0) \tag{3.14}$$

This coupling allows chamber pressure to drive phaser rotation, while the rotor's angular displacement determines the change in chamber volume.

3.3.3 Actuator

The actuator converts the controller output into the mechanical force required to move the Oil Control Valve. It consists of the PID controller and Solenoid templates, which represent the control and electromagnetic behaviour, respectively.

3.3.3.1 PID

The *PIDController* template represents a continuous closed-loop controller. At each time step, the controller calculates the error between the target and measured input signals and generates an output to reduce this error [27].

$$e(t) = r(t) - y(t) \tag{3.15}$$

The controller output is determined by the proportional, integral, and derivative contributions:

$$u(t) = K_P e(t) + K_I \int e(t)\,dt + K_D \frac{de(t)}{dt} \tag{3.16}$$

where K_P, K_I, and K_D are the proportional, integral, and derivative gains, respectively. The proportional term is proportional to the instantaneous error, the integral term to the accumulated error over time, and the derivative term to the rate of change of the error.

3.3.3.2 Solenoid

The *SolenoidMap* template represents the electromagnetic behaviour of a solenoid using characteristic maps rather than a detailed electromagnetic model. The solenoid force is defined by a force map as a function of electrical current and armature air gap [28]:

$$F_{\text{sol}} = f(I, x) \tag{3.17}$$

where I is the coil current and x is the armature air gap.

The magnetic behaviour is represented using either a magnetic flux map or a permeance map:

$$\Phi = f(I, x) \qquad P_{\text{m}} = f(I, x) \tag{3.18}$$

where Φ is the magnetic flux and P_{m} is the magnetic permeance.

3.3.4 Cam Torque Map / Valvetrain

The *Lookup2D* template represents a two-dimensional lookup table. The two inputs correspond to the X and Y axes of the map, while the corresponding Z value is returned as the output [29]. This relationship can be expressed as:

$$(X_i, Y_j) \longmapsto Z_{ij} \tag{3.19}$$

This allows the cam torque to be obtained from predefined data for the specified operating conditions, without explicitly modelling the valvetrain dynamics.

3.3.5 Others

3.3.5.1 Boundary Pressure: Oil Supply & Atmosphere

The *BoundaryPressure* template defines the pressure, temperature, and fluid composition at the boundary of a flow system. It acts as a large reservoir, allowing fluid to enter or leave the connected system depending on the pressure difference across the boundary [30]. Therefore, this template is used to represent the oil supply and the surrounding atmosphere. The fluid composition is specified using a corresponding fluid reference object, allowing the same template to represent both boundaries.

3.3.5.2 Crankshaft

The *EngineCrankShaft* template represents a simplified engine crankshaft that drives the cam phaser, acting as a rotational speed boundary [31].

4
Modelling

This chapter continues on how the templates in GT-Suite mentioned in Chapter 3 were used to build the Cam Torque Actuated Phaser Model.

4.1 Cam Torque Actuated Phaser Model

The system consists of Oil Control Valve, Phaser, Actuator and Valvetrain. Subassemblies of each component were created to build the model as given in Figure 4.1.

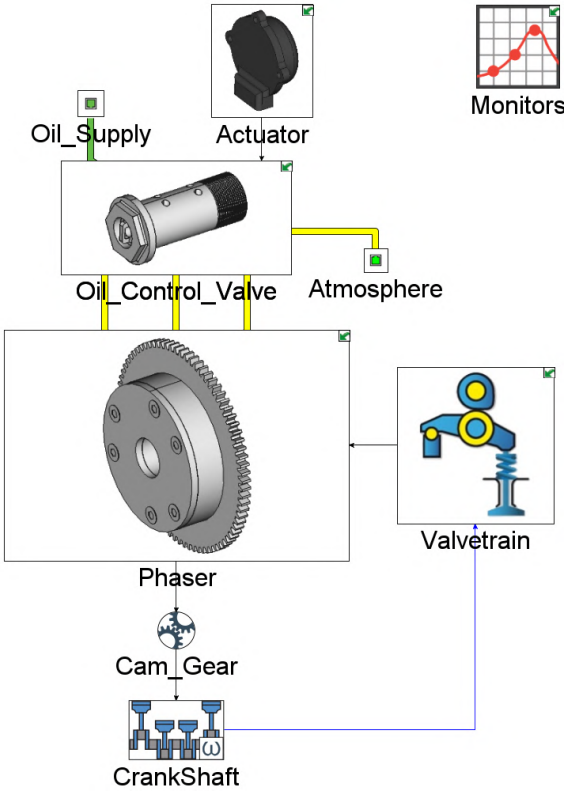


Figure 4.1: Cam Torque Actuated Phaser Model

4.1.1 Oil Control Valve

The Oil Control Valve subassembly model was the primary focus of this thesis. As mentioned in Chapter 3, the Oil control valve is modeled using the templates in

GT-Suite such as Spool Cover Holes, Volume Flow Splits and Check Valves.

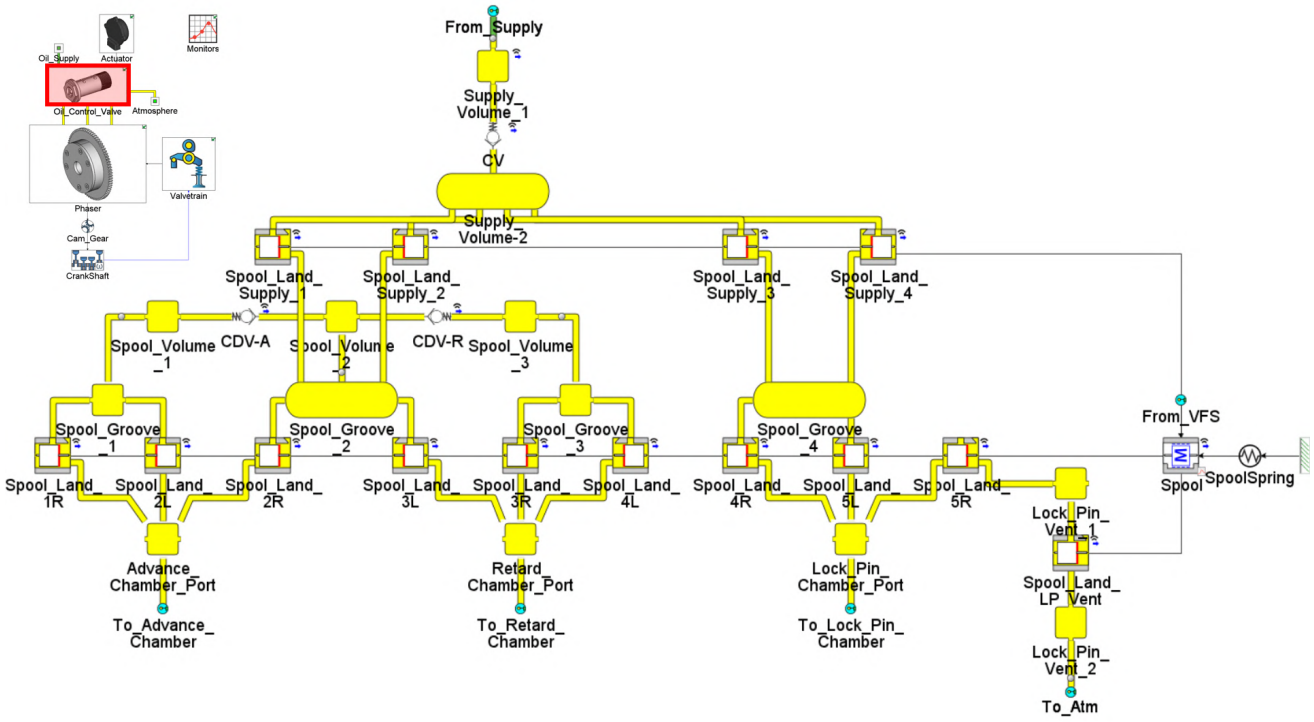


Figure 4.2: Oil Control Valve Model

4.1.1.1 Spool Cover Hole

The Oil Control Valve subassembly model was developed by first defining the spool sections, which control the oil flow based on the spool position. A single *SpoolCoverHole* template represents one edge of the spool metering a circular hole. By combining two templates, both edges of the spool can be represented, as shown in Figure 4.3. This concept is extended to the Oil Control Valve, as shown in Figure 4.4, where the spool sections responsible for metering the oil flow to the chambers are highlighted. The GT-SUITE model is shown on the left, while the corresponding spool diagram is shown on the right.

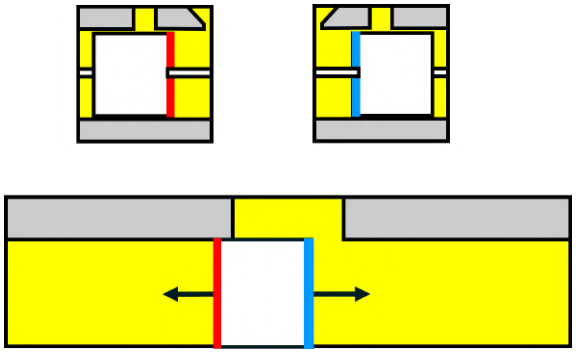


Figure 4.3: Spool Cover Hole

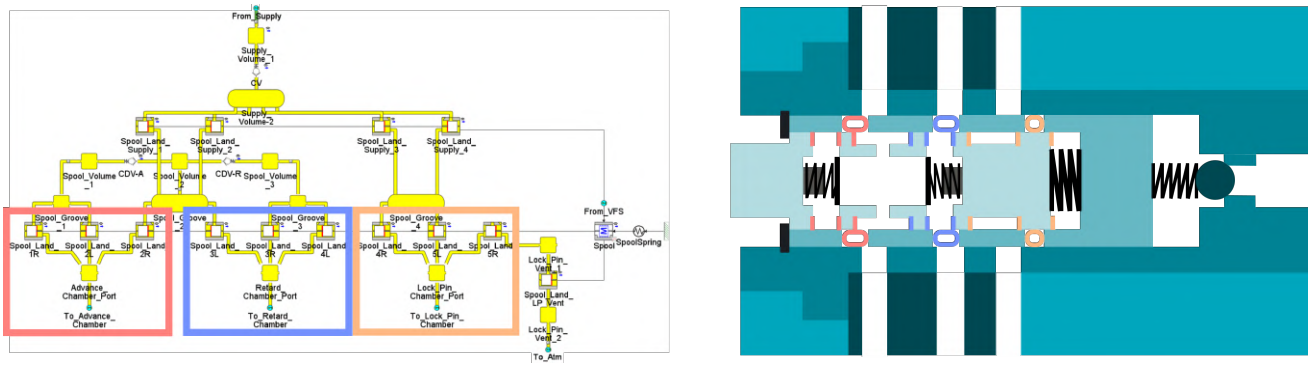


Figure 4.4: Oil Control Valve: Spool Cover Holes

4.1.1.2 Volume Flow Split

The *FlowSplitGeneral* template was used to represent the internal fluid volumes of the Oil Control Valve. The component was segmented into individual volumes based on its internal geometry, and each volume was assigned a name, as shown in Figure 4.5. The connections between these volumes were then identified to represent the oil flow paths through the Oil Control Valve. The boundaries of each volume were defined according to the connections with adjacent volumes and other components in the model. The resulting arrangement of volumes and connections formed the basis for modelling the internal oil flow through the Oil Control Valve.

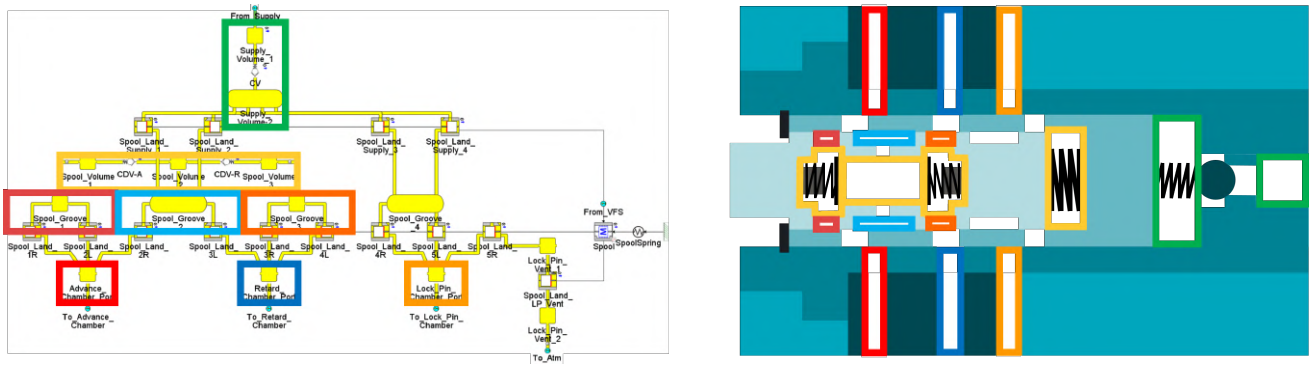


Figure 4.5: Oil Control Valve Volume: Flow Splits

4.1.1.3 Check Valve

The *CheckValve* templates were used to model the check valves within the Oil Control Valve, which prevent reverse oil flow and enable net oil transfer between the chambers, as shown in Figure 4.6.

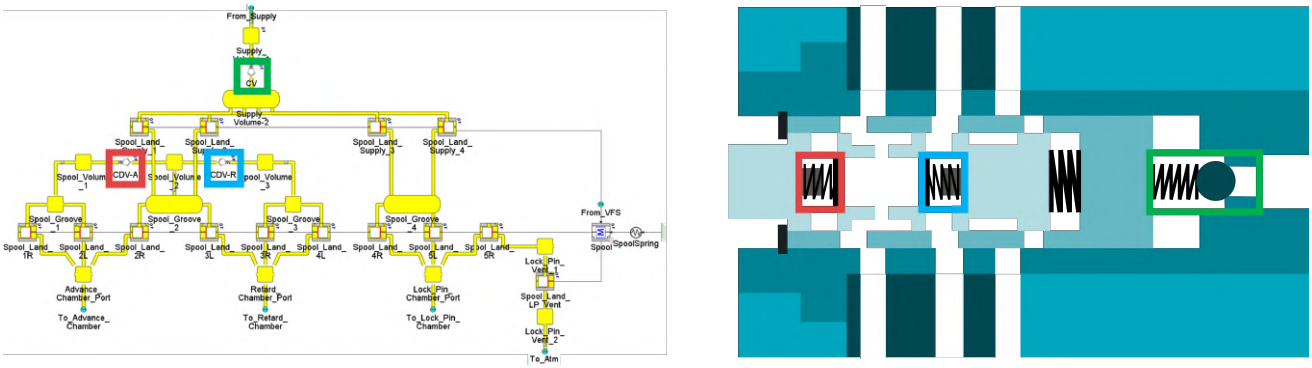


Figure 4.6: Oil Control Valve: Check Valves

4.1.2 Phaser

The phaser model was developed with reference to the *CamPhaser* subassembly in the GT-SUITE example model *CamPhaser-Assy.gtm* [16] and the previous thesis work conducted at Scania by Johan Germundsson (2023) [6]. The main difference in the model developed in this thesis is that it includes a lock pin chamber but does not include the lock pin functionality itself.

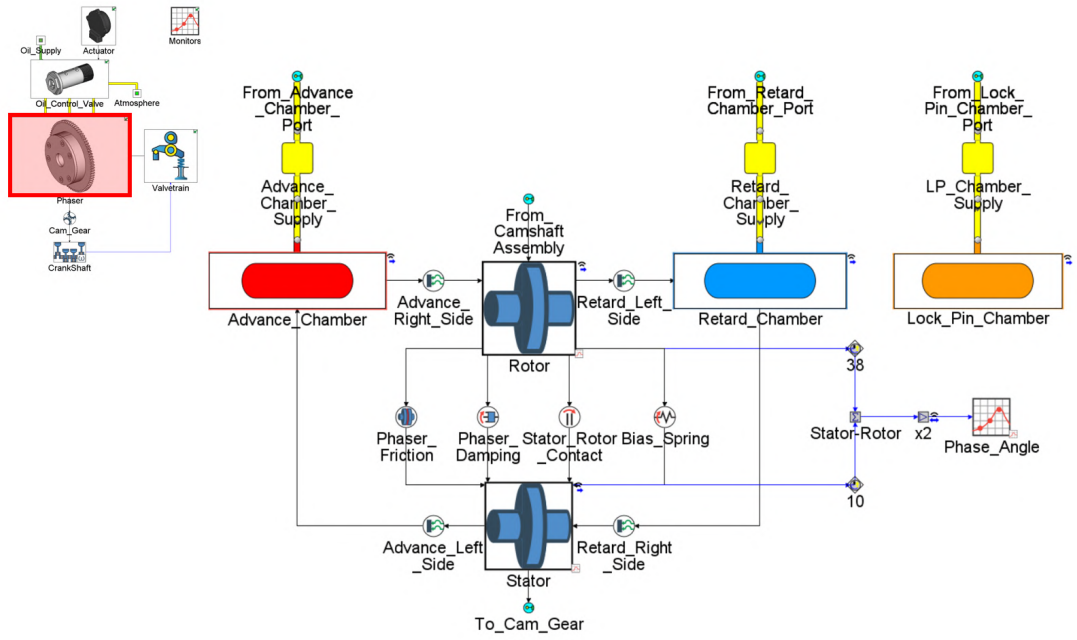


Figure 4.7: Phaser Model

4.1.2.1 Chamber

The advance, retard, and lock pin chambers are shown in red, blue, and orange, respectively, in Figure 4.8. These chambers are modeled as Volume Flowsplits in the phaser model, as shown in Figure 4.7.

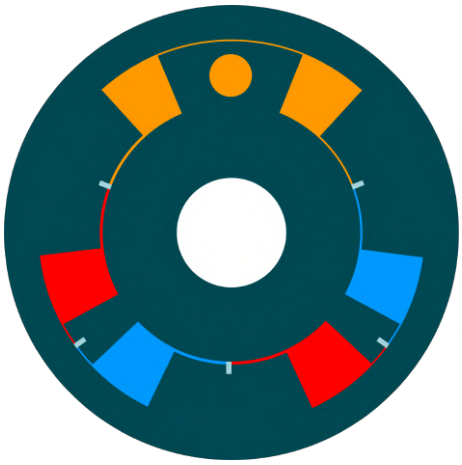


Figure 4.8: Phaser Model

How does the chamber volume change as the rotor moves?
When the rotor moves relative to the stator, one chamber expands while the opposing chamber contracts. The amount of volume displaced for a given rotor movement determines how much oil must enter or leave the chambers. This relationship between rotor motion and hydraulic volume displacement is described by the swept volume.

Swept volume is defined as the change in chamber volume caused by a unit angular displacement of the rotor relative to the stator:

$$V_{\text{swept}} = \frac{\Delta V \text{ [cc]}}{\Delta\theta \text{ [rad]}} \tag{4.1}$$

Since the chamber volume depends on the rotor position, it can be defined as a function of the phase angle:

$$V(\theta) = V_{\text{swept}}\theta \pm V_{Clearance} \tag{4.2}$$

where $V_{Clearance}$ is the clearance volume.

The chamber forms an annular sector, therefore, to derive an equation for the swept volume, the area of a circular sector is first given by:

$$A_{\text{sector}} = \frac{1}{2}r^2\theta \tag{4.3}$$

For an annular sector, the area is obtained by subtracting the inner sector area from the outer sector area:

$$A_{\text{annular}} = \frac{1}{2}\left(r_o^2 - r_i^2\right)\theta \tag{4.4}$$

For one radian, $\theta = 1$, therefore the area per radian becomes

$$\frac{A}{\theta} = \frac{r_o^2 - r_i^2}{2} \tag{4.5}$$

Multiplying the area per radian by the rotor width w_r and the number of active chambers N_{ac} gives the swept volume per radian. If w_r, r_o, and r_i are given in mm, the resulting volume is in mm^3 and is divided by 1000 to convert it to cc:

$$V_{\text{swept,rad}} = \frac{N_{ac}\, w_r \, \dfrac{r_o^2 - r_i^2}{2}}{1000} \qquad [\text{cc/rad}] \tag{4.6}$$

Chamber Volumes
The swept volume is used to calculate the initial volume of the individual phaser chambers based on their corresponding angular positions. Since the angular dimensions are defined in degrees, they are converted to radians by multiplying by $\pi/180$.

The advance chamber volume is given by

$$V_{ac} = V_{\text{swept}} \left(\theta_{\text{clearance}} + \frac{\theta_{\text{init}}}{-2} \right) \frac{\pi}{180} \tag{4.7}$$

The retard chamber volume is given by

$$V_{rc} = V_{\text{swept}} \left(\theta_{\text{clearance}} + \frac{\theta_{\text{ROA}} + \theta_{\text{init}}}{2} \right) \frac{\pi}{180} \tag{4.8}$$

The swept volume V_{swept} represents the combined swept volume of all N_{ac} active chambers. Since the lock-pin chamber represents a single chamber, V_{swept} is divided by N_{ac} to obtain the swept volume corresponding to one chamber.

The lock-pin chamber volume is given by

$$V_{lc} = \frac{V_{\text{swept}}}{N_{ac}} \left(\theta_{\text{clearance}} + \frac{\theta_{\text{ROA}}}{2} \right) \frac{\pi}{180} \tag{4.9}$$

The clearance volume is given by

$$V_{\text{clearance}} = V_{\text{swept}} \theta_{\text{clearance}} \frac{\pi}{180} \tag{4.10}$$

For the advance and retard chambers, the calculated chamber volume represents the total chamber volume of all active chambers. to represent the individual chambers corresponding to each vane the *Number of Identical Flowsplits* option is used. The total volume remains unchanged and is equally distributed among these identical flowsplits. Therefore, the volume assigned to each flowsplit represents the chamber volume per vane:

$$V_{ac,\text{vane}} = \frac{V_{ac}}{N_{ac}} \tag{4.11}$$

$$V_{rc,\text{vane}} = \frac{V_{rc}}{N_{ac}} \tag{4.12}$$

The number of identical flowsplits is then set to N_{ac}.

Supply Volumes
The advance, retard, and lock-pin supply volumes are annular cylinders. Since all three share the same geometry, their volume and surface area are calculated using:

$$V = \pi \left(r_o^2 - r_i^2 \right) w \tag{4.13}$$

$$A = 2\pi \left(r_o^2 - r_i^2 \right) + 2\pi w \left(r_i + r_o \right) \tag{4.14}$$

where r_o, r_i, and w are the outer radius, inner radius, and width of the corresponding supply volume, respectively.

4.1.2.2 Lumped Rotational Inertia: Stator & Rotor

The rotor and stator assemblies are represented using *Lumped Rotational Inertia* templates to account for the rotational inertia of the components connected to each assembly. Instead of modelling the inertia of each component separately, the components rotating together are represented by a single equivalent rotational inertia for each assembly. The inertia assigned to each assembly includes the following components:

- **Rotor Assembly:** Rotor, Oil Control Valve, camshaft endplate, and camshaft.
- **Stator Assembly:** Stator, cam phaser assembly excluding the rotor, and cam gear.

Both assemblies rotate at half the engine speed, as the camshaft completes one revolution for every two crankshaft revolutions. The initial angular position of the rotor is defined by the initial phase angle, while the initial angular position of the stator is set to zero.

4.1.2.3 Stator & Rotor Mechanical & Hydraulic Connections

The mechanical and hydraulic interactions between the stator and rotor are represented using different GT-SUITE templates described in Chapter 3. These connections account for friction, torsional damping, rotational limits, bias spring forces, and the interaction between chamber pressure and rotor motion. The following templates were used to represent these interactions in the phaser model.

Phaser Friction
The *FrictionConnRot* template was used to represent the friction between the stator and rotor. The dry friction coefficient was defined using μ_p, and the equivalent radius for torque calculation was set to the mean radius of the phaser:

$$r_{\text{eq}} = \frac{r_o + r_i}{2} \tag{4.15}$$

Phaser Damping
The torsional damping between the stator and rotor was modelled using the *DamperTorsion* template. The damping coefficient was defined as c_p to represent the resistance to relative rotational motion between the two assemblies.

Stator–Rotor Contact
The *ContactRot* template models the rotational limits between two rotating bodies by applying a contact torque when a rotational limit is reached. In the phaser model, it was used to represent the mechanical end stops between the stator and rotor, with the reference and maximum gap angles both set to ROA/2. The connection multiplicity at both ends was set to N_{ac} to account for the number of active vanes.

Bias Spring
The bias spring is defined using its torsional stiffness k_{bias} and preload.

Vane Interaction
The hydraulic interaction between the chambers and the rotor/stator was modelled using the *MechFlowConnRot* template. The effective vane area was defined based on the radial dimensions of the phaser and the rotor vane width:

$$A_{\text{eff}} = (r_o - r_i)w_r \tag{4.16}$$

The equivalent radius for torque calculation was set to the mean radius of the phaser:

$$r_{\text{eq}} = \frac{r_o + r_i}{2} \tag{4.17}$$

The torque multiplier was set to N_{ac} to account for the number of active vanes contributing to the hydraulic torque. These parameters define the relationship between the chamber pressure and the torque acting on the rotor and stator.

4.1.3 Actuator

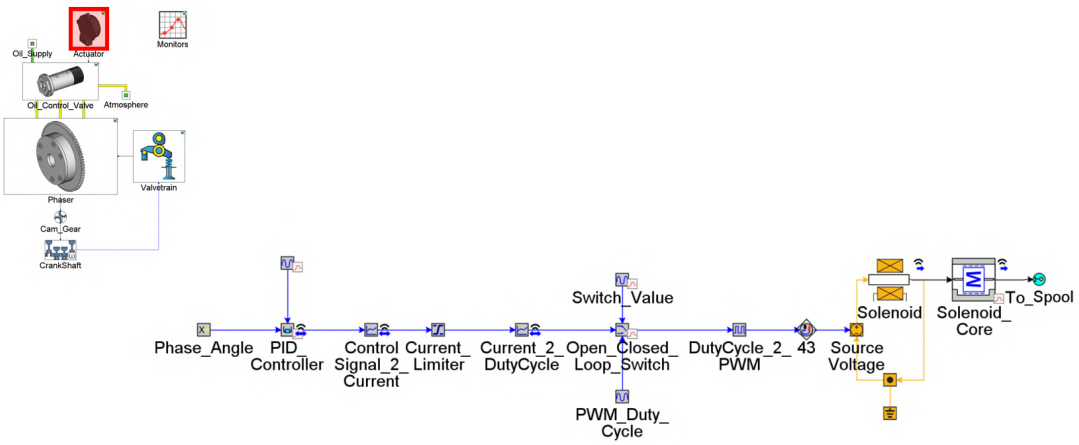


Figure 4.9: Actuator Model

The actuator is modelled using a simple PID controller and a solenoid, since the controller itself was not the focus of this thesis and implementing the actual engine controller was considered outside the scope of the work. This simplification should

be considered when comparing the simulation results with test data obtained using the actual engine management software.

Figure 4.9 shows the actuator model and the signal flow from the phaser feedback to the Oil Control Valve. From left to right, the phase angle from the phaser model is fed to the PID controller, which generates the control signal. This signal is converted to current and limited according to the hardware limits. The current is then converted to a duty cycle and subsequently to a PWM voltage, which drives the solenoid. Finally, the solenoid generates the force acting on the Oil Control Valve.

4.1.3.1 PID

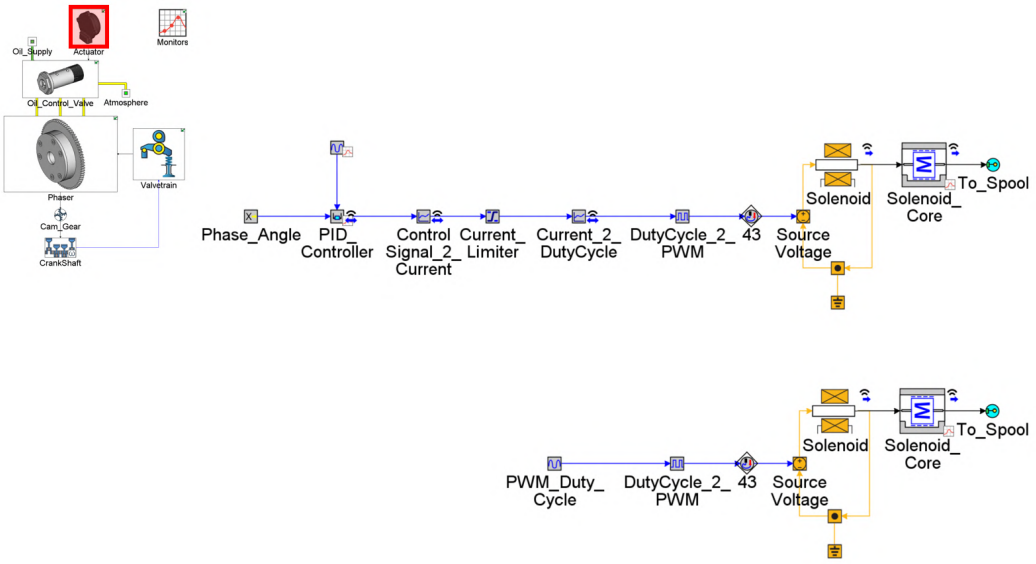


Figure 4.10: Actuator Model

As shown in Figure 4.10, the actuator model can operate in either closed-loop or open-loop control. A switch is used to select between the two control modes. In closed-loop control, shown at the top of the figure, the PID controller determines the control signal based on the error between the demanded and actual phase angles. Since the PID controller is a simplified representation of the actual engine controller, its parameters were iteratively adjusted to approximate the response observed in the test data.

In open-loop control, shown at the bottom of the figure, the PID controller is bypassed and the PWM duty cycle is provided directly as an input. The duty cycle is converted to a PWM signal, which generates the voltage applied to the solenoid.

4.1.3.2 Solenoid

The solenoid was modelled using the *SolenoidMap* template. The model inputs were based on previous thesis work conducted at Scania by Johan Germundsson

(2023) [6], which used supplier data. The main inputs include the coil resistance, air gap, force map, and inductance map.

The solenoid force and inductance are represented using lookup tables as functions of armature position and electrical current, based on supplier data. The force map determines the force generated by the solenoid for a given current and armature position, while the inductance map describes how the inductance varies with these inputs. This approach simplifies the solenoid representation compared with a detailed physics-based model. Since detailed solenoid modelling was not the focus of this thesis, further development of the model was considered outside the scope of the work.

4.1.4 Cam Torque Map / Valvetrain

The cam torque acting on the rotor is represented using a cam torque map obtained from a detailed valvetrain model. This allows the cam torque to be applied based on engine speed and crank angle without simulating the complete valvetrain within the cam phaser model.

4.1.4.1 Valvetrain Model

The detailed valvetrain model used to obtain the cam torque is shown in Figure 4.11. The model represents the mechanical connections between the crankshaft, camshaft, cam lobes, rocker arms, and exhaust valves. The valve spring forces acting through these components generate the alternating cam torque during valve opening and closing. The cam torque contributions from all six cylinders are combined to obtain the total exhaust camshaft torque.

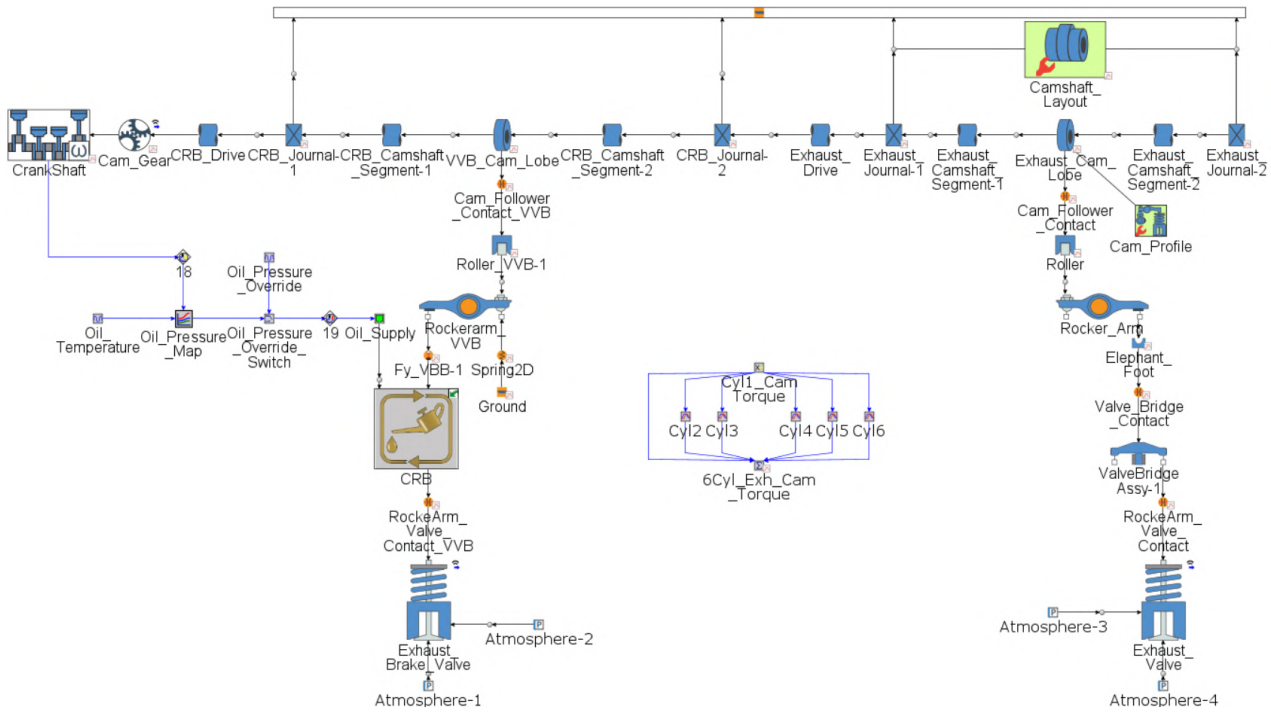


Figure 4.11: Cam Torque Map Valvetrain Model

Figure 4.12 shows the implementation of the cam torque map in GT-SUITE. The map takes engine speed and crank angle as inputs and provides the corresponding cam torque. Since the camshaft rotates at half the crankshaft speed, the camshaft angular position is multiplied by two to obtain the crank angle. The resulting crank angle is expressed within the range of 0° to 720°, corresponding to one complete engine cycle. The resulting cam torque is multiplied by −1 to match the sign convention of the phaser model and is applied to the rotor assembly.

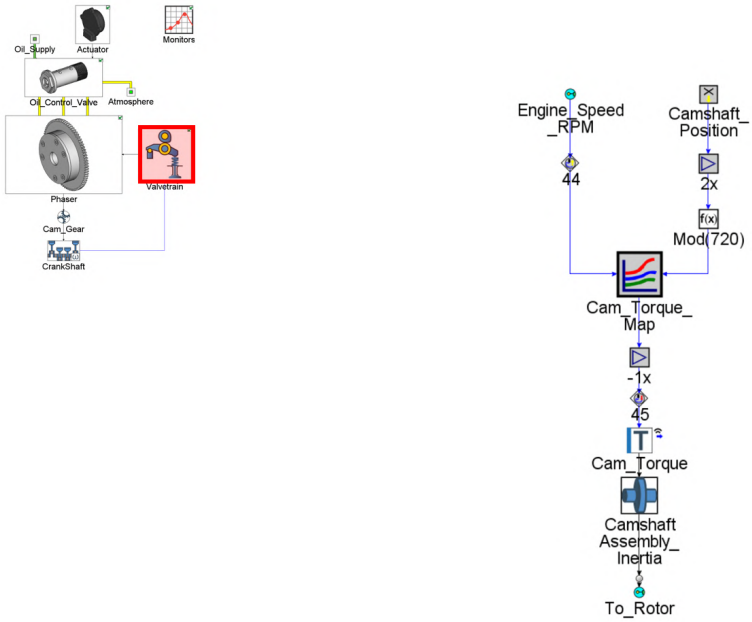


Figure 4.12: Valvetrain Model

4.1.4.2 Cam Torque Map

The resulting cam torque map from the valvetrain model is shown in Figure 4.13. The horizontal axis represents one complete engine cycle of 720 crank angle degrees, while the vertical axis represents engine speed ranging from 500 to 2400 RPM, covering a typical diesel engine operating range. The colour scale indicates the corresponding cam torque.

The map shows six repeating patterns of positive and negative cam torque over one engine cycle, corresponding to the successive valve events of the six-cylinder engine. These patterns repeat approximately every 120 crank angle degrees, with sharp positive torque peaks followed by broader negative torque regions. As explained in Chapter 2, positive cam torque occurs during valve opening when the valve spring force opposes camshaft rotation, while negative cam torque occurs during valve closing when the spring force assists camshaft rotation.

The magnitude and shape of the cam torque vary with engine speed. The positive torque peaks remain relatively similar in magnitude across the speed range, while the negative torque regions become more pronounced at higher engine speeds, as indicated by the darker blue regions near the top of the plot.

The red horizontal line in Figure 4.13 represents the cam torque profile at 1000 RPM, plotted in Figure 4.14. The complete cam torque map consists of cam torque profiles at each RPM, arranged horizontally across the entire engine speed range.

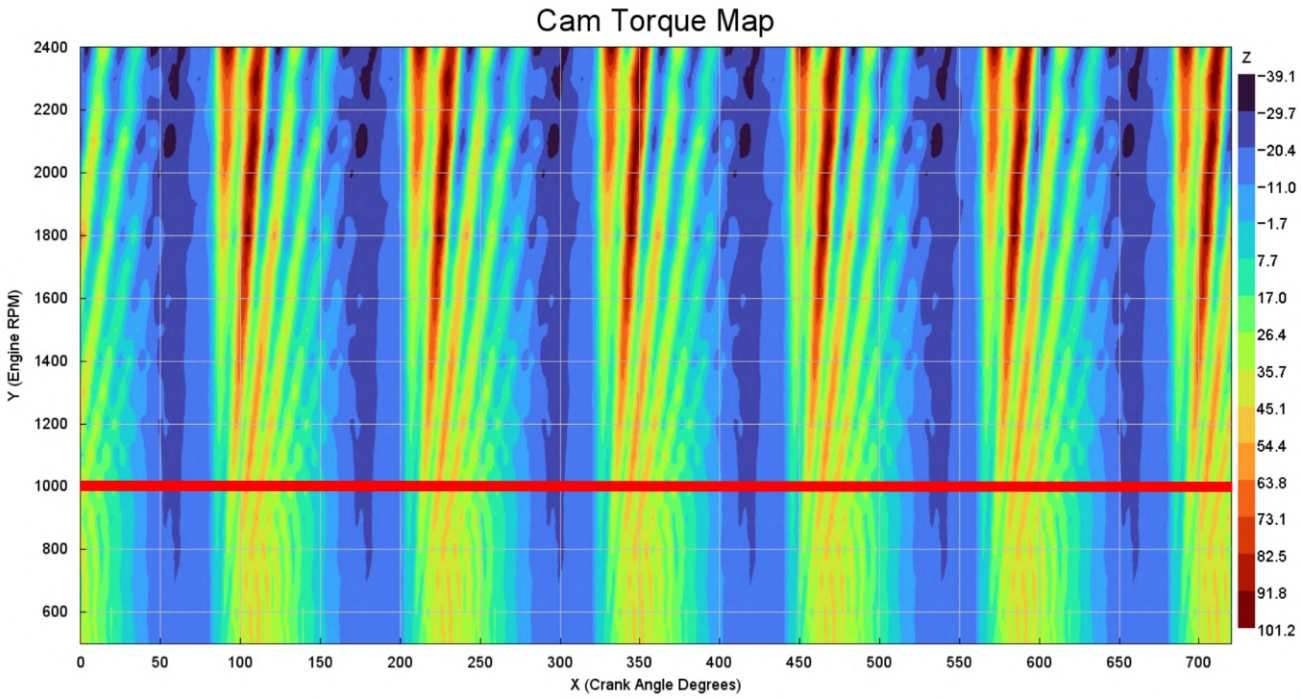


Figure 4.13: Cam Torque Map

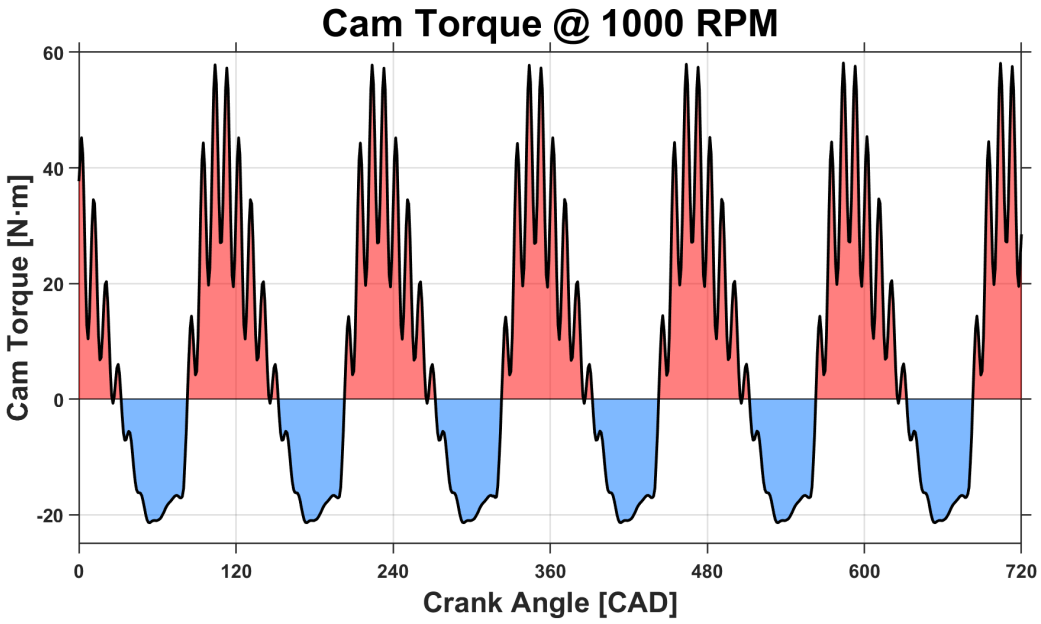


Figure 4.14: Cam torque over one engine cycle at 1000 RPM

4.1.5 Others

4.1.5.1 Boundary Pressure: Oil Supply & Atmosphere

The oil supply and atmospheric boundaries are modelled using the *BoundaryPressure* template. The engine oil composition is given using a representative oil from the GT-SUITE fluid library with appropriate fluid properties. The test cell supplies oil at constant pressure and temperature; therefore, the oil supply is modelled as a constant-pressure and constant-temperature boundary, without explicitly modelling the engine oil supply system or accounting for engine-speed-dependent variations in oil pressure.
The atmospheric boundary represents the surroundings of the cam phaser and is defined using ambient pressure and temperature. The fluid composition at this boundary is defined as air, representing the actual system where oil is vented from the cam phaser.

4.1.5.2 Crankshaft

The *EngineCrankShaft* template is included as the engine speed boundary, providing the rotational input to drive the cam gear and valvetrain. The template is configured as a four-stroke engine, with the engine speed specified according to the simulation setup. Crankshaft inertia is neglected, as crankshaft dynamics are outside the scope of this model.

5
Calibration

5.1 Test Cell

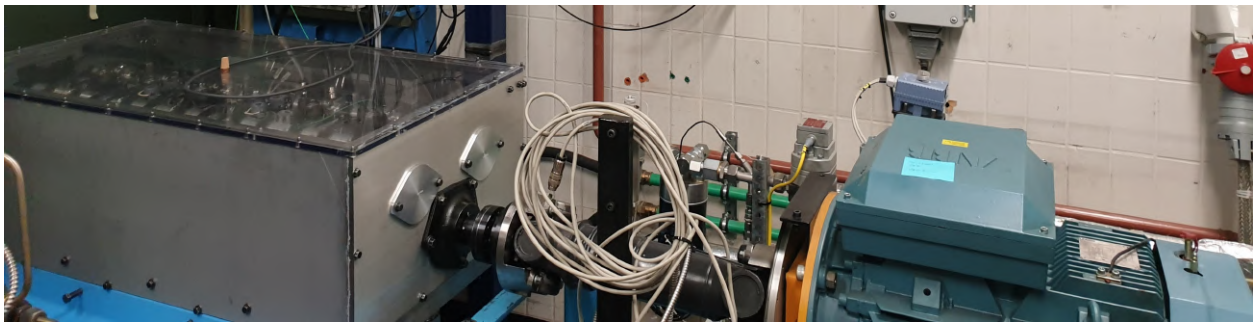


Figure 5.1: Valvetrain Test Cell

The test data used for model calibration was obtained from a component-level test cell specifically designed for testing valvetrain components, as shown in Figure 5.1. This test cell was selected for two main reasons:

1. **Component-Level Testing:** As the focus of this thesis is the development of the cam phaser model, this test cell provides all relevant cam phaser measurements required for model calibration. The test cell contains a cylinder head with the valvetrain driven by an electric motor at a specified engine speed. It represents the operation of the valvetrain and cam phaser system without combustion pressure, with oil supplied at specified pressure and temperature. The model is therefore developed to represent the same test conditions, including the absence of combustion pressure and the controlled oil supply.
2. **Oil Control Valve Displacement Measurement:** The test data includes the Oil Control Valve displacement measured using an LVDT sensor, which is important for model calibration since the actuator controls the cam phaser system solely through displacement of the Oil Control Valve.

5.2 Test Data

The test data used for calibration is divided into open-loop and closed-loop control cases. The open-loop cases are used to evaluate the actuator response independently of the cam phaser dynamics, while the closed-loop cases evaluate the response of the complete cam phaser system under representative operating conditions.

Due to confidentiality, the numerical values of Duty Cycle and PWM Frequency are not presented and are instead represented using qualitative terms.

1. **Open-Loop Control**
   - Boundary Conditions: 0 RPM, 0 bar, 25 °C
   - Duty Cycle: Retard / Hold / Advance
   - PWM Frequency: Low / High
2. **Closed-Loop Control**
   - Boundary Conditions: 1000 RPM, 3 bar, 125 °C
   - Phasing Direction: Advance / Retard
   - PWM Frequency: Low / High

5.3 Open-Loop Control

The open-loop cases evaluate the actuator response. Since the actuator controls the cam phaser system solely through displacement of the Oil Control Valve, reproducing this response is an important step in calibrating the complete cam phaser model.

Figures 5.2 and 5.3 compare the test measurements (red) and simulation results (blue) at Low and High PWM Frequencies, respectively. From top to bottom, the rows represent the duty cycles corresponding to the Retard, Hold, and Advance positions, respectively. From left to right, the columns illustrate the actuation sequence: the input duty-cycle command is converted into a PWM voltage applied to the solenoid, generating the solenoid current and corresponding electromagnetic force. This force moves the solenoid core, which subsequently displaces the Oil Control Valve.

The primary calibration objective is to reproduce the actuator position for a given duty-cycle input. The solenoid is modelled using force and inductance maps rather than a detailed physics model of its electromagnetic behaviour. Therefore, the voltage and current comparisons are mainly used to assess whether the waveform characteristics are reproduced, while exact agreement in the electrical response is a secondary objective. The voltage and current plots are zoomed in to show individual PWM cycles. At this scale, measurement resolution and timing alignment may also influence the comparison.

5.3.1 Low PWM Frequency

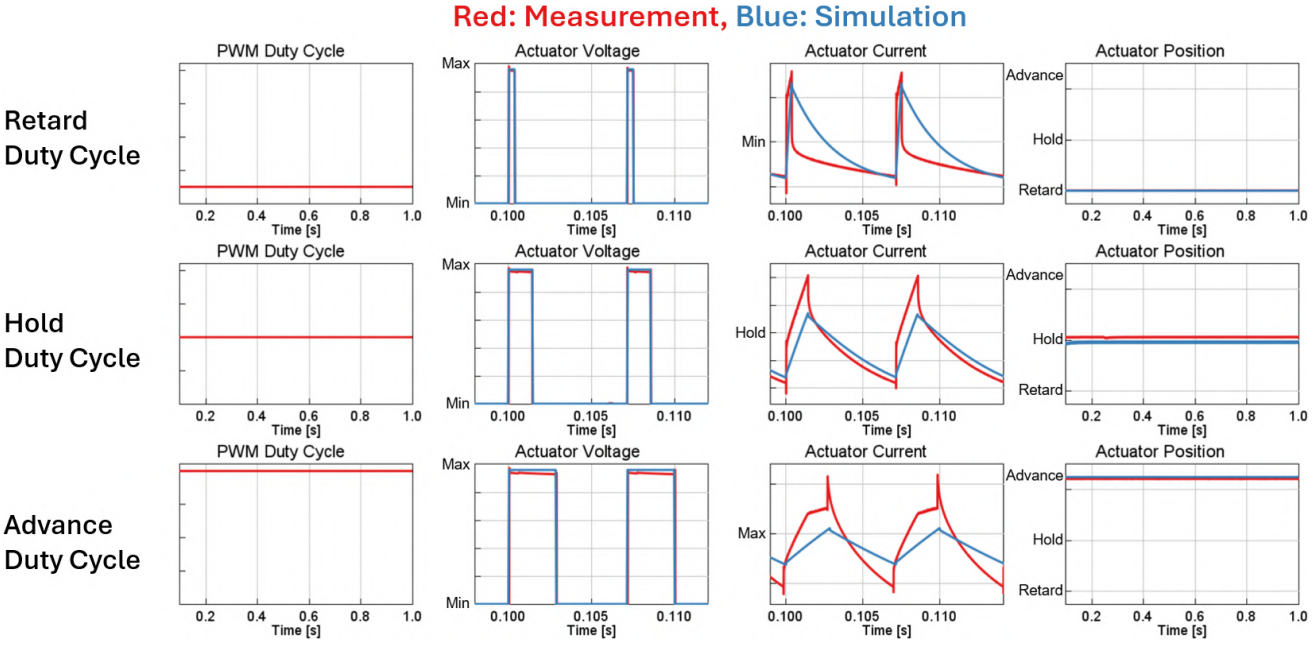


Figure 5.2: Open Loop: 0 RPM, Low PWM Frequency

As shown in Figure 5.2, the lower PWM frequency results in a relatively long PWM switching period compared to the higher frequency, allowing the solenoid current to rise and decay substantially within each switching cycle.

Although the applied voltage pulses agree closely, differences remain in the current peaks and decay profiles. The differences may arise from the simplified solenoid model, which influences how the current rises during energisation and decays after switch-off.

The actuator positions agree closely at the Retard and Advance duty cycles. At the Hold duty cycle, the Oil Control Valve is at an intermediate position that depends on the balance between the solenoid force, spring force, and hydraulic force. Consequently, inaccuracies in these forces may explain the small position underprediction. In comparison, the Retard and Advance positions correspond to the displacement limits of the Oil Control Valve, where differences in the acting forces have little effect on the final position once the valve reaches its mechanical stop. Overall, the position agreement validates the actuator model for the given test data, while the solenoid's electrical response could be further improved.

5.3.2 High PWM Frequency

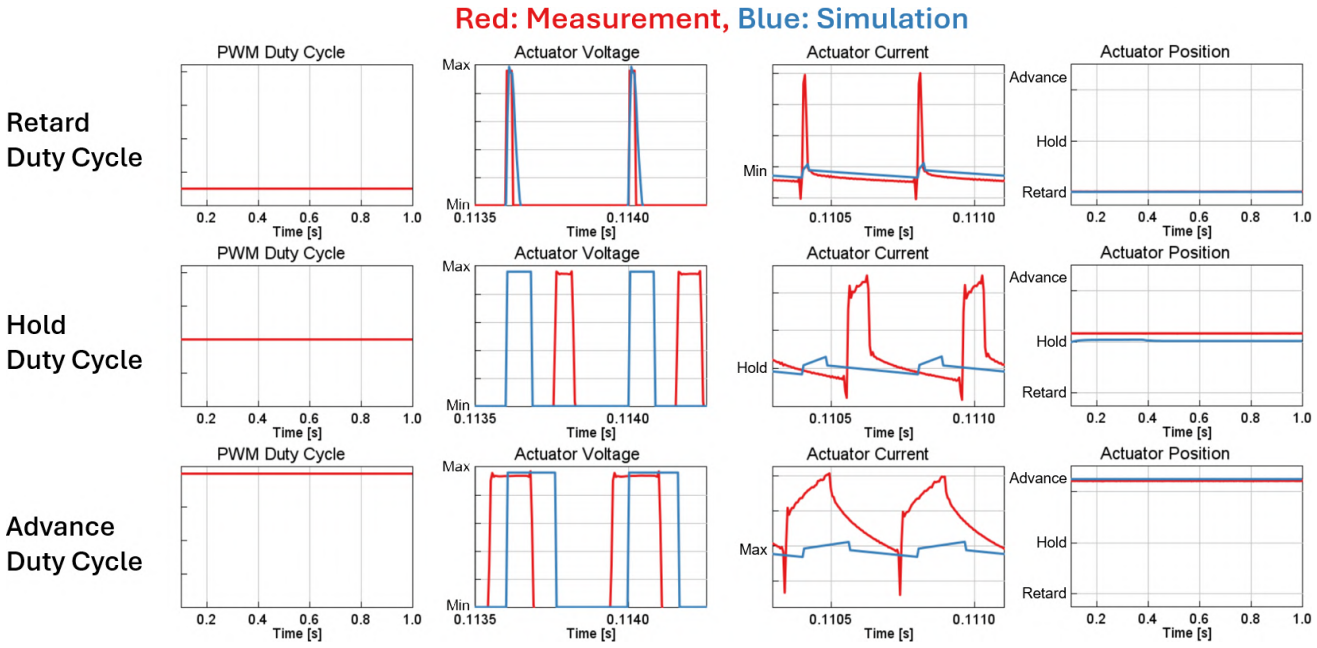


Figure 5.3: Open Loop: 0 RPM, High PWM Frequency

The higher PWM frequency is intended to improve responsiveness and provide smoother current control. As shown in Figure 5.3, increasing the PWM frequency reduces the switching period. The shorter switching period provides less time for the solenoid current to rise and decay within each cycle, resulting in a smoother current response compared to the Low PWM Frequency case. The simulation reproduces this overall behaviour, although differences remain in the current amplitude. Since the current plots are examined over a very short time scale, measurement resolution and timing alignment may also contribute to the observed differences.

Similar to the Low PWM Frequency case, the actuator positions show good agreement, with a small difference at the intermediate Hold duty-cycle position. Overall, the position agreement further supports the validation of the actuator model for the given test data, while the solenoid's electrical response could be further improved.

5.4 Closed-Loop Control

The closed-loop cases evaluate the response of the complete cam phaser system under representative operating conditions. The following points are important when interpreting the closed-loop results:

- **Simplified Controller:** The model uses a simplified PID controller rather than the actual engine control unit used in the test cell. Therefore, an exact match with the test data is not expected; the focus is on evaluating how the complete model responds during phasing.
- **Closed-Loop Compensation:** The closed-loop response reflects the combined behaviour of the hardware, control logic, and controller calibration. The controller can compensate for some modelling inaccuracies in the physical system, meaning that agreement in the phase angle response alone does not necessarily validate the underlying hardware model.
- **Error Propagation:** Small modelling inaccuracies within the actuation chain can affect the phase-angle error and consequently the subsequent controller commands. These differences can therefore propagate through the feedback loop and influence several signals throughout the response.

For these reasons, the closed-loop results are primarily used to evaluate the overall system behaviour, while the open-loop results are more important for calibration of the physical model.

Figures 5.4 and 5.5 compare the test measurements (red) and simulation results (blue) for Advance and Retard phasing, respectively. The demanded phase angle from the engine control unit is shown in green. The left and right columns correspond to Low and High PWM Frequencies, respectively. From top to bottom, the plots illustrate the actuation sequence: the controller determines the duty-cycle command from the difference between the demanded and actual phase angles. The duty-cycle command is converted into a PWM voltage applied to the solenoid, generating the solenoid current and corresponding electromagnetic force. This force displaces the solenoid core and Oil Control Valve, regulating the oil flow to the phaser chambers and producing the resulting phasing motion.

The primary objective of the closed-loop comparison is therefore to reproduce the overall phasing behaviour, including the response time, overshoot, settling behaviour, and final phase angle, rather than achieving an exact match of the individual signals throughout the actuation chain.

5.4.1 Advance

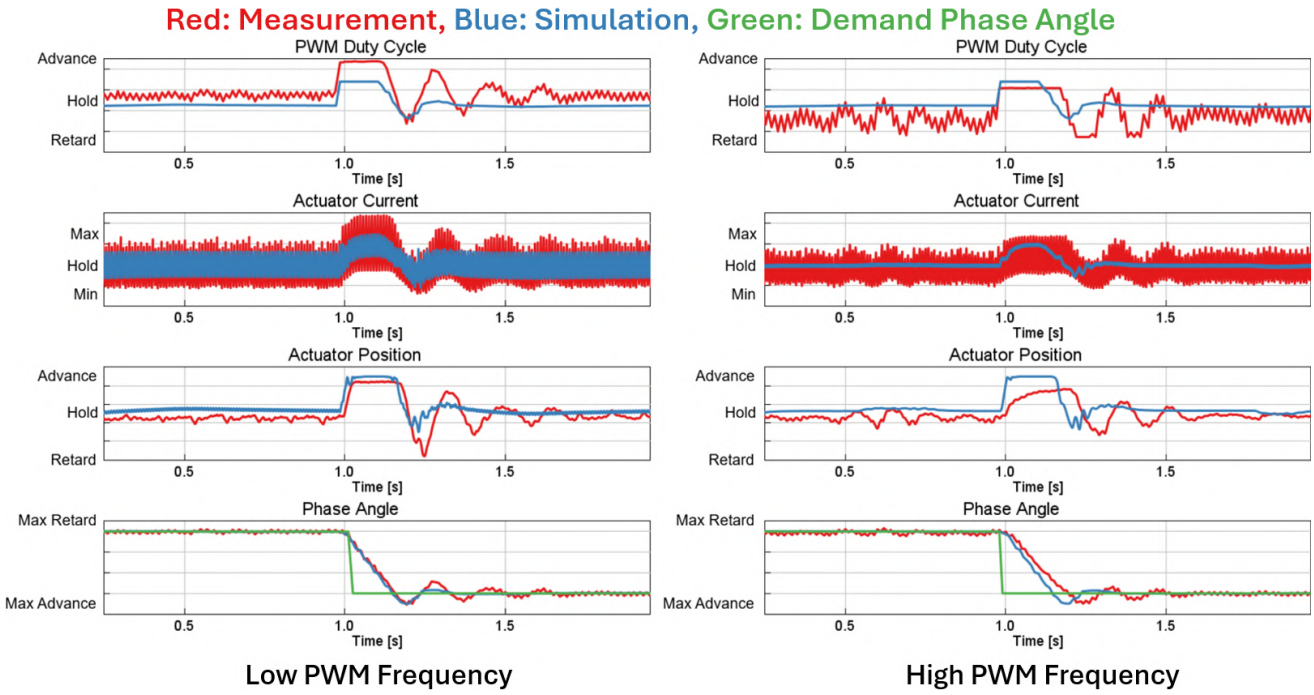


Figure 5.4: Closed Loop: 1000 RPM, Advance

As shown in Figure 5.4, the increase in duty cycle raises the actuator current and moves the Oil Control Valve to the Advance position. The resulting oil flow drives the phase angle from its initial position towards the demanded Advance position. As the phase angle approaches the demand, the controller reduces the duty cycle to slow the motion and return the valve towards its holding position.

At Low PWM Frequency, the simulation closely reproduces the initial phasing rate and first overshoot. However, the measured phase angle exhibits larger subsequent oscillations and takes longer to settle. Similar oscillations are visible in the duty cycle and actuator position, indicating repeated controller corrections as the phase angle oscillates around the demanded value. In comparison, the model exhibits a more strongly damped closed-loop behaviour. This difference may result from differences in the controller.

At High PWM Frequency, the simulated actuator reaches a larger displacement more quickly than the measured actuator, resulting in a faster simulated phasing response. The simulation consequently reaches the demand and overshoots earlier, while the measured response shows more prolonged corrective oscillations.

At both frequencies, the simulated phase-angle response shows good agreement with the test measurements and reaches the demanded phase angle. This demonstrates that the model reasonably captures the overall closed-loop phasing behaviour, with scope for further refinement.

5.4.2 Retard

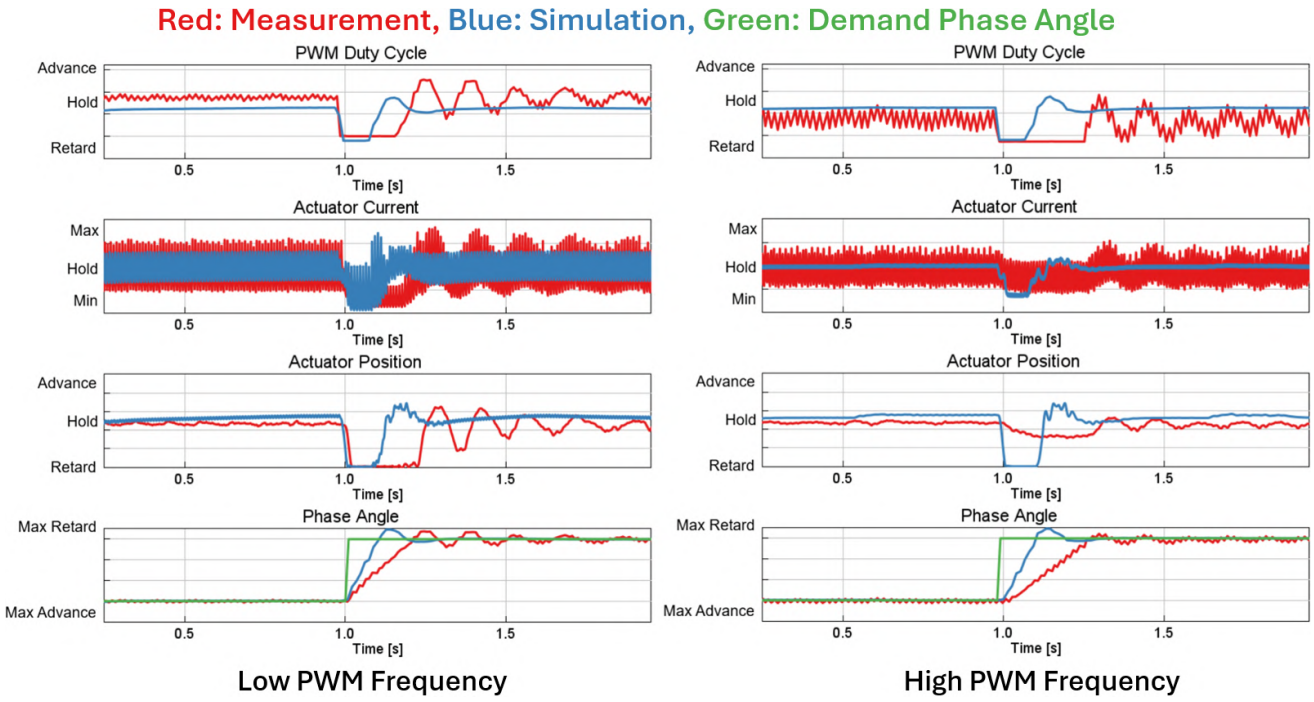


Figure 5.5: Closed Loop: 1000 RPM, Retard

As shown in Figure 5.5, the controller initially reduces the duty cycle, lowering the actuator current and allowing the Oil Control Valve to move towards the Retard position. The resulting oil flow drives the phase angle from its initial position towards the demanded Retard position. As the phase angle approaches the demand, the controller increases the duty cycle to slow the motion and return the valve towards its holding position.

At Low PWM Frequency, the measured and simulated actuator positions show similar behaviour and both initially move towards the Retard limit. However, unlike the Advance case, the simulated phase angle changes noticeably faster than the measurement and reaches the demanded angle earlier. The measured phase angle also exhibits larger subsequent oscillations, while the simulation settles more quickly.

At High PWM Frequency, the simulated actuator reaches the Retard position more quickly than the measured actuator, resulting in a faster simulated phasing response. The simulation consequently reaches the demand earlier, while the measured response takes longer to reach the demanded phase angle and shows more prolonged corrective oscillations. The difference between simulation and measurement is more pronounced than in the corresponding Advance case.

At both frequencies, the simulation reaches the demanded phase angle. However, the transient response is consistently faster than the test measurements, with a larger deviation than observed in the Advance case. The difference between the Advance and Retard responses is likely influenced by the direction-dependent effect of the bias spring.

6
Conclusion

6.1 Overview

The aim of this thesis was to develop an in-house GT-SUITE model of a Cam Torque Actuated phaser to improve the understanding of its internal operation and support further development of the system.

The thesis work focused on three main objectives:

1. **Visualization:** The internal oil flow of the Cam Torque Actuated phaser were visualized in Chapter 2. The flow paths during Hold, Advance, and Retard were illustrated, explaining how the alternating cam torque, oil control valve, and check valves work together to control oil transfer between the chambers and produce phasing.
2. **Modelling:** An in-house physics-based GT-SUITE model was developed in Chapter 4, combining the oil control valve, phaser, actuator, and cam torque into a complete system-level model.
3. **Calibration:** The model was calibrated against test-cell measurements in Chapter 5. The open-loop results showed good agreement in actuator position, with a small deviation at the intermediate Hold position. The closed-loop simulations reproduced the overall Advance and Retard phasing behaviour and reached the demanded phase angles. However, differences remain in the transient response, particularly the faster simulated Retard phasing and smaller oscillations compared to the measurements.

How White Is the Box Now?

Referring to the Black, Grey, and White Box modelling approaches introduced in Chapter 1, the developed model represents a significant step towards a White Box model. Unlike the supplier's Black Box model, the developed model provides access to internal quantities, such as chamber pressures and oil flow rates, that cannot be directly measured in the test cell. This enables further investigation of the cam phaser's hydraulic behaviour and supports future system development.

However, the model is not entirely a White Box, as certain components, such as the solenoid and valvetrain, are represented using lookup tables, while the controller is simplified, as described in Chapter 4.

6.2 Future Work

Further development of the model can be divided into two main areas:

1. **Model Refinement**
   - Investigate the faster simulated Retard response, particularly the possible influence of the bias spring.
   - Further calibrate the solenoid model to improve the agreement in electrical current.
   - Investigate the effects of internal and external oil leakage on phaser response.
   - Implement additional functions, such as the lock pin circuit, and extend calibration to a wider range of operating conditions.
2. **Model Extension**
   - **Valvetrain Model:** Replace the cam torque lookup map with a valvetrain model.
   - **Engine Controller Model:** Incorporate the actual engine control logic to improve the closed-loop comparison and distinguish controller-related deviations from physical modelling inaccuracies.
   - **Combustion Model:** Include the effects of combustion pressure to represent the actual cam torque experienced during engine operation.

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