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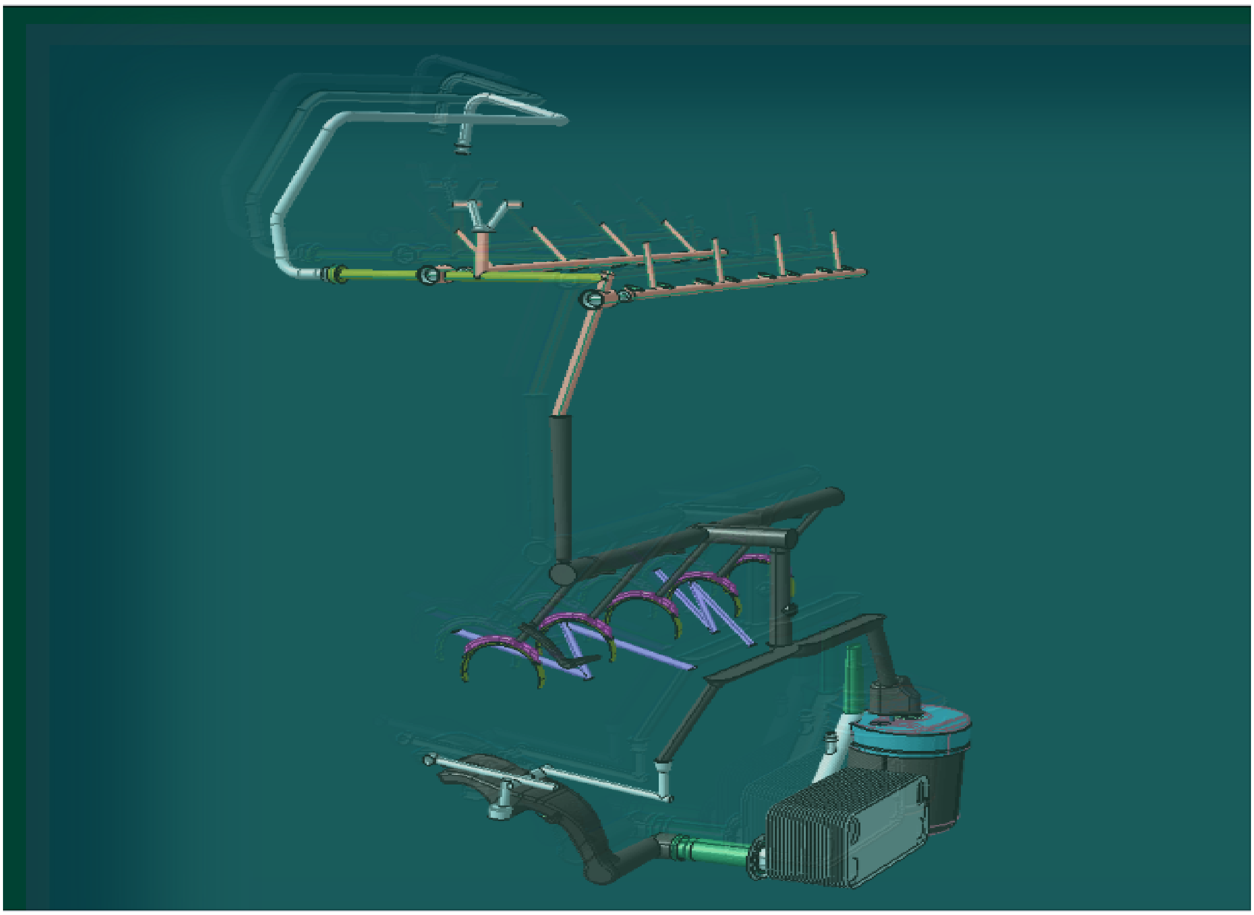

Engine Lubrication System Model Calibration and Fidelity

Master's Thesis in Automotive Engineering

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CHALMERS UNIVERSITY OF TECHNOLOGY
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Abstract

In the present day, tough competition in the automotive industry and shorter timelines for the introduction of new passenger vehicles have forced vehicle manufacturers to cut down on development time. Virtual simulations help in shortening the time vastly and increase efficiency in the development phase avoiding unnecessary waste. The engine is one of the most important subsystems of a car making up the heart of any combustion-driven vehicle, the blood of this engine being the oil, making the lubrication system one of the most critical parts of any engine.
With the help of today's advanced technology, computer software programs like GT-Suite can be used to simulate and predict the oil's behavior inside an engine in a rather accurate way. This report explores the calibration and validation process of a turbocharged four cylinders engine's oil system and analyses its accuracy at steady- state conditions.
A model of the engine's oil system is given and the calibration process begins where nine different calibration parameters, i.e, oil flow rate multiplier, orifice hole diameters, thicknesses, and pressure drop multiplier are used. These are eventually reduced to just two parameters. The calibration process is carried through at a high main gallery pressure mode at temperatures of 60 and 130 °C. Moreover, the system sensitivity is tested where different parameters like the bearing clearances and engine load among others are changed to see their effect on the simulations' accuracy and time. The model is then validated at all pressure levels and different temperatures.
Successful calibration and tuning of the model have yielded highly accurate results in a reasonably short computational time, thus achieving the most relevant part of this project.

Keywords:Accuracy, Calibration, Calibration parameters, Engine, Engine lubrication system, Flow, GT-Suite, Lubrication, Model, Oil circuit, Pressure.

Contents

Preface

This project has been carried out at Aurobay, and has had great assistance from the industrial supervisor, Lars-Olof Carlsson. The examiner, Lucien Koopmans, and the main supervisor, David Sedarsky, have offered important guidance throughout the process. Rodrigo Aihara, and Stephan Weber, at GAMMA Technologies, have offered great support regarding the calibration process. Finally, it is noteworthy to thank Ove Kaldemark at Aurobay who has helped with bearing related questions, and Ulf Christiansson at Aurobay who has provided the CAD model for this project.

Christian Tsobanoglou
Simon Tsobanoglou
Gothenburg, June 2022

Nomenclature

Greek symbol	Description	Unit
α_{inner}	Thermal expansion coefficient of bearing inner material	grad C^{-1}
α_{outer}	Thermal expansion coefficient of bearing outer material	grad C^{-1}
δh	Height change	m
δP	Pressure change	Pa
δv	Fluid velocity change	m/s
η	Fluid viscosity	Nsm^{-2}
ρ	Density	kg/m^3

Roman symbol	Description	Unit
A	Area	m^2
B	Bunsen coefficient	-
C_r	Bearing clearance after thermal expansion	μm
$Corr$	Correlation factor	-
$D_{inner,ref}$	Diameter of bearing inner surface at reference temperature	mm
$D_{outer,ref}$	Diameter of bearing outer surface at reference temperature	mm
F	Force	N
f_D	Darcy friction coefficient	-
f_h	Friction loss	m
f_p	Pressure loss due to friction	Pa
g	Gravitational acceleration	m$^2/s$
h_1	Height at pipe beginning	m
h_2	Height at pipe end	m
L	Pipe length	m
N	Number of measured values	-
P_A	Pressure at pipe beginning	Pa
P_B	Pressure at pipe end	Pa
P_a	Ambient Pressure	Pa
P_{meas}	Measured pressure	Pa
P_s	Relative ambient Pressure	Pa
P_{sim}	Simulated pressure	Pa
Q	Volumetric flow rate	L/min
r	Pipe radius	m
T	Oil temperature	$°C$
T_{ref}	Oil temperature at room temperature	$°C$
V_{air}	Air volume at a pressure of 105 Pa	m^3
v_1	Fluid velocity at pipe beginning	m/s
v_2	Fluid velocity at pipe end	m/s
X	Vector	-
$\overline{X}$	Vector mean value	-
Y	Vector	-
$\overline{Y}$	Vector mean value	-

Acronyms

PCJ	Piston cooling jet
rpms	Revolutions per minute
VVT	Variable valve timing

1 Introduction

Oil to an engine is very vital and makes up one of the most important circulatory systems in an engine. It plays a key role in the engine's efficiency, reliability, and durability, by preventing metal to metal contact in the engine's most critical parts such as bearings, pistons and cam-lobes, creating oil films between critical rotational parts and making sure they're well and sufficiently lubricated. Thus, reducing friction which leads to an improved overall efficiency and longevity. In recent years the term right sizing has been used for designing engines, which means that a smaller displacement is opted for and turbochargers and superchargers are coupled with the engine. Consequently, engines are running at higher temperatures and higher pressures than before. This has increased the demand on an efficient engine lubrication system that lubricates and cool down different components across a wide engine running temperature range.

Aurobay is a company that develops and produces world-class powertrain solutions. Combustion engines are thus one of the products offered by the company. As a key system to the engine, it is essential to have an accurate model of the lubrication system, which correctly predicts the oil behavior inside the engine's circuit.

1.1 Background

The proposed thesis work revolves around the modelling of an oil circuit in GT-Suite. GT-Suite is a software provided by Gamma Technologies and is used for a wide variety of modelling application, for example, flow, thermal and mechanical models. The model received in the beginning, was calibrated and verified against engine-lab data measured in steady-state conditions. However, there is a need to use the model outside the available parameter space and to make it valid in transient simulations. The model does not meet those requirements.

1.2 Problem Statement

The model is heavily dependent on the use of calibration parameters. Also, it is calibrated in such way that at each oil temperature level there exists a set of calibration constants. To make the model more general, the model needs to be further developed so that one set of calibration parameters can account for a wider temperature and pressure range. Thus, a specific methodology has to be set. To do that, a number of questions have to be answered; this is discussed in the subsections below.

1.3 Objective

The aim of this project is to improve the calibration method of the existing steady-state lubrication system model and assessing its accuracy to make it more general and inclusive for the engine's different operating speeds, oil pressure levels, and temperatures. This is achieved by making the model less dependent on the use of calibration parameters. This will then allow the model to be valid outside of the parameter space that is used by the measurement data and ultimately make it ready for transient simulations. The project is also done with the overall objective to create a documented calibration procedure. The secondary purpose is to have a model that is as accurate as possible while keeping the computational time needed for the simulation under control.

1.4 Model accuracy and detail level

An important aspect when modelling, is what level of accuracy should be expected from the model, which allows the designer to know how good it is. Thus, the model accuracy must be calculated, in order to know how well the model represents the reality when comparing the simulated results with test data.
The next question to be tackled is how detailed each component should be modeled. This is directly related to the model accuracy as the answer to the question will bring knowledge about what components to be modelled in a detailed way and what components to be modelled in a simple way.
At last, the model should have an accuracy of more than 90% at the same time while not costing more than around one hour of computational power. In other words, the relative deviation between the simulated and measured values should be less than 10%, and the time to run the model for a specific case should not exceed one hour. Thus, it is important to observe the relationship between the accuracy and the simulation run-time. The question then becomes whether the gain in accuracy is worth the extra processing power. Naturally, the accuracy gain and run-time have to be quantified as well.

1.5 Limitations

As with many projects, there exists a few limitations related to this project as well. As mentioned earlier, the calibrated model should not be computationally heavy requiring several hours to run. The upper limit for this simulation time is, however, not clearly defined by Aurobay as of now. Moreover, no flow measurements exist for this work, thus, the comparison between simulation results and test results will only concern oil pressure measurements. It is also worth mentioning that the focus in this work lies mainly on the modelling part and not as much on the fluid properties and equations that affect the flow of the oil in the circuit.

1.6 Engine lubrication system

An engine consists of several moving parts that move against each other at high speeds. A warm engine usually operates at 90 °C, however, during the warm-up phase, it can run at temperatures as low as -40 °C for periods of time and reach temperatures as high as 130 °C. The metal-to-metal contact creates friction, which in turn creates heat. Heat temperatures that arise inside an engine are capable of melting the metal parts and causing them to bond together. That's where lubrication comes into play; by forming a thin film between the moving parts, it greatly reduces this friction, thus allowing the parts to move and rotate easier, hence reducing power loss and decreasing engine wear. This results in increased efficiency, as more energy is transferred to mechanical work. It also improves the reliability and durability of the engine as the internal parts are subjected to less stress with the decreased engine wear. Lubrication also provides a cooling effect to the engine; with the oil continuously circulating inside the engine, it absorbs a part of the heat produced. The lubrication system also has a sealing effect; it seals the gap between the cylinder wall and the piston, thus preventing gases escaping from one of the cylinders into the crank case. Finally, it also removes combustion residues like carbon and contaminants thus protecting the engine from knocking and increasing its lifespan [1].

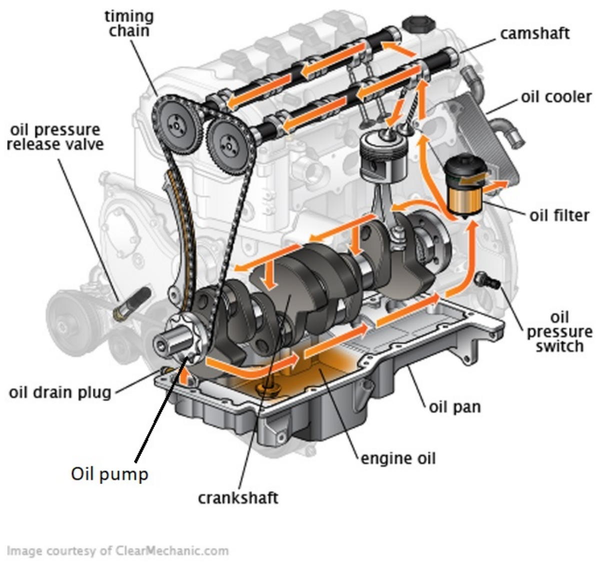

Figure 1: Schematic of engine lubrication system

The engine lubrication system is a closed network of pipes, holes, bores, and bends that provide the necessary oil to lubricate major engine components all of which contribute to a more efficient engine. As shown in figure 1 above, the oil reaches the different components with the help of a pump connected to the engine crankshaft. The pump sucks the oil from the oil sump and pumps it up through the oil cooler which cools down the oil if needed. The oil passes then through the filter that filters out tiny harmful particles for the engine. Next, the oil moves to the main gallery and from there it feeds the main and big-end bearings, the fuel pump, intake- and exhaust cam bearings, VVT, and finally the Turbo. The oil is then drained back and ready to be re-circulated.

1.7 Lubrication system components

In this section, the different components of the oil circuit will be explained.

1.7.1 Oil sump

Figure 2: Oil sump

The oil sump, as shown in figure 2 is located under the crankcase of the engine. Its main job is to store the oil that circulates in the lubrication system. Another main function of the sump is to reduce oil foaming which is when air bubbles are mixed in the oil. It also helps prevent oil splashing with the help of its design that usually incorporates baffles when the vehicle is subjected to sudden movements like acceleration, deceleration, and going through sharp corners [2].

1.7.2 Oil pump

The oil pump is the device responsible for ensuring oil delivery to all the parts that need lubrication. There are mainly two types of oil pumps: Gerotor pumps and External gear pumps. Figure 3 shows an example of an external pump that uses two spur gears to create suction and move the oil from the inlet side to the outlet side.

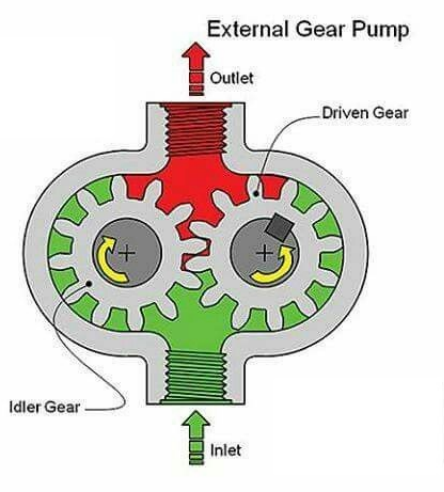

Figure 3: External pump

Figure 4 shows an example of the gerotor pump which features trochoid gears.

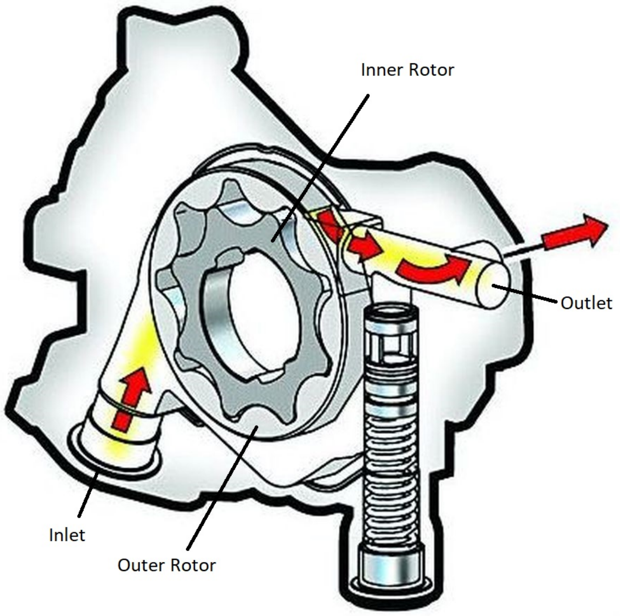

Figure 4: Gerotor pump

The inner rotor is driven by the crankshaft and it drives the eccentrically mounted outer rotor in its turn. Since the two rotors have different rotating axes, space is being constantly created alternatively on either side of the driving gear. One side is the suction side and the other is the pressure side. This allows the oil to be sucked from the sump and into the oil circuit.
Furthermore, pumps can have a fixed capacity or a variable capacity. Fixed capacity pumps produce a flow that steadily increases with engine speed. Variable-capacity pumps, shown in figure 5, can vary the flow rate to keep a constant pressure level in the main gallery regardless of the engine speed and oil temperature.

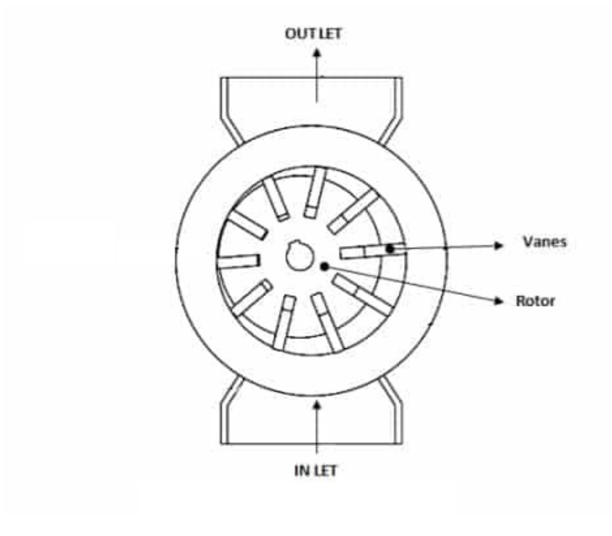

Figure 5: Variable capacity pump

This is achieved by having variable-length teeth called vanes, on the inner gear which changes the rotating axis of the outer gear, thus changing the volume of the oil being sucked. The pump used in this thesis work by Aurobay is a variable capacity pump that can deliver different levels of pressure: low (1.8 bar), medium stage 1 (2 bar), medium stage 2 (3 bar), and high pressure (3.7 bar) expressed in a relative way. Depending on the piston temperature, engine load, and speed, the pump gets actuated and adjusts its flow to deliver the target pressure at the main gallery[3].

1.7.3 Oil cooler

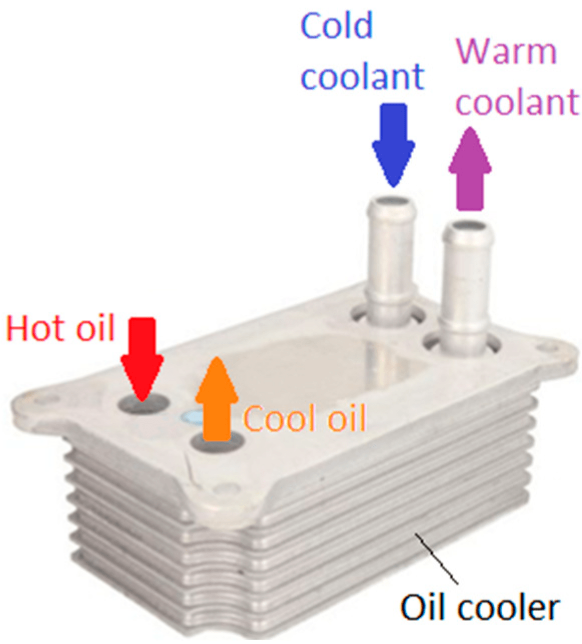

Figure 6: Oil cooler

The oil cooler is the component after the oil pump that cools the oil when needed and prevents it from reaching a maximum temperature that is usually specified by the manufacturer. It does that by taking away the oil's heat and exchanging it with the coolant as shown in figure 6. It should be noted however that the cooler does not reduce the oil's temperature if it is lower than 90 °C, if it's higher than the cooler starts to reduce the oil's temperature, the higher it is, the higher the reduction is.

1.7.4 Oil filter

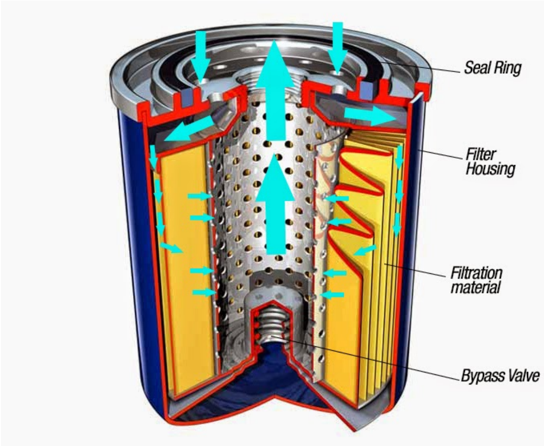

Figure 7: Oil filter

The oil filter is a critical component in the lubrication system. It helps filter out tiny particles that are harmful to the engine, and cause it to wear faster. The oil enters via the base plate of the filter through perforated holes along the perimeter, as shown in figure 7. It then gets filtered with the help of synthetic fiber inside the filter and then comes out through the center hole. In case the filter gets clogged, a bypass valve exits to permit the oil to continue flowing past the filter [4].

1.7.5 Main gallery

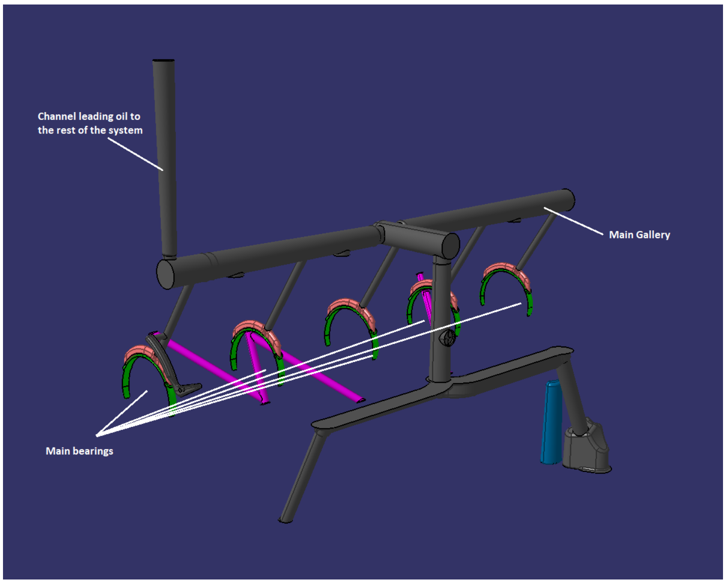

Figure 8: Main gallery

The main gallery is an oil channel that lies in the engine block, and is the main oil distributor to the main bearings, big-end bearings, cylinder head, PCJs and all the remotely located components in the engine, as seen in figure 8 [5].

1.7.6 Crankshaft, main and big-end bearings

The crankshaft is the main shaft in the engine that transmits the combustion force of the pistons to work that moves the vehicle. In other words, it transmits the linear motion of the pistons into rotary motion. As figure 9 shows, connecting rods sit on the big-end bearings that sit on the shaft in their turn. The shaft is supported by five main bearings.

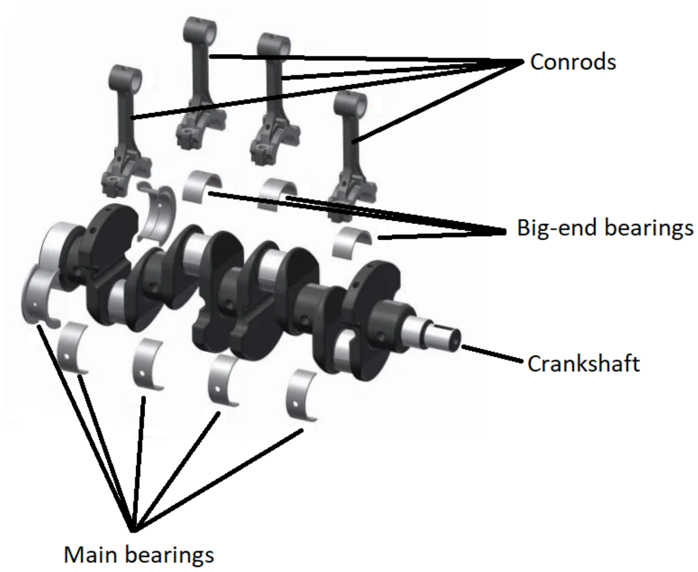

Figure 9: Crankshaft and bearings' position

Main bearings are what holds the crankshaft in its place and allow it to rotate. In total 5 main bearings are used in a four-cylinder engine with two at the beginning and end of the crankshaft and 3 in between the connecting rods.
The type of bearing used, as seen in figure 10, is a journal bearing which consists of two semi-circular metal shells that encompass the crankshaft axis as shown in figure 10. This forms a ring that slides around the axis with the help of an oil film in between the metal surfaces that prevents metal-to-metal contact. Lubrication is made possible with the help of several holes that run through the ring and lead the the oil into a groove along the centre of the ring as shown in figure 11.

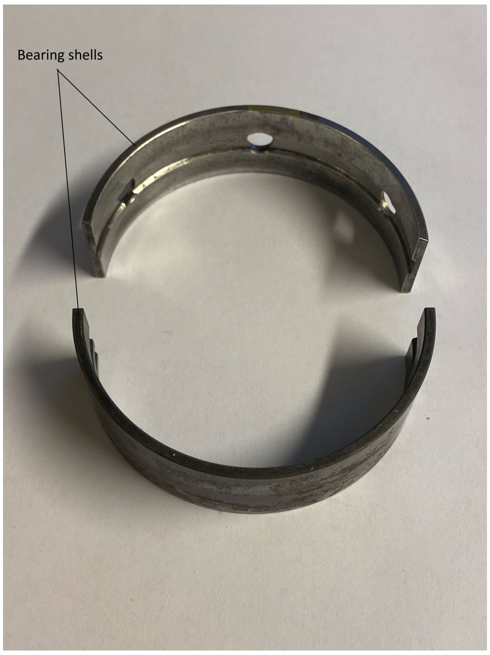

Figure 10: Journal bearing (main)

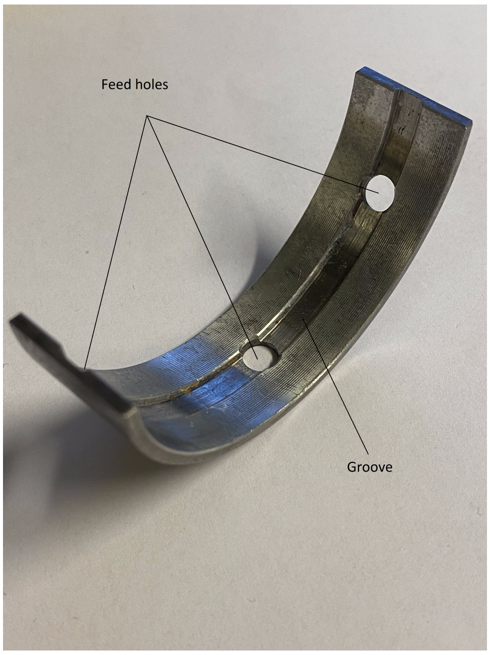

Figure 11: Main bearing groove

Big-end bearings sit on the large-end part of the connecting rods and there are 4 such bearings in total, corresponding to the number of pistons. These bearings are similar to the main bearings but they only have a single feed hole which is cut through the ring and no groove exists. Lubrication is made possible with the help of holes through the crank pins and drilled channels inside the crankshaft. These channels lead the oil to the big-end bearings as figure 12 shows [6].

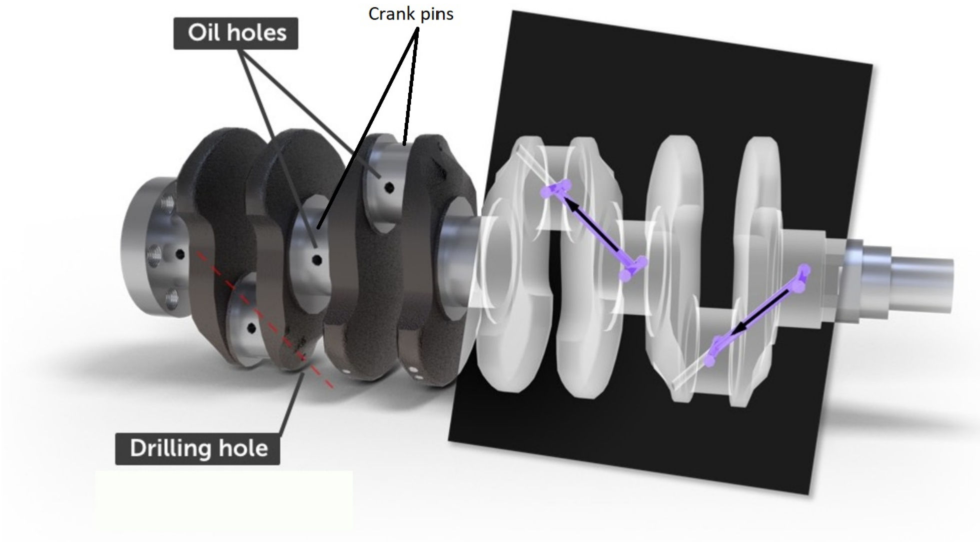

Figure 12: Crankshaft holes

In general, bearings' lubrication in the engine can be categorized in three types of lubrication which are best described and illustrated using a Stribeck curve as seen below.

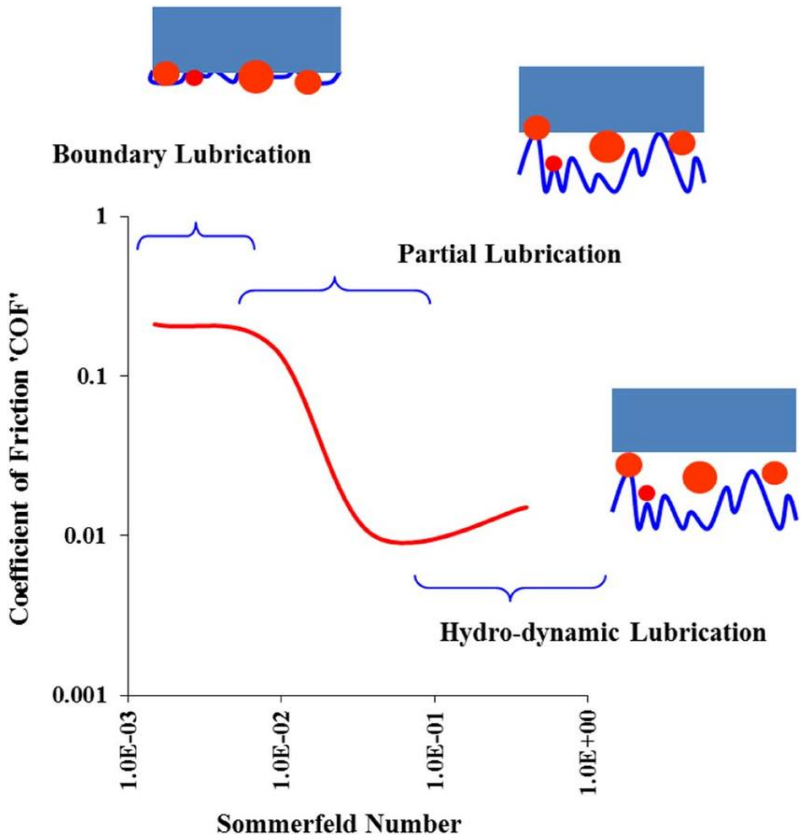

Figure 13: Stribeck curve

As it can be seen in the Stribeck curve, the y-axis quantifies the friction in a value from 0

to 1 called coefficient of friction or COF, while the x-axis shows the Sommerfeld number, a dimentionless number, which is a product of velocity and viscosity divided by load. As seen in figure 13 there are three main types of lubrication; Boundary, Partial and Hydro-dynamic. Boundary lubrication is a highly undesired state where there is high contact area between the bearings and journals, amounting to high friction, or a COF value greater than 0.1. On the other hand, Hydro-dynamic lubrication is the optimal and desired state for the bearings, where the surfaces are completely separated by the oil-film so contact between surfaces is minimal to non-existent and friction is very small, mostly viscous, having a value that can be as low as 0.01. Partial lubrication on the other hand is, as the name states, a state between the boundary and hydrodynamic lubrication. Meaning that contact between surfaces still exist, especially contact between high or peak asperities, however an oil film is also present between at some spots. Friction can range from low to relatively high in partial lubrication and is therefore also not desired.
Generally speaking, the friction and type of lubrication are a function of the amount of load present on the bearings, and the available oil pressure and velocity between the surfaces of the journals and bearings. In high load cases, a high pressure and velocity oil flow is desired while in mid to low torque cases, less pressure and flow is required.

1.7.7 Piston cooling jets

A piston cooling jet is a nozzle located below the piston that squirts jets of oil to cool the underside of it, as shown in 14. Its main function is to cool down and lubricate the piston walls, chamber walls, and the inner parts of the piston where it's connected to the rod. The squirted oil absorbs the heat from the piston, thus protecting it from overheating. It is usually actuated by a valve at a specified oil pressure level that opens at a certain engine speed and load according to a speed and load map.

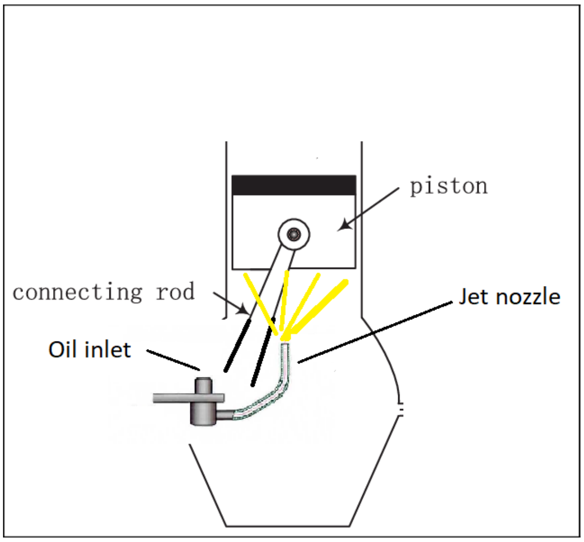

Figure 14: Piston cooling jet

1.7.8 Fuel pump

The fuel pump is a pump that pressurizes the fuel and provides it to the combustion chamber. Since it runs at very high pressure levels, it gets subjected to a lot of heat, and thus needs lubrication for its internal parts to keep functioning properly. In this thesis work, no channel exists inside the pump where the oil can travel, instead, the oil is sprayed under pressure onto the fuel pump and crankshaft rotational mechanism.

1.7.9 Turbo

Turbo is a device that compresses air into the engine into the combustion chambers, allowing more fuel to be mixed, thus increasing power and fuel efficiency. Physically it consists of a turbine and compressor connected by a shaft that is supported by journal bearings.
Figure 15 provides a cross-sectional view of the turbo. The housing of the turbo guides the exhaust gases from the engine to the turbine and spins it. The turbine in its turn spins the compressor which draws air from the atmosphere, pressurizes it, and forces it into the engine. The turbine rotates to speed upwards of 150 000 rpms and the housing can reach temperatures upwards of 200 C. Lubrication is therefore crucial in the turbo to avoid metal-to-metal contact since the clearances in the bearings are extremely small and for cooling the turbo by taking away the heat accumulated. Oil is fed into the housing to lubricate the bearings and the shaft.

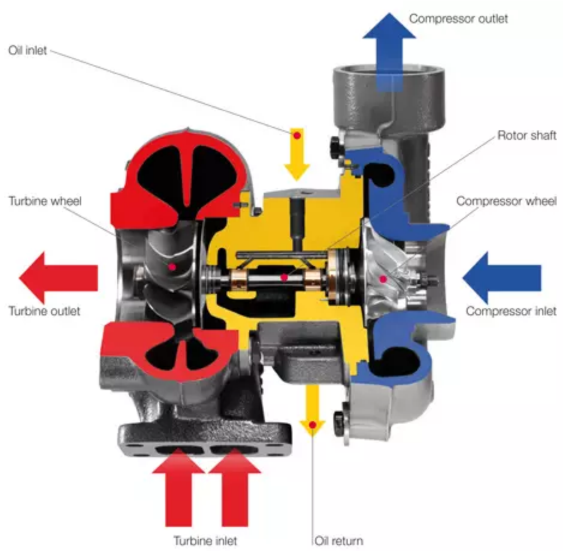

Figure 15: Turbo

1.7.10 Intake and exhaust camshafts and VVT

Valves are mechanical components that open and close to let the air-fuel mixture into the engine and let out the exhaust gases. What controls the valves are cam lobes, shown in figure 16, that sit above the valves and dictate the valve timing when the valves open and close, valve duration, how long the valves stay open, and valve lift, how deep the valves enter the combustion chamber. Cam lobes sit in their turn on the intake and exhaust camshaft, located at the top of the engine in the cylinder head. They are responsible for opening and closing the intake and exhaust valves while rotating. They rotate at a speed that is half the crankshaft speed.

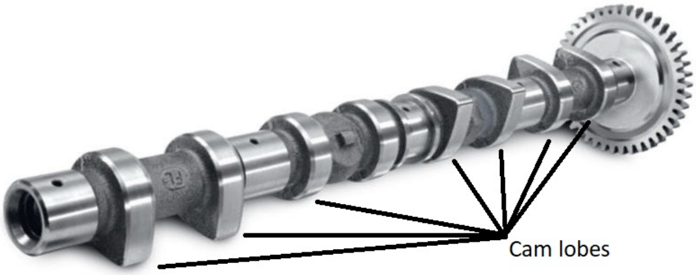

Figure 16: Camshaft

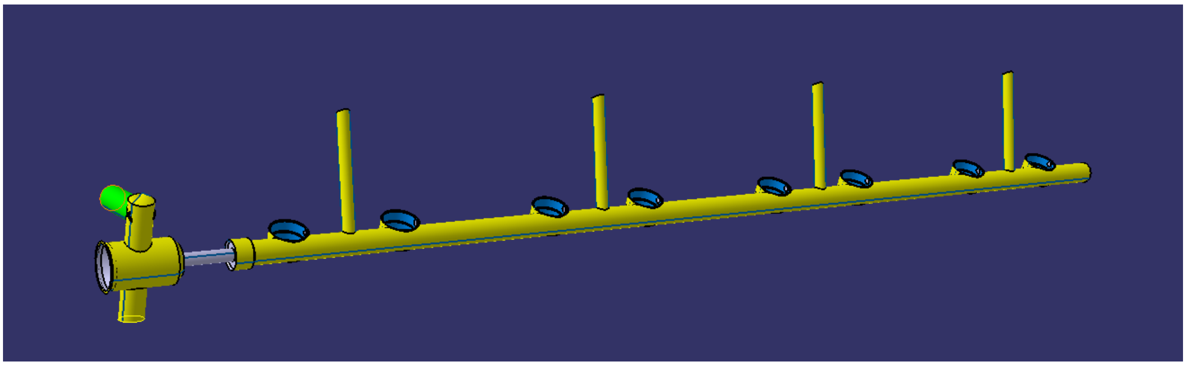

Figure 17: Cylinder head oil gallery

Camshaft lubrication can differ from one engine to another. In this thesis work, oil channels that are drilled through the cylinder head exist to allow the oil to pass as shown in figure 17. The oil is then led through narrow channels up to the camshafts to lubricate the cam bearings. During different engine operating conditions, different cam phases, durations, and lifts might be desired. Valve timing represents the crank angle degree at which the valve opens and closes while valve duration is the amount of time that the valve stays opened or closed. Lift is the amount that the cam lobe moves from its resting position. For the engine considered in this work, only different cam phases can be achieved. That is done with the help of VVT which is short for variable valve timing. This type of VVT is of the cam phasing type which only allows for earlier or later valve opening and doesn't vary the duration of valve opening. Physically it consists of an inner and outer rotor as shown in figure 18. The camshaft sits at the centre hole of the inner rotor. Oil is fed through the oil galleries applying pressure on the inner rotor which then forces it to rotate and thus changes the valve timing.

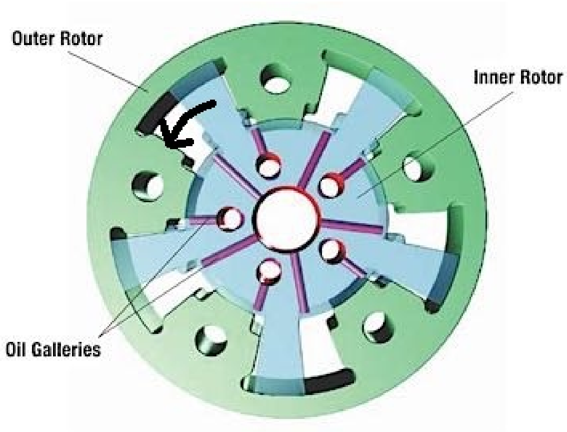

Figure 18: VVT

1.7.11 Cam bearings

Similar to main and big-end bearings, cam bearings are journal bearings that provide support to the camshafts and allow them to rotate as shown in figure 19. What differentiates them from the former bearings, however, is that no shells exist. Instead, a cylindrical hole is drilled through the cylinderhead creating a space between its inner wall surface and the camshaft. This space will constitute the clearance. The inner side of the hole will constitute the shell.

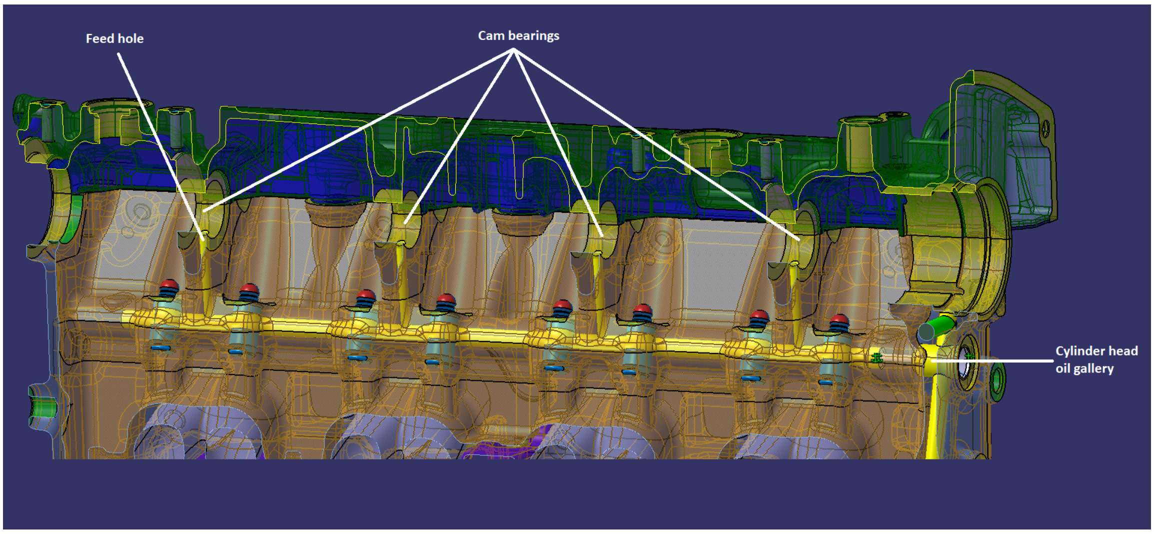

Figure 19: Cam bearings

Four single feed holes exist in the cylinder head, allowing the oil to enter and form a film that fills out the clearance between the hole's inner surface and the journal.

1.7.12 Hydraulic lash adjusters

Hydraulic lash adjusters, shown in figure 22, are small devices that are designed to eliminate any space that might arise at the valve's inlet port. The reason they are used is that the valves' material can expand due to the heat from the combustion. This creates a space at the valve's opening and the combustion chamber won't be completely closed anymore. The way the lash adjusters operate is that they use oil pressure to activate a spring that allows the adjuster body to expand and maintain zero valve clearance. This is shown in figure 20.

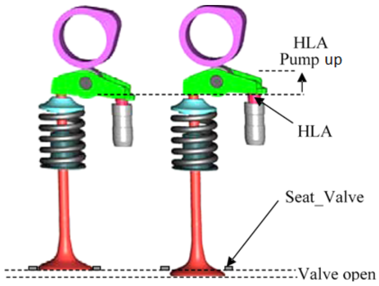

Figure 20: Hydraulic lash adjuster's function

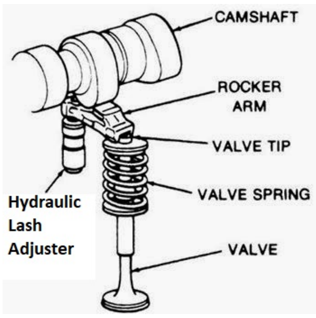

Figure 21: Hydraulic lash adjuster

1.7.13 PCV system

A simple but quite important and overlooked system, is the positive crankcase ventilation system. A system which allows the ventilation of the build-up gases thus allowing the engine to breathe. During combustion, a significant amount of the post-combustion gases such as exhaust fumes and unburnt fuel manage to find their way past the piston rings and into the crankcase, known as the blow-by gases. These gases mix with the hot oil vapor and start to build-up, if left unventilated or untreated these can lead to catastrophic failures. Therefore it is of uttermost importance to ventilate those gases without harming the environment. Back in the old days, all the engines used to ventilate those gases into the atmosphere, polluting the environment many nasty pollutants. But thankfully, the technological advancement has led to the PCV system becoming a standard and mandatory on all engines.

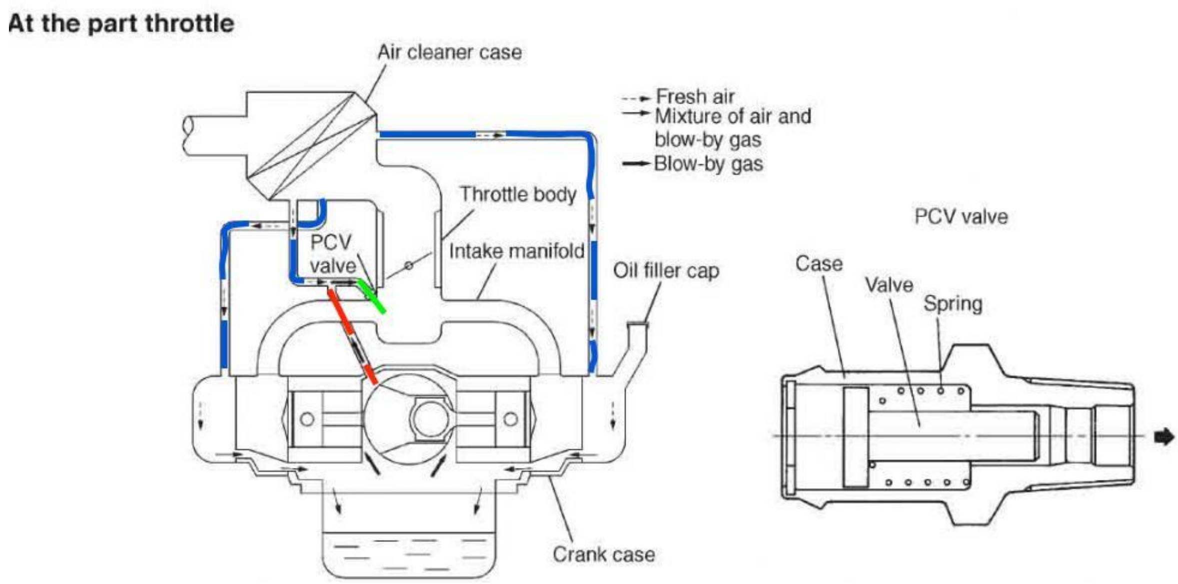

Figure 22: Postive Crankcase Ventilation

As it can be seen in the figure above, the PCV system, compromises of a series of pipes and valves, it is vacuum operated by the engine, which allows it pull all the built-up gases through the crankcase. The system then, reintroduces them into the intake manifold, giving the uncombusted nitrous oxides and the hydrocarbons that blew by the rings another chance for complete combustion, or to be managed by the engine's emission control system. Moreover the system has to regulate the ventilation in such as to not cause too much negative vacuum to the engine which can be bad for sealings, this is done by providing positive ventilation into the crankcase, replacing the built-upgases and maintaining a vacuum balance. Here comes the PCV valve role to regulate the vacuum and the flow of the blow-by gases into the intake manifold while maintaining the balance.

1.8 Engine oil

Oil is the fluid that flows through an engine's lubrication system. It is classified using the "XW-XX" system. The "W" stands for winter and the number that precedes it describes the oil's viscosity at 0 °C. The lower this number becomes, the less viscous it gets at low temperatures. The number noted by "XX" describes the oil's viscosity at 100 °C. The higher this number becomes, the thicker it gets at high temperatures. The oil used for the engine considered in this thesis work is 0w-20 [7].

1.9 General design guidelines

A lubrication system can generally be described as a system of individual flow resistances. The different components in the system represent these resistances. Each of these components consumes a certain rate of flow, with the main consumers usually being the main bearings and the piston cooling jets. The sum of all the consumption should add up to the total flow rate produced by the pump. When designing an oil circuit, the main design guidelines for optimizing an engine lubrication system are:

1. Minimizing the hydraulic resistance of the oil network

2. Improving the oil consumption behavior of the oil consumers

Following these guidelines will result in a more efficient system as the friction will be reduced and the pump won't have to work as hard. Furthermore, the system is usually designed to deliver a minimum absolute pressure of 2.8 bar specified by the manufacturer at hot idle conditions. These conditions prevail when the engine speed is low, around 750 rpms and the oil temperature is around 130 °C. Moreover, the oil should not exceed a maximum temperature set by the manufacturer. The oil cooler is thus sized accordingly without any bypassing[8]. It is worth mentioning that some components are governed by pressure whereas other components are governed by the flow rate. The main gallery is pressure controlled and the aim is to keep a minimum pressure level through different engine speeds and temperatures. If a variable capacity oil pump is being used then the flow rate can be adjusted in such way that a constant pressure level can be obtained at the main gallery.

Components like bearings and piston cooling jets are governed by the volumetric flow rate. Bearings need a minimum oil film thickness to reduce friction and avoid metal-to -metal contact whereas piston cooling jets need enough oil quantity to provide sufficient lubrication and cooling for the pistons and ensure a balanced heat transfer [9].

When it comes to the oil flow velocity at the oil pump inlet, it should not exceed more than 3 m/s, since it results in aeration and to some extent cavitation. Cavitation is a phenomenon that occurs when the velocity of a fluid increases and a pressure drop takes place. If the pressure drops lower than the pressure at which the liquid becomes a gas, vapor bubbles can form. When the liquid's pressure recovers these bubbles can implode and travel at supersonic speed. If they hit the surface of a pipe, they can erode it which can lead to mechanical failure over time [10]. Moreover, to avoid aeration and air bubbles formations in the sump, the velocity of the oil coming back to the sump shouldn't be more than 0.5 m/s as well [9].

1.10 Oil aeration

Aeration is a process that can occur in any combustion engine that has a lubrication system. As the name indicates, aeration is the phenomenon where air is entrapped in the lubricating oil. The agitation of fluid through the channels where oil splashes on different surfaces lead to the formation of air in oil, this phenomena worsens as the oil's velocity increases.

Oil aeration comes in three different forms; Bubbles, dissolved, and foam. Dissolved air cannot be seen and doesn't necessarily affect the oil's lubrication properties, however since air is easier compressed and lighter than oil, it can change the compressibility characteristics of the oil. In other circumstances, it can be released as bubbles or foam. Bubbles are small air pockets entrapped and dispersed throughout the oil while the foam is pockets of air that have travelled to the surface of the oil. When Aeration is in form of bubbles, it can significantly affect the oil's lubrication properties which can lead to a faster rate of wear especially to critical components such as bearings. to a reduction of pressure in the oil system.

Aeration is mostly influenced by engine speed and temperature. According to the aeration measurements taken by Aurobay engine testing and to previously done studies, aeration increases with engine speed and temperature. However, the influence is not as big as the speed. As the oil gets aerated, it remains so until the air can escape when oil is residing in the sump, the more time the oil spends there the more air will escape from it, however when the engine speed increases, oil will circulate faster and therefore will have less time to spend in the sump and hence less air will escape resulting in higher aeration. When temperature increases, dissolved air will turn into air bubbles thus also increasing aeration.

Oil pressure also affects aeration as described by Henry-Dalton law :

$$V_{air} = \frac{B \cdot V_{oil} \cdot (P_a + P_s)}{P_a} \tag{1}$$

$$g_x = \frac{1 + a_1.x + a_2.x^2 + a_3.x^3 + a_4.x^4}{b_0 + b_1.x + b_2.x^2 + b_3.x^3 + b_4.x^4} \tag{2}$$

$$e = sqrt(1/K * \sum_{k=1}^{K}(g(\hat{x_i}) - g(x_i))^2) \tag{3}$$

Where V_{air} is the volume of air at 105 Pa and 273 K, B is the Bunsen coefficient, V_{oil} is the volume of oil, P_a is the ambient pressure and P_s is the relative ambient pressure.
The Bunsen coefficient, B, is a proportionality factor that describes how much gas can be dissolved in a given liquid, in other words, air solubility. What the equation above basically says is that when the oil pressure decreases so does the amount of air it can absorb and vice versa. Moreover, aeration can cause pump cavitation, loss of precision, control, vibration, horsepower loss to a certain degree, and increased engine wear due to lack of lubrication. Since aerated oil, as the name suggests, contains air, and air doesn't have the lubrication nor the cooling properties as the oil has, when in the state of bubbles, it significantly reduces the oil's lubrication capabilities and the cooling abilities of the oil leading to hot spots in the engine. In extreme cases, this can affect the oil's pressure at the main gallery, resulting in lower oil's pressure and insufficient lubrication of the main bearings and the system that lies at the end of system, such as the turbo, VVT mechanism, and cam bearings. The lubrication of hydraulic systems such as lash adjusters, journal bearings, and con-rod bearings are especially affected by aerated oil.[11].
However, according to a study done by S.A.E Japan (Nemoto et al.,1996) it shows that different oil formulations have an effect on oil aeration, and most importantly it states that the entrained air bubbles, the ones that can be bad for the engine, disappear and get dissolved after passing though the pump and getting pressurized and only reappear when the oil's pressure drops to near or below atmospheric pressure.

2 Theory

In this chapter, underlying theories, definitions, scientific relations, and major findings from the literature study will be presented and explained.

2.1 Oil circuit physics

2.1.1 Flow and Pressure

Two important physical quantities that govern the lubrication system are flow and pressure. The oil pump produces a flow rate into the system. Pressure is the force exerted on an area and it is derived with the formula $P = F/A$. It is proportional to force and inversely proportional to area. In fluid mechanics, two types of pressure are used, static pressure and dynamic pressure. Together they make the total or stagnation pressure. Static pressure represents the measured force exerted by fluid, perpendicular to its motion, onto the walls of the object that it is passing. Dynamic pressure is the pressure manifested when the fluid is moving, in other words, when it has a velocity. It is worth noting, however, that the dynamic pressure is not technically a pressure but rather kinetic energy that describes the change in pressure due to velocity. The pressure change between a pipe's inlet and outlet increases with the resistance in the system. The more resistance there is, the higher the pressure, assuming the flow is constant. To maintain the same pressure despite a higher resistance, then the flow will be reduced. Resistance is influenced by the oil's viscosity, the number of components, length, and radius of pipes, etc. To better understand this, a look can be taken at Poiseuille's law. According to this law which applies to incompressible fluids and laminar flow [12]:

$$Q = \frac{\pi \cdot r^4 \cdot \delta P}{8 \cdot \eta \cdot L} \tag{4}$$

Where Q is the fluids's flow, r, the radius of the pipe, δP the pressure change between the pipe's inlet and outlet, η, the fluid's viscosity, and L, the length of the pipe. As it can be seen, in a pipe the flow rate is directly proportional to the fourth power of the radius of a pipe and the pressure change between the pipe's inlet and outlet. It is also inversely proportional to the fluid's viscosity and pipe length. If the flow rate is fixed at a specific level, then it can be observed that the pressure change in a pipe will increase if the viscosity of the fluid and the pipe length increase, and it will decrease if the radius of the pipe increases.

Applying this theory to a closed system like the engine lubrication system, can be considered as the pressure change in the system between the oil pump outlet and the sump. Considering the sump pressure is fixed at the atmospheric pressure level of 1 bar, the total pressure measured at the pump outlet will then increase by increased resistance. Nevertheless, since Poiseuille's law assumes laminar flow it cannot be applied everywhere in a lubrication system as depending on the section of the circuit, the flow can be turbulent as well. It gives however an idea of how the pressure, flow, and other physical quantities relate to each other when the flow is laminar.

To understand what happens locally in the system before and after each component, Bernoulli's equation serves as a good tool. According to Bernoulli

$$P_A + \frac{1}{2} \cdot \rho \cdot v_1^2 + \rho \cdot g \cdot h_1 = P_B + \frac{1}{2} \cdot \rho \cdot v_2^2 + \rho \cdot g \cdot h_2 \tag{5}$$

With P being the static pressure, $\frac{1}{2} \cdot \rho \cdot v^2$ being the dynamic pressure, $\rho \cdot g \cdot h$ being the potential energy expressed as pressure, and assuming that the fluid is incompressible and frictionless. From this equation, it can be seen that for the equality to be valid, if the static pressure decreases, the dynamic pressure increases and thus, velocity goes up and vice versa [12]. Thus, the total pressure remains constant. Looking at figure 23 below, to illustrate Bernoulli's theory with an example, there is a pipe with a wider section at the inlet and a tighter one at the outlet.

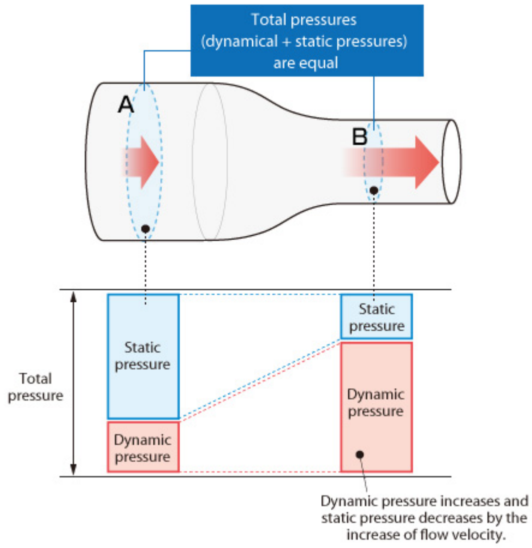

Figure 23: Image illustrating Bernoulli's theorem

According to Bernoulli's theorem, when the area decreases, the velocity will increase and the static pressure will decrease. Thus $P_A > P_B$ and $V_A < V_B$. As a result of higher velocity in the tighter section, the local flux will increase as well but the flow continuity is preserved [13]. Furthermore, if the equation is rearranged to find pressure P_A in the following way:

$$P_A = P_B + \frac{1}{2} \cdot \rho \cdot \delta v^2 + \rho \cdot g \cdot \delta h \tag{6}$$

Since no change in diameter occurs, velocity remains the same, thus dynamic pressure can be deleted and the equation becomes as follows:

$$P_A = P_B + \rho \cdot g \cdot \delta h \tag{7}$$

Rearranging the equations gives:

$$P_B = P_A - \rho \cdot g \cdot \delta h \tag{8}$$

It can now be observed that the pressure in A and B solely depends on the potential energy. To better understand this, a look can be taken at figure 24. It can be seen that when height is included, a pressure difference will occur. Thus, assuming that the flow moves from A to B, the farther up in height one is from the initial point, the lower the static pressure will be as a result of the potential energy loss that results from the height.

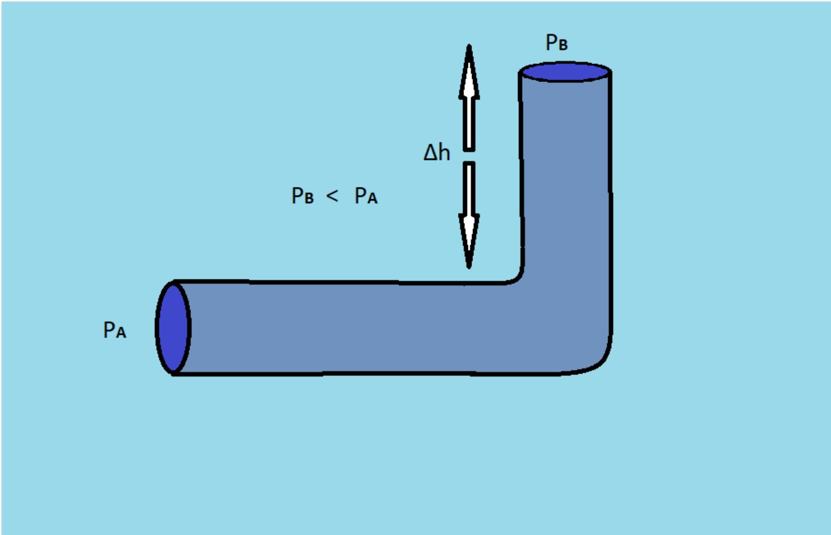

Figure 24: Image illustrating Bernoulli's theorem

If friction is accounted for, then Bernoulli's equation is changed to include one more term that accounts for the pressure losses that occur as following [14]:

$$P_A + \frac{1}{2} \cdot \rho \cdot v_1^2 + \rho \cdot g \cdot h_1 = P_b + \frac{1}{2} \cdot \rho \cdot v_2^2 + \rho \cdot g \cdot h_2 + f_p \tag{9}$$

$$f_p = f_h \cdot \rho \cdot g \tag{10}$$

With f_p being the friction term as pressure and f_h being the friction loss calculated according to the following Darcy-Weisbach equation [**fric**]:

$$f_h = \frac{4 \cdot f_D \cdot L \cdot v^2}{2 \cdot g \cdot d} \tag{11}$$

With f_D being the Darcy friction coefficient that depends on the nature of the flow and the roughness of the pipe's inner surface, L being the length of the pipe, v being the velocity of the fluid, and d being the diameter of the pipe. It can then be observed that the friction loss will increase with the length of the pipe, the velocity of the fluid and the more turbulent the fluid is. It will however decrease with increasing pipe diameter [15]. In this case, recalculating P_B will give:

$$P_B = P_A - \rho \cdot g \cdot \delta h - f_p \tag{12}$$

In other words, P_B will decrease not only due to the decrease in potential energy but also due to the friction loss that occurs along the wall of the pipe.

2.2 Load, speed, and temperature influence on the oil circuit

2.2.1 Engine Load

Regarding engine load, it has an effect on pressure, oil temperature, and the flow rate distribution. Engine load represents the force with which the piston is being pushed. Taking into consideration that the main and big-end bearings sit on the crankshaft, they are directly affected by the load. As the load increases, the max bearing clearances increase, resulting in a higher leakage. Consequently, more flow is directed to the bearings, the flow distribution in the engine changes, and the system experiences less flow restriction. Pressure decreases as a result [16].

2.2.2 Engine Speed

Engine speed has a directly proportional relationship with the system pressure. When a variable capacity oil pump is being used, as in the case of this thesis work, then the pressure will increase with engine speed up until it reaches the target level which is then maintained across the rest of the rpms band. The reason behind the pressure rise that occurs with engine speed is that the bearings' leakage rate increases. More flow is thus demanded and the pump responds by producing more flow which in its turn causes the pressure to rise. Moreover, the pressure drop across components is also affected in such way that it increases with increasing flow rate [17].

2.2.3 Engine Temperature

Engine temperature has an inversely proportional relationship with pressure. Thus, the higher the temperature the lower the pressure and vice versa, assuming that the flow in the system is constant. The reason for that is that the oil's temperature increases with increasing engine temperature, the viscosity decreases as a result and pressure becomes lower as Poiseuille's equation states [17].

2.3 Modelling theory

During the development phase, models are an important tool to gain a deep understanding of a system and to predict and calculate parameters that might be difficult to capture in a test rig. An engine lubrication system is no exception and can be modelled in 3-D to fully capture the flow physics, analyse how the individual components react with their surroundings, and get excellent accuracy. However, such a model would be too complex and require a lot of computational time. 1-D models are a great alternative to 3-D models and enable the possibility to study the system as a whole and optimize it while being much less computationally costly. Nevertheless, that comes at the expense of reduced fidelity. One of the tools commonly used for this purpose is GT-Suite. GT-suite is an industrial simulation tool aimed to aid engineers in designing concepts, running system simulations, and optimizing them. It allows this for mechanical, thermal, and fluid models among others.

2.3.1 1D CFD models

CFD stands for computational fluid dynamics and is a branch of fluid mechanics that utilises computing power to analyse and study flow problems. It makes use of the Navier-Strokes differential equations which describe the motion of viscous fluids. It then turns these equations into a linear system.

These equations are then solved to obtain information about physical quantities such as velocity, pressure, and temperature. Flow through pipes is one of many examples where CFD is used, and since oil systems consist in simple terms of a fluid that runs through a channel of pipes, they also make use of CFD.

A CFD problem is valid over a closed domain that is limited by its boundary. Boundary values, as well as initial values, should be known. These might consist of fluid temperature, pressure, etc. The phenomena should be well defined as well, such as the type of flow, material properties, heat coefficients, the presence of heat transfer, etc. The problem is then solved by dividing the geometry of the domain into shapes known as cells. If fast computing is the priority, the cell size should then be increased, as larger cells take up less memory in the computer. However, if accuracy and precision are favored, then the size should be decreased [18].

2.3.2 Steady-state vs transient simulations

Steady and transient states exist in every system. Steady-state is the state where the properties that affect the system are unchanging over time. In other words, the partial derivative of these properties with respect to time remains equal to zero. This state is usually achieved after a certain amount of time.
The transient state describes how the system changes over time from the beginning of the event and the time it reaches a steady-state. Thus, the partial derivative of the properties that affect the system is not equal to zero [19].

2.4 GT-Suite lubrication system model

The 1D model in this thesis work is created in GT-Suite. It is consisted of a series of pipes, splits, and bends that connect the different lubricated components in the system, as shown in figure 25. The model in this project is run at steady-state. The important behavior that has been important to capture is thus how accurately the system predicts the pressure in different locations at a certain engine speed, oil temperature, and pressure level. One commonly used assumption in steady-state that has been used is that the oil temperature is similar to the engine block temperature. Temperature measurements exist for this study at different locations in the circuit and they show that the oil temperature hovers around the block temperature. Nominal clearances for the bearings are used in this work and a discretization length of 20 mm is used in the software. Discretization length corresponds to the cell size, explained in section 2.3.1.

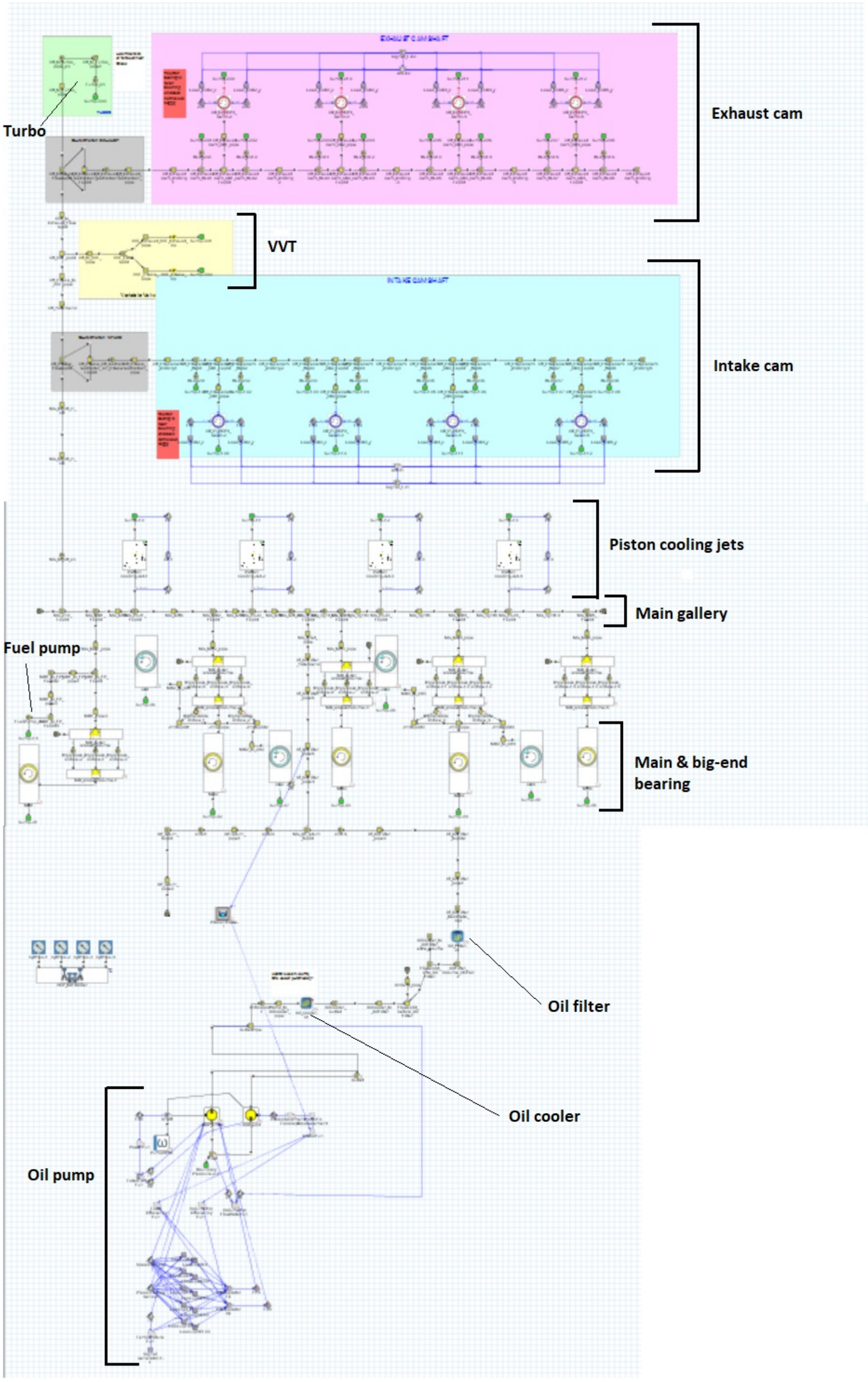

Figure 25: 1D lubrication system model in GT-Suite

As it can be seen in figure 25, the oil system model consists of a pump connected to a sucking pipe in the sump, modelled as a boundary pressure. The pump is connected through a series of pipes to the oil cooler and filter. The pipes connections continue to the main gallery, where a series of oil holes and tunnels emerges to supply the main and con rod bearings with oil. From the main gallery, another tunnel goes up to the fuel pump, intake and exhaust cams, VVT, and turbo. The cam lobes, bearings, and hydraulic lash adjusters get lubricated lastly in the circulation. The oil then drips down to the sump through a series of drainages, to be sucked again by the pump, cooled down, and recirculated.

2.4.1 Components' modelling

The different components in a lubrication system should be modeled in such way that the pressure drop across them is correct. Consequently, there exists a different way to achieve that. They can be modeled in a detailed way in such manner that they physically resemble the real system as much as possible. That is however complex and in many cases renders the model computationally heavy. Thus, engineers look for other ways of modelling that are still representative enough. Another way would be then to model a volume that represents the physical volume that the real component rakes up followed by a pressure-drop map that describes the resistance that it forms to the flow. This is usually done in steady-state simulations, as the behavior of the component itself with time is not relevant as in the case of transient simulations. The map usually contains different pressure drop values as a function of flow rate at different temperatures as shown in the table below. It is either provided by the component manufacturer or by running an individual test. Such a map can be used for the oil cooler, filter, and VVT among other components.

Table 1: Pressure drop map

Temperature (°C)	**Mass flow rate (kg/s)**	**Pressure drop (kPa)**
40	0	0
40	0.5	20
40	1	40
90	0	0
90	0.5	10
90	1	20
120	0	0
120	0.5	7
120	1	15

Components can also be modelled by a simple orifice. This is true for the ones that lie at the ends of the circuit. As they don't lie in the middle of the flow, there is no need to accurately capture the flow passage through these components, thus they don't have to be modelled in a detailed way. The orifice's diameter is then decreased to match the pressure level obtained from measurement data. This is possible because the orifice will be representing a flow restriction to the system which will result in a pressure rise. According to Stephan Weber, a fluids application engineer who works at GAMMA TECHNOLOGIES, the previously explained approach is usually used for components like the fuel pump and turbo. Since the model is run at steady-state conditions, and one of the objectives being to have simple approach to reduce computational time, and since the turbo lies at the end of system and the oil pump has outlet channel that goes back to the sump, meaning that the oil that circulates through them will drop back to the oil sump, the oil temperature at the sump is predefined in the GT-suite model taking into account the temperature increase of the turbo and fuel pump, and assumed constant throughout the system.

2.4.2 Bearing theory

Bearing clearance is the total distance that the bearing shaft can move till it touches one of the bearing shells as figure 26 shows.

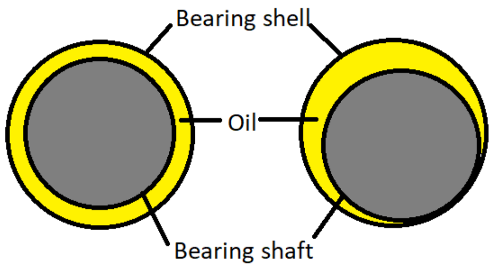

Figure 26: Bearing clearance

A range is usually given by the supplier. This range is bounded by a minimum value and a maximum value. This means that the bearing can have any of the clearance values specified in the range, at room temperature (20 °C). When the engine is operating the value of the clearance does not remain the same but will increase with temperature. An important aspect related to these bearings is the lemon-crush shape that they can get when installed. As seen in figure 27 this shape arises when the housing bolts of the bearing are torqued down; stress will be put on the bearing shell which will change its shape and become oval, resembling a lemon, hence its name. According to Ove Kaldemark however, a design engineer that works at Aurobay with engine bearings, the former explanation applies to main bearings but not to the big-end bearings. Big-end bearings' round form remains even after the bolts are torqued down. A lemon shape is consequently added there on purpose to create more space horizontally. This space will then allow for an increased oil flow in the bearings which decreases the local load at their ends.

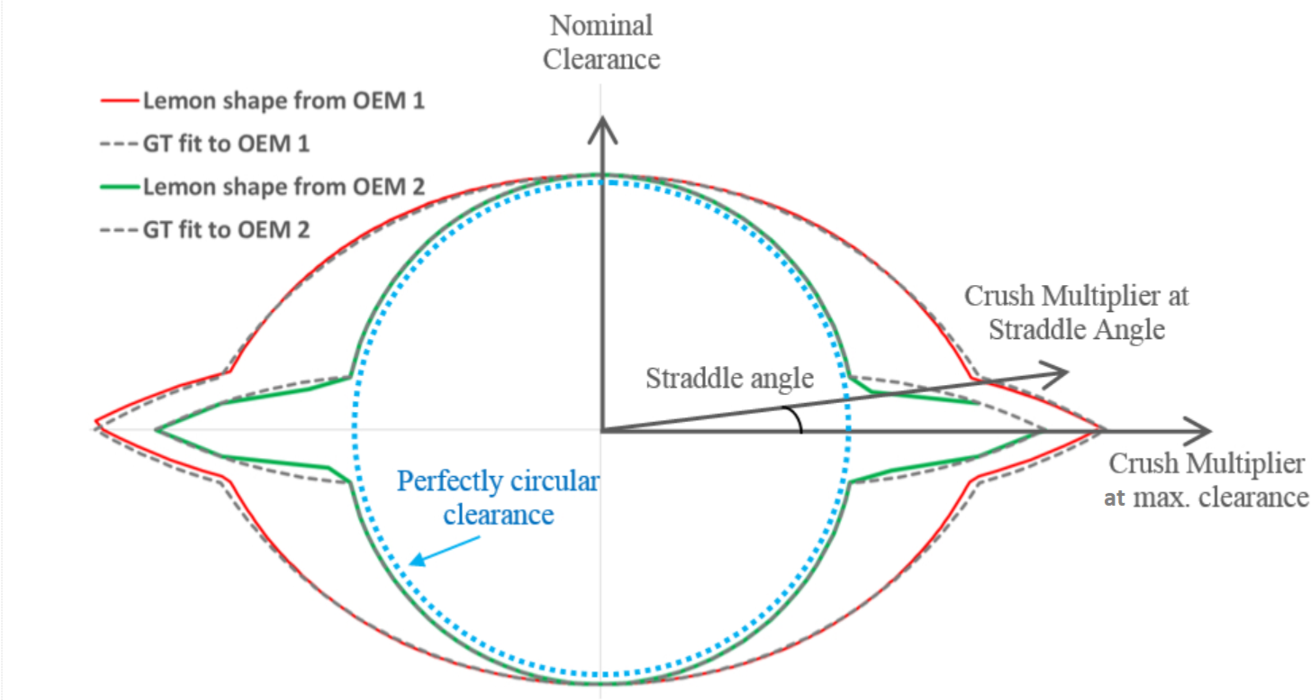

Figure 27: Lemon shape

The shape can resemble the red or the green circle in figure 27 and will usually have some features that describe it. The first one is the straddle angle which is the angle at which the edge of the lemon shape starts. The second one is the crush multiplier at maximum clearance which is a value multiplied by the nominal clearance at the reference temperature, to mimic the clearance at the location of the maximum clearance. This location corresponds to the point of the shape lying on the x-axis and the farthest away from the origin. The third feature is the crush multiplier at straddle angle, and it also represents a value multiplied by the nominal clearance at the reference temperature, but to match the clearance at the start location of the straddle angle. (GT Manual, lubrication manual)
Furthermore, bearings need to be modelled correctly and their temperature and load dependence should be accounted for as well as changes in their clearances entail pressure and flow changes in the system. In GT-Suite, the load is accounted by a table of clearance per load values expressed in micron/kN in X and Y radial directions seen in figure 27. These values adjust the clearance according to the load which translates into a change in the oil's flow-rate. When it comes to temperature dependence, bearings' clearances at engine operating temperatures are significantly larger than at ambient temperature, as mentioned earlier in this section. Templates for different types of bearings are available to use in GT-Suite. The software also uses an equation for the temperature dependent clearance increase as follows:

$$C_r(T) = \frac{D_{outer,ref} \cdot (1 + \alpha_{outer}(T - T_{ref})) - D_{inner,ref} \cdot (1 + \alpha_{inner}(T - T_{ref}))}{2} \tag{13}$$

where
C_r=the new radial clearance at T
α_{outer}=thermal expansion coefficient of outer material
α_{inner}=thermal expansion coefficient of inner material
T= Oil temperature

T_{ref}=reference (typically room) temperature
$D_{outer,ref}$=$D_{inner,ref}$+$2C_{r,ref}$
$D_{inner,ref}$=diameter of the inner surface at reference temperature

2.4.3 Calibration theory

Going from a 3D model to a 1D model means that simplifications will be made. Hence, the physics of the flow and how the flow interacts with the system and its surroundings won't be fully captured. As a result, the model might not be predictive enough and simulations done on such a model, might not give accurate results. That's where calibration comes into place; by tuning specific, carefully chosen system parameters, the model can be adjusted so that its predictability is improved and simulations results lie in good agreement with the measurement data.
For a lubrication system, the choice of calibration parameters can vary. According to Weber, however, the main ones that can be used during the calibration process are:

1. Piston cooling jet inlet orifice diameter
2. Fuel pump hole diameter/thickness
3. Turbo hole diameter/thickness
4. Nominal bearing clearances

Other parameters that can be used as well are:

1. Flow multipliers
2. Friction multipliers
3. Pressure drop multipliers

The reason why the orifice diameter for the piston cooling jet is used as a tuning parameter is that piston cooling jets are often not modelled in a predictive way. The volumetric flow rate and pressure might then not match the measured data and tuning the orifice diameter is thus the solution. For the fuel pump and turbo, having a detailed model of them is usually not essential on the oil system level as a whole, since they are auxiliary components in the system. A specified pressure level still has to be maintained at these locations, however. Computational power can thus be saved by modelling them in a simple way, with an orifice for example. Nevertheless, this means that friction and heat losses that occur in the real system won't be accounted for which affects the results. This is then fixed by tuning the orifices' hole diameters. Hole thickness can be tuned as well for these components; what is meant by hole thickness is the length of the orifice. This parameter is tuned in case a more linear pressure drop is desired as opposed to a more quadratic behavior obtained when no thickness is used.
The calibration process can be performed against measured values at a specific temperature or range of temperatures. The choice of temperature depends on the purpose that the engineers might have for the model. Additionally, a good check when performing the calibration is to make sure that the fluid properties and boundary conditions are correct.
As mentioned in the previous section, bearing suppliers give a range of clearances that a bearing might have. The majority of the clearances will come with the nominal clearance which is the average of the minimal and the maximal one but some will deviate from the nominal value. The nominal clearance can be measured even after the engine is assembled. However, if this is not the case, the clearance can be used as a tuning parameter to obtain correct pressure and flow results.
Flow multipliers directly affect the flow and can be used if the flow rate isn't correct at a certain position. Pressure drop multipliers directly affect the pressure drop and can be used when a component isn't causing an accurate pressure drop. Finally, friction multipliers can be used in

pipe sections and directly affects the friction in a pipe, which in its turn affects the pressure drop.

2.4.4 Test rig measurement

Only having a model with nothing to validate against won't give valuable insights to engineers, if the purpose is for the model to predict the behavior of a real system. In other words, the model cannot be quantified and observed too see how good it is. Measurement data from a test rig are thus needed to compare them to results from model simulations. When it comes to a lubrication system, mainly pressure and flow rate measurements across different locations in the system and at different oil temperatures are of interest. These are measured with the help of pressure sensors, thermocouples, and flow meters. In the case of a flow meter being installed on the engine in the test rig, it most probably will result in an increased pressure drop. If this is the case, this should be accounted for in the model. Thus, the flow meter should be modeled and incorporated into the model.
Typical locations for pressure measurements are as follows:

1. Pump outlet: To ensure correct pump delivery
2. Before and after oil filter: To measure pressure drop across the filter
3. Before and after oil cooler: To measure pressure drop across the cooler
4. Main oil gallery: To ensure sufficient pressure for main and big-end bearings
5. Cylinder head oil gallery: To ensure sufficient pressure at the intake and exhaust cam
6. Before fuel pump: To ensure sufficient pressure at the fuel pump
7. Before turbo: To ensure sufficient pressure at the turbo

Flow measurements can however be difficult as the positions of interest lie at the bearings and piston cooling jets. The difficulty lies in the unavailability of space at these locations. Fitting a measuring device becomes fiddly. Thus, measuring the total flow at the pump outlet is usually opted for instead.

2.4.5 Validation and accuracy estimation

A model is only a representation of the real system. How good the representation is, depends on how accurate the system components capture the behavior of the real system. To validate the model, the first step is to make sure the model is behaving as expected; the system's flow rate should increase with increasing engine speed. Moreover, the total pressure should be decreasing downstream of the flow; static pressure might increase if there is a sudden increase in a pipe area but the sum of the static and dynamic pressure should still be decreasing, moving down through the system. and the sum of the flow rates exiting a node should be equal to the sum of these entering it [20].
The second step is to verify the simulated values against measured values. The comparison should take place at a different set but cover the same parameter space at which the model got calibrated. Accuracy can then be estimated by calculating the error between the two results. Error calculation can be done in different ways: The average relative error and the maximum relative error. Both types of error determine the model's accuracy. Having said that, the average relative error gives a more general insight into the model's fidelity while the maximum relative error serves more as a guide to identifying the model's weakness and where it lies.

$$RE_{average}(\%) = \frac{\sum_{n=1}^{N} \frac{|P_{sim} - P_{meas}|}{|P_{meas}|}}{N} \cdot 100 \tag{14}$$

$$RE_{maximum}(\%) = Max\frac{|P_{sim} - P_{meas}|}{|P_{meas}|} \cdot 100 \tag{15}$$

With P_{sim} being the simulated pressure value, P_{meas} being the measured pressure value, and N being the total number of measured values.
Generally speaking, the level of acceptable accuracy differs from domain to domain. In fact, the purpose of the model and how it is going to be used determines its minimum accuracy requirement. In the studies done by Gamma technologies and after communications with GT-support oil experts, an accuracy of +/-10% is considered acceptable during the validation of a lubrication system model. From the literature research, one report mentions an accuracy within 5.5 % for their model [16]. Another report displays a model accuracy of 7% for the pressure values, 5% for the flow rate values, and 3% for the temperature values when compared to test data on a V8 engine [21].

Moreover, the error percentage itself is not enough to determine how good the results are. How well the simulated data follow the measured data also informs how good the model is. That is measured by the correlation factor which is calculated in the following way for vectors and matrices [22]:

$$Corr = \frac{\sum_m \sum_n (X_{mn} - \overline{X})(Y_{mn} - \overline{Y})}{\sqrt{(\sum_m \sum_n (X_{mn} - \overline{X})^2)(\sum_m \sum_n (Y_{mn} - \overline{Y})^2)}} \tag{16}$$

Where X and Y represent the vectors or matrices used, and m and n, the vectors' length that ranges from the first position in the vectors to its last position. $\overline{X}$ and $\overline{Y}$ represent the mean values of the vectors respectively. In a lubrication system, the vectors used would typically be the vector of the simulated pressure values and the vector of the measured pressure values which have the same length.
To better clarify how both the relative error and the correlation factor should be used to judge the performance of a model and the results, a look can be taken at figures 28 and 29 below.

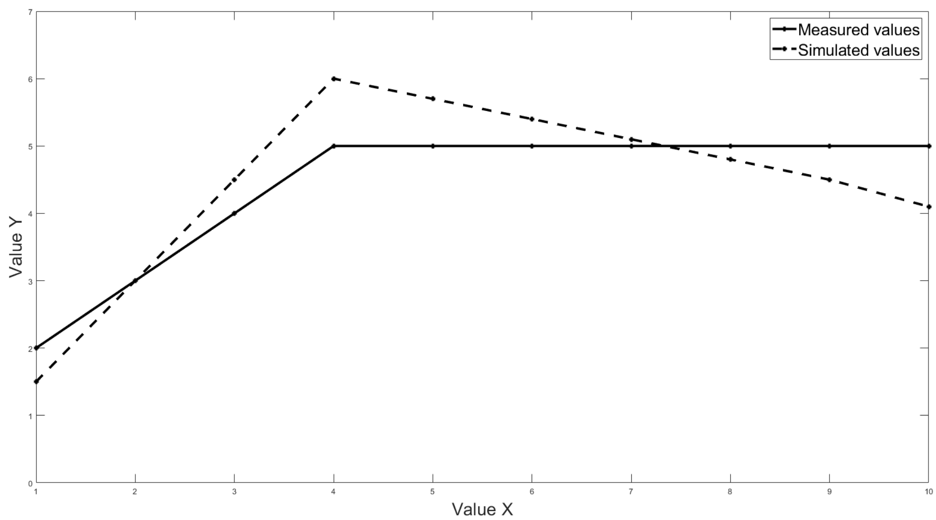

Figure 28: First example plot of measured vs simulated values

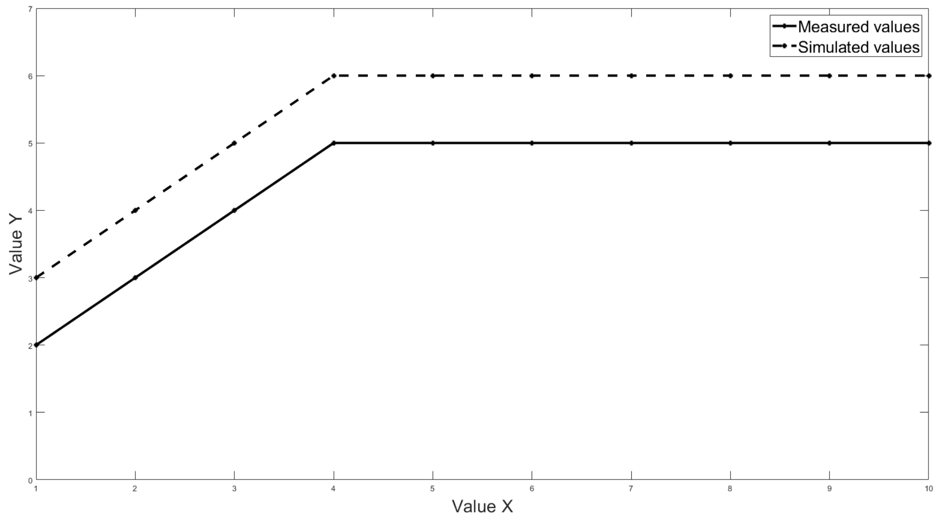

Figure 29: Second example plot of measured vs simulated values

Table 2: Relative error and correlation factor

	First example plot	Second example plot
Relative error (%)	11.35	24.83
Correlation factor (%)	90.2	100

In figure 28, the relative error between the simulated and measured values lies at 11.35 % while in figure 29 it lies at 24.83%. Looking only at these values, the first plot is considerably better in terms of accuracy. However, this neglects the fact that the simulated and measured values don't correlate well in the first plot, unlike the second plot where the correlation is excellent. A high correlation factor serves then as a tool to reliably analyse the system behavior across different engine speeds while a low relative error serves more as a way to analyse the model's predictive capabilities compared to measurement data. All in all, a good balance should be obtained between accuracy and correlation with a general aim to keep the correlation above 95%.

3 Method

In this section, the method implemented to calibrate the model and perform various studies is described.

3.1 Calibration procedure

3.1.1 Previous calibration method

The model as received had 9 tuning parameters consisting of flow multipliers, pressure drop multipliers, friction multipliers, and hole diameters to produce as accurate results as possible. These parameters adjust the flow, friction, and pressure drop in different sections and components in the system to match the pressure values obtained from the measurement data. Furthermore, these parameters had one set of values for each temperature and pressure level. In other words, they were actually variables whose values changed as a function of oil temperature and the system pressure level, as shown in the table below. This approach was adopted because the model is heavily temperature dependent and pressure values change considerably when varying the temperature.

Table 3: Previous calibration parameters

Calibration parameters	60 °C - High pressure	130 °C - High pressure	60 °C - Low pressure	130 °C - Low pressure
Oil cooler pressure drop multiplier	0.8	1.2	0.9	1.1
Oil filter pressure drop multiplier	0.6	0.8	0.5	0.7
Main gallery inlet orifice diameter	10 mm	8 mm	11 mm	9 mm
Main bearing flow multiplier	3	4	1	2
Intake cam flow multiplier	1.3	2	1.1	1.5
Exhaust cam flow multiplier	1.2	1.9	1	1.4
Head gasket friction multiplier	1.5	3	1	2
Fuel pump orifice diameter	1.8 mm	2 mm	2.1 mm	2.3 mm
Turbo orifice diameter	3 mm	2 mm	4 mm	5 mm

As seen in the table, all parameters that adjust the pressure drop, flow, and friction are non-physical quantities apart from orifice diameters. By adjusting them, the quantity which they describe either increases or decreases.

3.2 Updated procedure

3.2.1 Guidelines for parameters' choice

To be able to reduce the model's dependency on the previously discussed parameters, the model should first be thoroughly checked to make sure that the correct input parameters and conditions are being used. Next, all calibration parameters should be reset and/or removed. Then, the calibration process can begin. As few calibration parameters as possible should be chosen in the initial phase. These parameters must be related to the components that haven't been modelled physically, for example, the turbo and the fuel pump. The diameter of these orifices is then decreased to match the target pressure at these locations and the rest of the system. The calibration procedure should cover the temperature range which is used in the measurement test. Including the whole range is optimal but time-consuming, however. Choosing the first and last value of the range, for example, 60 and 130 °C, can thus be representative enough while being time- efficient by ensuring that the pressure requirements in the system are met at the two extremes. The same reasoning applies to the choice of rpms values. The difference is that the rpms range is bigger than the temperature range, thus values between the beginning and the end of the range should be chosen as well to still understand how the system reacts at speeds where the engine operate the most in typical driving conditions. Including several pressure levels in the calibration process can be done as well, following the same logic applied to the temperature range. Having said that, it is not as important to consider, because if the system behavior is correct at a certain pressure level, changing the pressure shouldn't affect it significantly, the results will only be proportional to the pressure level chosen.

3.2.2 Calibration process

The next step should be to disconnect the oil pump from the model, in order to isolate its effect and better understand how the system is behaving. According to Weber, Flow rate values should be imposed at the beginning of the system that corresponds to the ones produced by the pump at different engine speeds, temperatures, and pressure modes. However, if flow rate measurements do not exist, pressure values that correspond to ones measured after the oil pump in the test rig can be imposed as boundary conditions before the oil cooler in the model by creating a sump environment. To proceed with the procedure, GT-Suite's own design optimizer can be used to find optimized values for the chosen calibration parameters in such way that the pressure target is met at different locations in the system. This is done by choosing lower and upper limit values for the calibrating parameters and a search algorithm. Since the nature of this optimization problem is to satisfy multiple pressure requirements, a suitable algorithm is a weighted-sum approach, where weights can be set on the pressure requirements if needed, in such way that the more important the requirement at a certain location is, the higher weight it gets. These weights correspond to values chosen between 0 and 1. The software would then try to find values for the parameters in such way that the pressure levels at different locations, temperature levels, and engine speed values match the target pressure values set by the user. The target pressure values would naturally correspond to the measured ones. A good aim is to strive to get balanced results at the different temperatures chosen, considering that only one set of values for the parameters is being used. To better clarify, a slightly higher than desired pressure can be opted for at 60 °C and a slightly lower than desired pressure at 130 °C.
The different locations usually used for the calibration are:
1. Pressure after oil cooler

2. Pressure after oil filter
3. Pressure at the main gallery
4. Pressure before fuel pump
5. Pressure at the intake cam end
6. Pressure at the exhaust cam end
7. Pressure before turbo

It can thus be observed that not all locations that are used in the test rig have to be chosen for the calibration. The reasoning behind that is mainly to reduce the optimization time and at the same time target the most valuable and important locations that would provide information about how different components are behaving and if enough pressure is being received to the system's most remote locations.

When the pressure results are satisfactory, the next step is to reconnect the oil pump to the system. The same control strategy used in the real system should be used in the model as well. If the outlet flow of the pump in the real engine is mainly controlled by a pressure target that needs to be maintained at the main gallery, the same strategy can be opted for by incorporating a PID controller that connects the pump to the main gallery with the correct target pressure profile. Simulations can then be run to check if the results are similar to those in the optimization process. If they don't correlate, it is most likely that the pump isn't producing the correct flow rate and pressure assuming the model is correct. Ensuring that the input of the characteristics of the oil pump is thus crucial to get the correct flow rate as this will result in correct pressure drops in the system.

3.3 Calibration guidelines

3.3.1 Pressure drop

After the optimization is finished, the pressure targets directly related to the calibration parameters should be checked to see if they are met. Other locations should then be checked to see how the system is behaving and if it is predicting the correct pressure losses along the circuit. If the pressure at a certain location is not met, for example at the main gallery, this can imply that the pressure losses before the gallery, that occur around the oil cooler and filter and the pipes that connect them are not being correctly predicted in the model, which leads to either higher or lower than desired pressure at the main gallery. That, in its turn, can be due to wrong input data in the oil cooler and filter's pressure drop maps. Alternatively, it can also be due to an inaccurate depiction of a certain section of the circuit. This section can have a complex shape in the real system with turbulent flow, and it might be difficult to model it. Oil channels are usually mainly described by round or square pipes and flow splits with dimensions that correspond to the actual ones. Thus, friction losses and flow physics might not be fully captured. A calibration parameter, like an orifice, can then be added in the relevant section and tuned to increase the pressure losses.

It should be noted however, that the pressure map used for the oil filter, corresponds to a new and clean filter since the rig-test measurements were carried through using a new filter to obtain a fair and accurate validation later-on, so no consideration were taken to clogged oil filters.

3.3.2 Bearing clearance

Even if the oil pressure values before the main gallery are correct, locations that lie after the main gallery can be affecting the results as well. If there is too little flow going to the main and big-end bearings or too much restriction in a remote location like the exhaust cam end, this leads to incorrect leakage to the sump and thus higher than desired pressure at the main gallery. As mentioned before, the whole lubrication system is highly sensitive to which clearances are being used in the system. Clearances that are too tight can thus restrict the leakage rate from the bearings which cause the pressure to increase. Hence, ensuring that the right clearances are being used is a high priority. The lemon crush option should be considered as well where needed, though it is usually mainly opted for the main and big-end bearings. It is however crucial to choose the correct values for the parameters related to this lemon shape. Specifically, the crush multiplier and crush multiplier at straddle angle have a big influence on the flow and pressure in the system; the values chosen will directly influence the bearings' leakage and the flow distribution between the main and big-end bearings.

3.3.3 Bearing modelling and feed hole geometry

It is important to note that in case inaccurate results are received, one factor that can have a strong influence is the modelling method of the bearings as different methods can affect the flow rate to the bearings and even change the flow distribution. Moreover, another factor that the model is highly sensitive to is the constants of the bearings' feed hole geometry, as seen in figures 30 and 31.

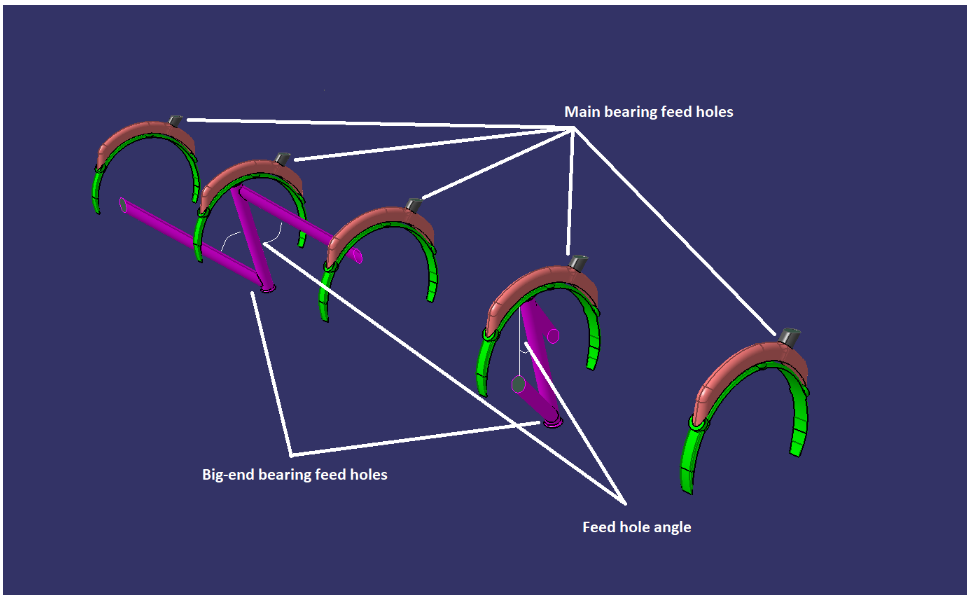

Figure 30: Main and big-end bearings feed holes

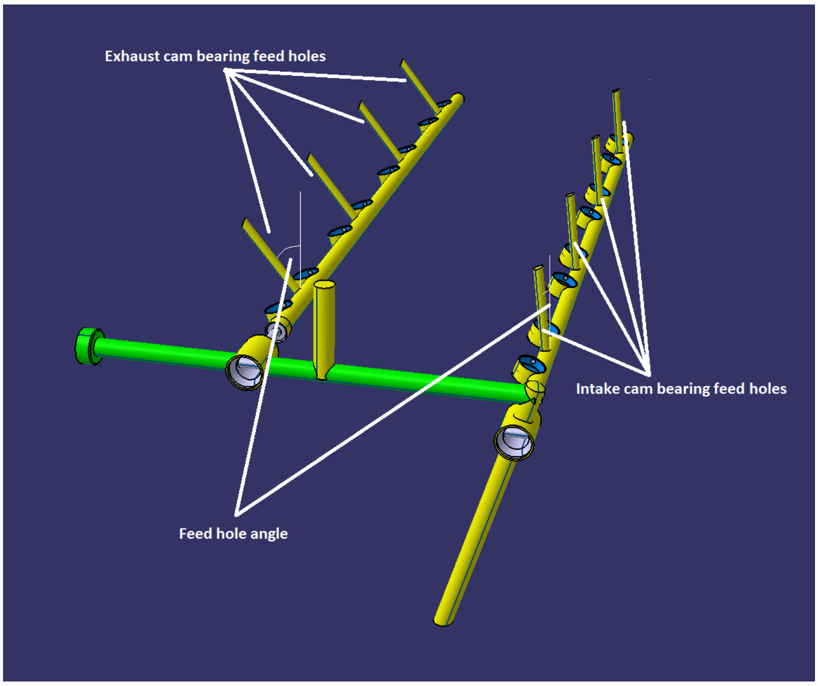

Figure 31: Cam bearings feed holes

The angle of the feed holes that describe the holes' location relative to the coordinate system used in GT-Suite and the rotational direction of the crank- and camshafts have a strong influence on the results. Using a different orientation for the coordinate system than the one implemented in GT-Suite leads to erroneous results. Also, inconsistencies in terms of using one orientation of axes for the feed hole geometry of the cam bearings and another orientation of axes for the feed hole geometry of the main and big-end bearings lead to illogical results as well. That is because the bearing leakage is highly dependent on the feed hole geometry which in its turn affects the direction of the bearing loads. If the load direction is directly against the feed hole, there will be a high pressure on the hole forcing the oil in but the oil won't be able to flow out as easily, the leakage rate will thus decrease and pressure will increase. Moreover, it is worth mentioning that the cam bearings' loads themselves also have a strong effect on the cam bearings leakage which in its turn directly affects the pressure measured there.

3.3.4 Pressure locations

What locations are being chosen to measure the pressure in the model is also of big importance and should correspond to the exact physical locations of the pressure measurements in the test rig. That way, pressure drops over the oil cooler or filter for example can be correctly compared to test data to check if they are representative enough. If the pressure is being measured in a pipe, then it is also important to know if the measuring place should be at the pipe inlet or outlet, specially if there is a change in the geometry after the pipe. If there is an orifice with a reduced diameter after the pipe, for example, the pressure at the pipe outlet will be lower than at the pipe inlet.

3.4 Validation

After the calibration process is completed and reasonable results are obtained, that more or less match the measurement data, the next step is to validate the model. The parameter space considered in this work is a 3D space defined by the temperature, pressure level, and rpms ranges. The validation should then take place in the same parameter space but using different data sets than the ones used in the calibration. To better clarify, at least one dimension of the parameter space should have different values then the ones used in the calibration. Table 3 below shows an example of how the difference between a validation and a calibration process can be. The same pressure level and temperature are being used but the difference is in the rpms range used.

Table 4: Calibration vs Validation

Process	Engine speed (rpms)	Temperature (°C)	Pressure level (bar)
Calibration	750, 2000, 4000, 6000	60	High pressure level
Validation	1000, 3000, 5000	60	High pressure level

Furthermore, the rest of the pressure measuring position is added to the validation process as well. These positions include:

1. Pressure at the oil return to the pump
2. Pressure at the end of the main gallery
3. Pressure before the head gasket
4. Pressure before the VVT

3.5 System sensitivity study

After the calibration and validation process, it is valuable to analyse how sensitive the system is when different parameters are changed. As the bearing clearances are given as a range from the manufacturer, there is both minimal and a maximal value. It is thus important to investigate how the results are changed and what level of accuracy is expected when either minimal or maximal clearances are opted for. Lemon crush parameters are also varied to their minimal and maximal values to see what effect they have on the system.

Engine load is varied as well from part to full load to study its effect on the system, which is later shown to be insignificant compared to temperature effect, therefore it is not shown but rather explained later in the results section.

The discretization length is also important to consider in this study as it can inform about what compromise between the computational time and results accuracy should be expected when the length is varied.

The study takes place by running simulations at an oil temperature of 90 °C, high pressure level, and the engine speeds of 750, 2000, 4000, and 6000 rpms. Minimal and maximal clearances are then chosen for one type of bearing, for example, the main bearings, keeping all other bearings at their nominal clearances. This is then done for each type of bearing. The purpose is to understand what type of bearings is most influential when its clearance is being changed. The next step is to choose maximal and minimal values for the lemon crush parameters and keep all bearings at their nominal clearances to isolate the effect of lemon crush on the system. The next part of the study tackles the discretization length. It is varied to see at which length, the best possible accuracy can be obtained and which length that gives the fastest computational without much compromise in accuracy. At last, the engine load is set to full load and the simulation is done in the same parameter space used for this study with the exception of including the oil temperature of 130 °C apart from 90 °C.

4 Results and Discussion

In this section results from this thesis work are presented and discussed. It is worth noting that the pressure plots presented below show the absolute pressure values of the various locations meaning that it includes atmospheric pressure of 1 bar. Moreover, the y-axis's starting point isn't from 0 but the lowest obtained pressure for the specific case. This has been done to clearly show how the results differ between measurement data and simulations. Furthermore, the plots presented below only show the static pressure since this is the type of pressure measured in the test rig.

4.1 Calibration results

The final tuning parameters used for the calibration are the following:

1. Main gallery inlet orifice diameter
2. Turbo orifice diameter

Table 5 below shows the values chosen for the calibration parameters compared to their original values, and the amount that they were decreased with.

Table 5: Calibration parameters' values

Parameters	**Original values (mm)**	**New values (mm)**	**Decrease percentage (%)**
Main gallery inlet orifice diameter	12.553	9.85	18
Turbo orifice diameter	8	2.2	72.5

Table 6 below shows the engine speed, temperature, and pressure level choices for the calibration.

Table 6: Final Calibration

Final calibration	**Engine speed (rpms)**	**Temperature (°C)**	**Pressure level (bar)**
Calibration	750, 2000, 4000, 6000	60,130	High pressure level

Figures 32 and 33 show the temperature measurements from the test rig taken at different locations in the circuit and different engine speeds when the engine is run at 60 °C and 130 °C and the oil pump is run on high pressure mode.

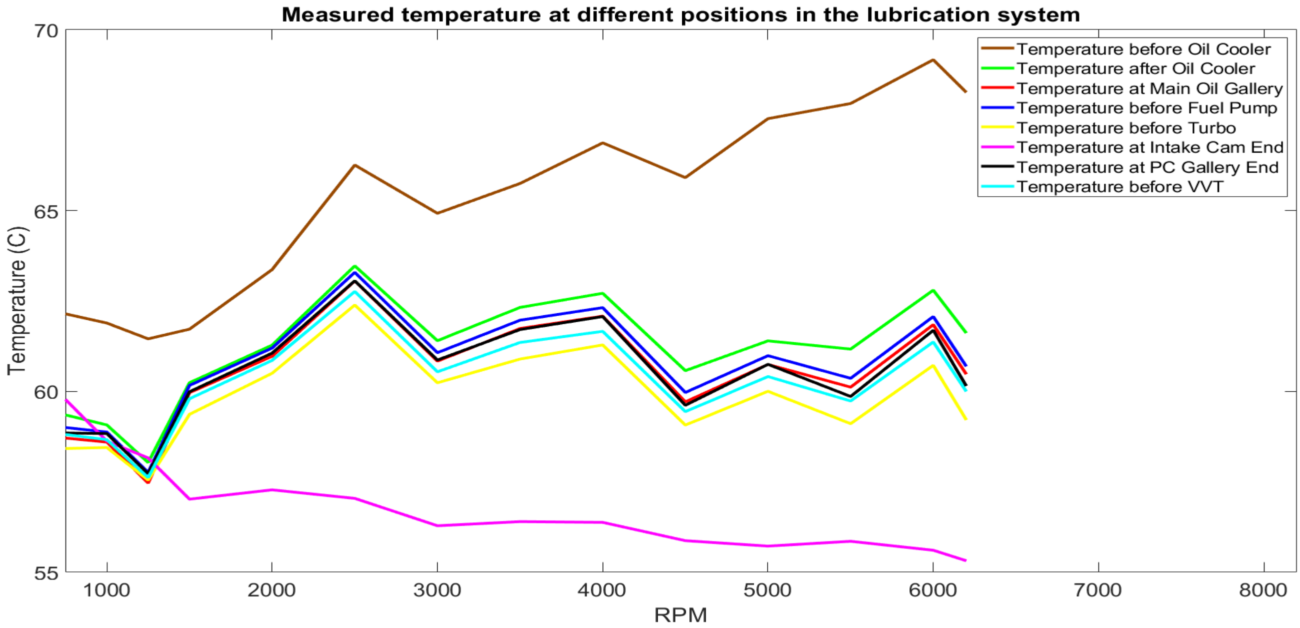

Figure 32: Temperature measurements at 60 °C and high pressure mode

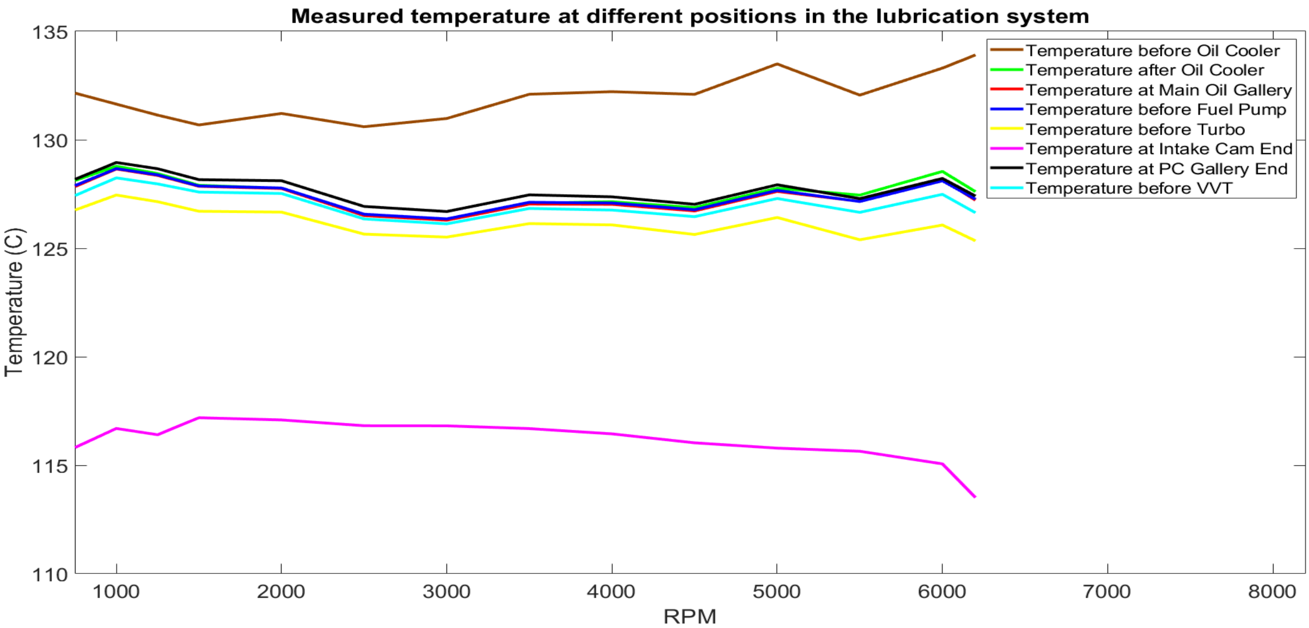

Figure 33: Temperature measurements at 130 °C and high pressure mode

It can be seen, that in both cases, the oil's temperatures at most locations don't fluctuate much and lie in good agreement with the set temperature. The two exceptions are the position before the oil cooler and the intake cam end. When the engine is run at 60 °C, the temperature before the oil cooler hovers steadily increases with engine speed till it reaches around 70 °C and the temperature at the end of the intake cam steadily decreases till it reaches 55 °C. At 130 °C, the temperature before the oil cooler is slightly higher than the set temperature and fluctuates around 132 °C whereas the intake cam end temperature fluctuates around 116 °C.

Table 7: Model's accuracy and correlation in calibration

Model's total average relative error in calibration (%)	Model's total maximum relative error in calibration (%)	Correlation factor in calibration(%)
2.4	9.37	99.03

Table 6 above shows the model's total accuracy obtained during the calibration process. The average relative error obtained lies at 2.4% while the maximum relative error lies at 9.37%. The correlation factor is 99.03%.
Figure 34 below shows the simulation results from the calibration compared to measured data.

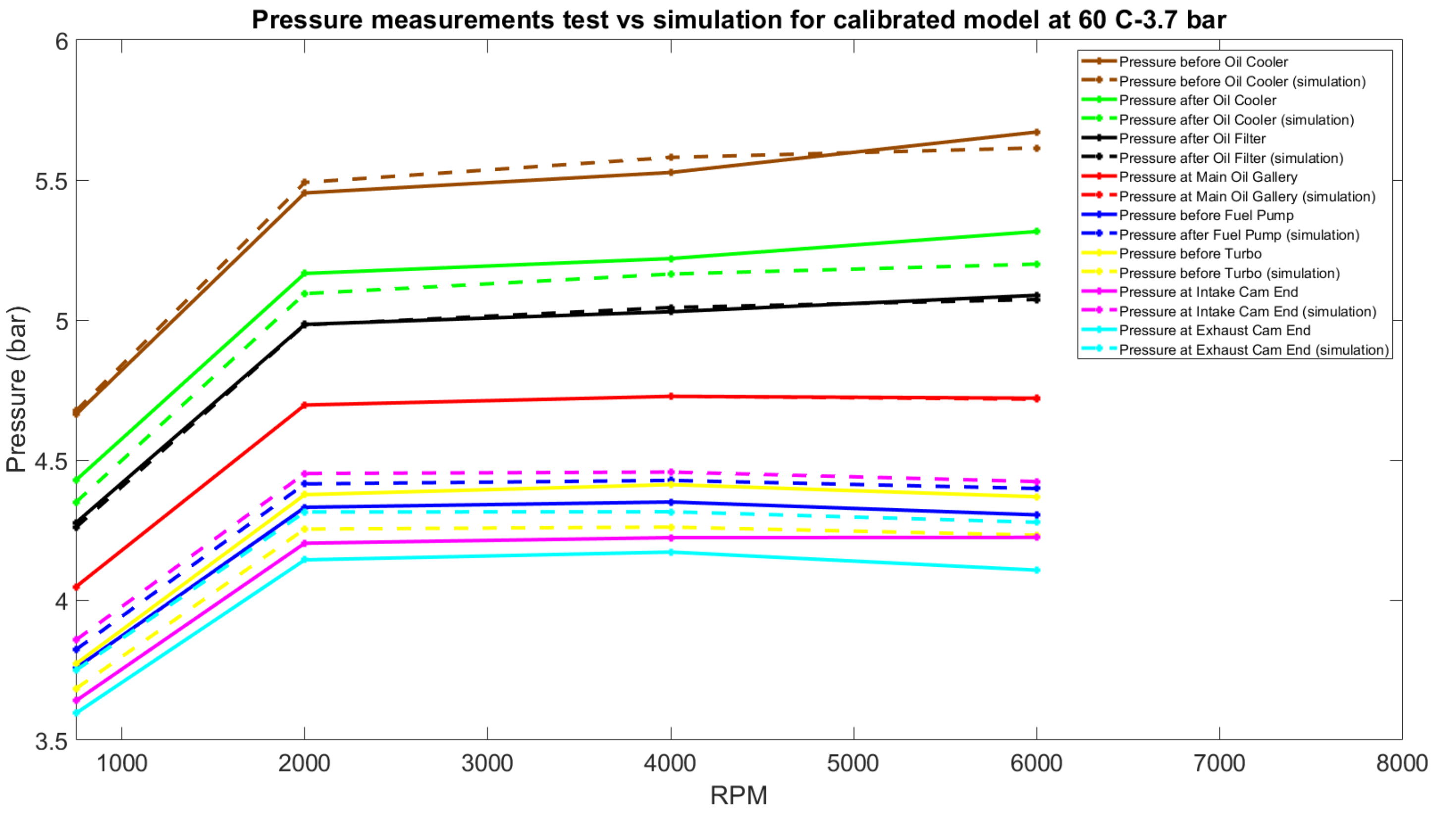

Figure 34: Calibration results at 60 °C and high pressure mode

Table 8: Accuracy and Correlation at 60 °C-High pressure

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 60 °C and High pressure	2.15	5.97	98.04

As it can be seen in figure 34 and table 7, a good agreement is obtained between the measurement data and simulated data. The accuracy lies within an average of 2.15% with a maximum relative error of 5.97% obtained at the position at the intake cam end. Excellent correlation is obtained as well. It can be observed that the pressure before the oil cooler which represents the pressure out of the pump is correct, but the pressure drop over the oil cooler is larger than the measurement data while the pressure drop over the oil filter is smaller. Moreover, the pressure values at the intake cam end, exhaust cam end, and fuel pump are higher in the simulation while the pressure before the turbo is lower.

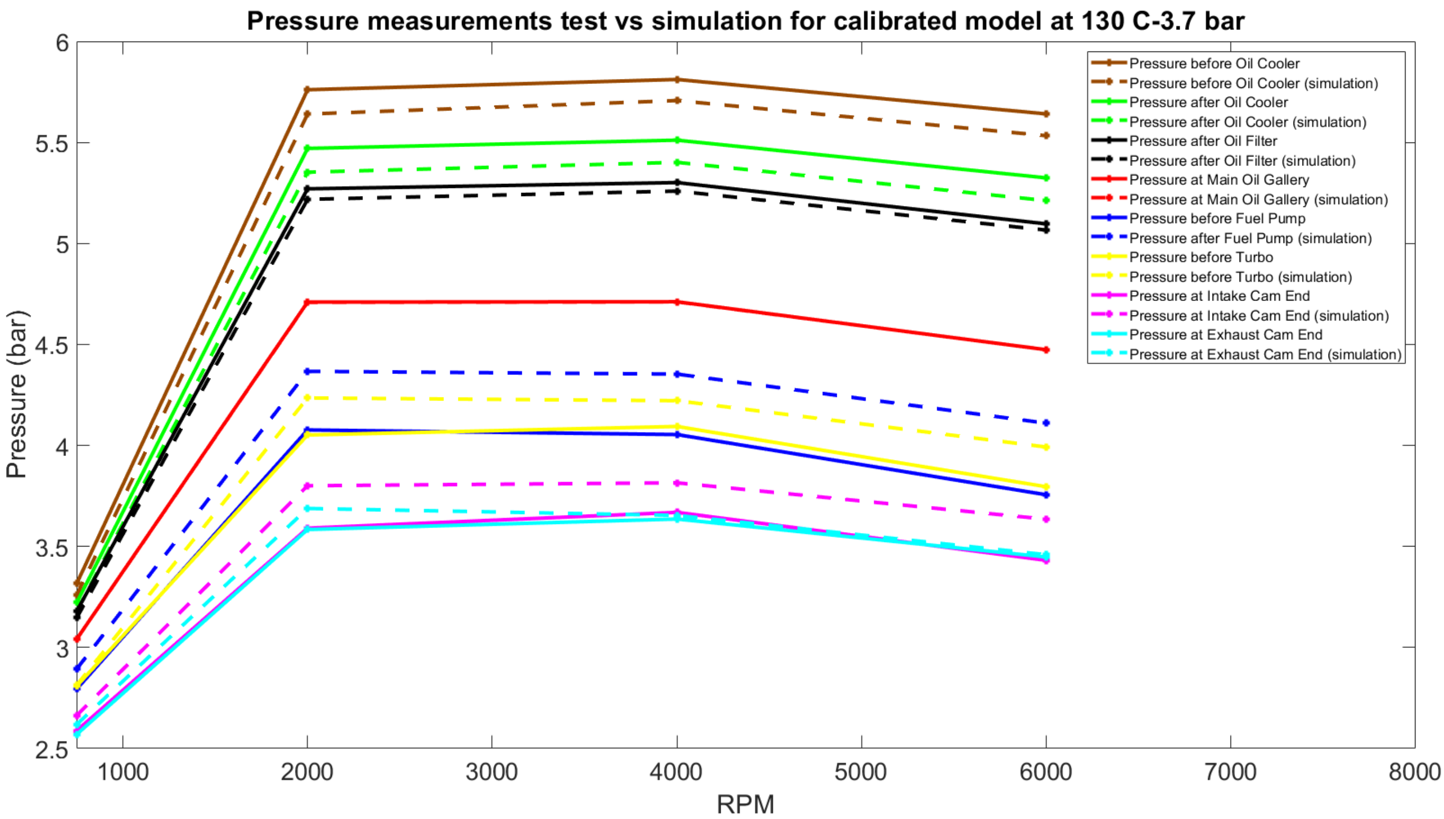

Figure 35: Calibration results at 130 °C and high pressure mode

Table 9: Accuracy and Correlation at 130 °C-High pressure

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 130 °C and High pressure	2.71	9.37	99.24

At an oil temperature of 130 °C, the system behavior is slightly different as seen in figure 35. Table 8 shows that good agreement is obtained here as well, with an average relative error of 2.65% and a maximum relative error of 9.37%, again at the position before the fuel pump. Pressure out of the pump is however now slightly lower than measurement data. The pressure drop over the oil cooler is correct, however. Similar to what is seen at 60 °C, the pressure drop over the oil filter is smaller. The pressure values at the intake cam end, fuel pump, and before turbo are higher as obtained in the simulation but the exhaust cam end pressure is now equal to the one from measurement data.
To get an idea of how the flow distribution looks like at the two temperature extremes, a look can be taken at figures 36 and 37.

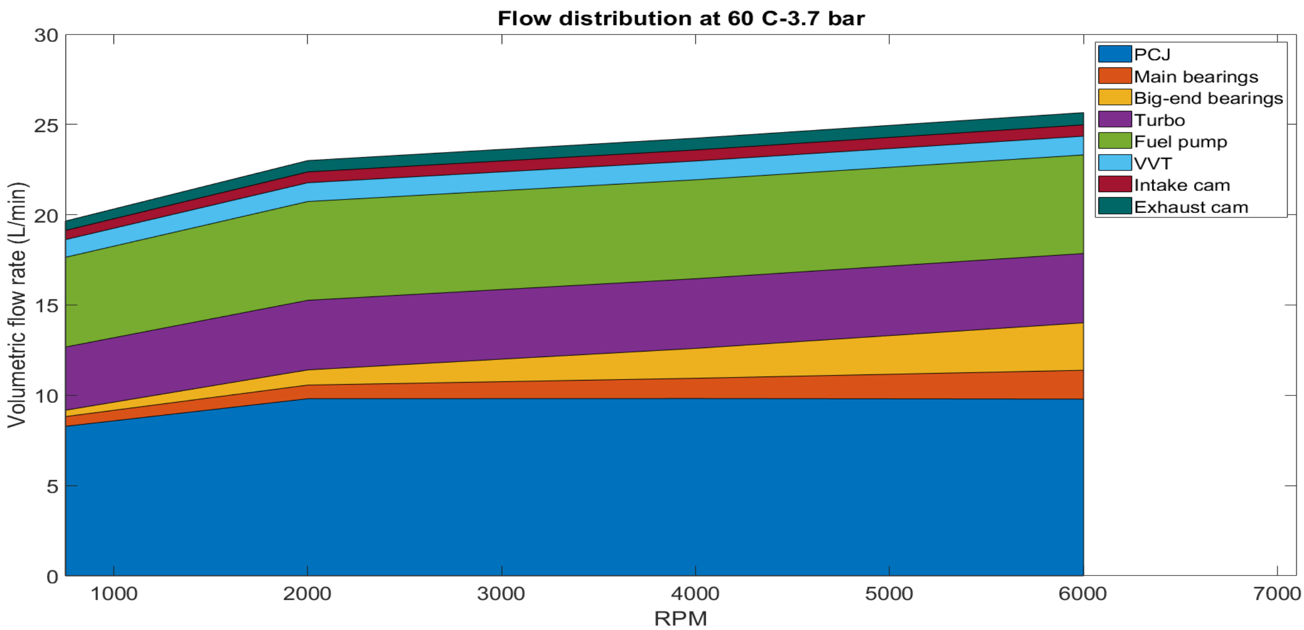

Figure 36: System flow distribution at 60 °C and high pressure mode

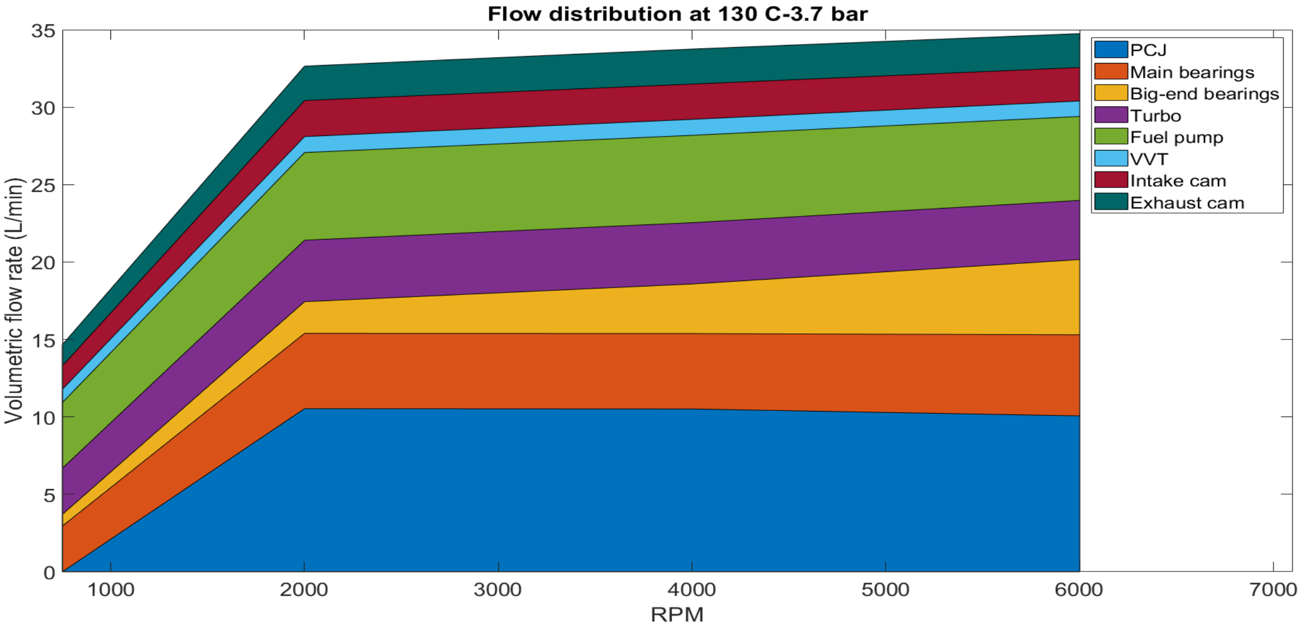

Figure 37: System flow distribution at 130 °C and high pressure mode

It can be observed that temperature affects the system flow distribution. In both cases, piston cooling jets are the biggest oil consumers followed by the fuel pump. At 60 °C however, big-end bearings consume more oil than main bearings whereas at 130 °C the opposite happens. The turbo goes from being the third biggest oil consumer in the system at 60 °C to the fourth biggest at 130 °C after the main bearings. What is seen in the flow graphs is logical and predictable, at 60 °C, the oil is viscous and relatively thick, hence the low oil consumption by main bearings. On the other hand the big-end bearings' oil consumption is highly determined by the rpms, so even at 60 °C the higher the rpms is, the more oil they will consume due to the rotational movement and the inertia caused by it. Which explains the seen trend in the oil flow graph regarding the big-end bearings, where the oil consumption increases proportionally with rpms. At 130 °C, it is observed that main bearing's oil consumption is considerably higher than the big-end bearings' consumption, it also seen that the main bearings' oil flow doesn't seem to change much at high rpms and remain more or less constant across a wide range of the rpms. The reason for this can be traced back to the different materials' expansions and reaction to high temperature. The engine block is made of aluminum where the steel-made bearings shell, sits, and on them sits the crankshaft which is made of steel. However when it comes to big-end bearing, both big-end and crank pin, in other words, the journal and bearing are of the same material, which is steel. Aluminum is more sensitive to heat than steel and expand significantly more than steel, which means that as temperature increases, reaching 130 °C, the gap between the big-end bearings and journals will remain constant as they will both expand at the same rate. While the main bearings will react differently, since the shells are made of steel, they will expand less and at a much slower rate than the aluminium block on which they sit on, resulting in a higher oil consumption. Furthermore, both the big-end and main bearings' oil consumption increase at 130 °C compared to 60 °C, due to the oil's low viscosity at higher temperatures.

4.2 Discussion

4.2.1 Calibration parameters discussion

As explained earlier, the orifice diameter of the turbo is being used as a calibration parameter to account for friction and heat losses that occur as the turbo is modelled in a simple way. The diameter has thus been reduced to match the target pressure at this position. Moreover, there hasn't been a need to tune the fuel pump diameter and it has been kept at its original diameter. This is because the pressure before the fuel pump has been found to reach and even exceed the target value. The inlet orifice diameter of the main gallery has been decreased as a way to increase the pressure losses between the oil filter and the main gallery. Leaving the diameter at its original diameter caused the model to under-predict these pressure losses. This meant that during the calibration process, when the pump was disconnected, the pressure at the main gallery was higher than the target value. When reconnecting the pump to the system, it would regulate the flow rate so that the target value at the main gallery was reached. However, this happened at the expense of obtaining a lower than desired at the pump outlet, before the oil cooler. Hence, for the pump to achieve the correct outlet and main gallery pressure at the same time, the pressure drop after the oil filter had to be correct.

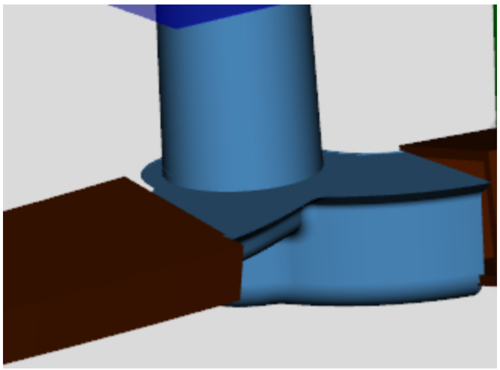

Figure 38: Flow split at the main gallery inlet

Figure 38 above, shows a flow split right before the main gallery where the pressure is measured. It can be noticed that the section is geometrically complex. The friction losses that occur here are most likely not accurately depicted in GT-Suite. To correct this and reproduce correct pressure losses, the inlet orifice, which is located right before the flow split has been calibrated.

4.2.2 Calibration results discussion

Looking at the calibration results at 60 °C, the large pressure drop observed over the oil cooler and small pressure over the oil filter can naturally be first attributed to inaccurate data in the pressure drop map used for the oil cooler and filter. These maps have been obtained from tests where these components are isolated and the flow rate and oil temperature are varied to obtain pressure drop data. The map for the oil cooler has been obtained using a different oil than that used in the test rig. The oil used in the component is a 5W-20 which has a higher viscosity at low temperatures whereas the oil used in the test rig is 0w-20. This higher viscosity explains the large pressure drop at low temperatures. At high temperatures both oils have similar viscosity, hence the correct pressure drop observed at 130 °C. With that said, the difference in the viscosity between the oils used, shouldn't necessarily result in the larger pressure drop observed at 60 °C. Taking into consideration that the pressure at the pump outlet at 130 °C is lower than the

target level, this can mean that the system flow rate is being over-predicted at low temperatures and under-predicted at high temperatures. This can explain the phenomena discussed above.
The map for the oil filter has been obtained using the same oil like the one used in the test rig but the pressure drops observed in the simulations are still lower than in the test rig. The explanation for this can lie at the position where the pressure is measured in the model. In the test rig, the pressure is measured at the oil filter bedplate. Naturally, the same position has been chosen in the model as well, as seen in figure 39.

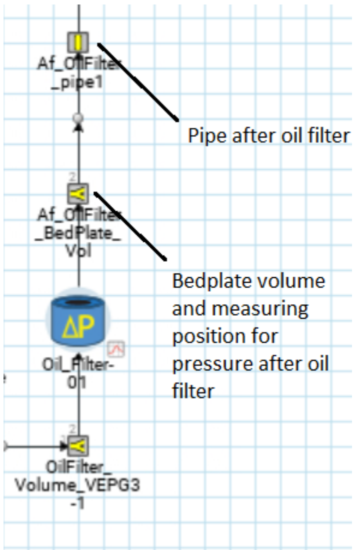

Figure 39: Measuring position for the pressure before the oil filter

The bedplate is modelled as a flow volume; so it might not be fully capturing friction losses that occur there. Add to that, the pressure sensor used in the test rig might eventually lie at the connection between the bedplate and the pipe after the oil filter, a position where the pressure decreases slightly. In other words, it is possible that the two measuring positions don't lie at the same exact location. All of the above can then explain the lower pressure drop observed in the simulation results.
Regarding the pressure before the fuel pump, it is noticed that it is higher than the target at both 60 °C and 130 °C, but specially at 130 °C where the highest relative error exists. To present a possible explanation, a look should be taken at figure 40 which shows the measuring position of the pressure before the fuel pump.

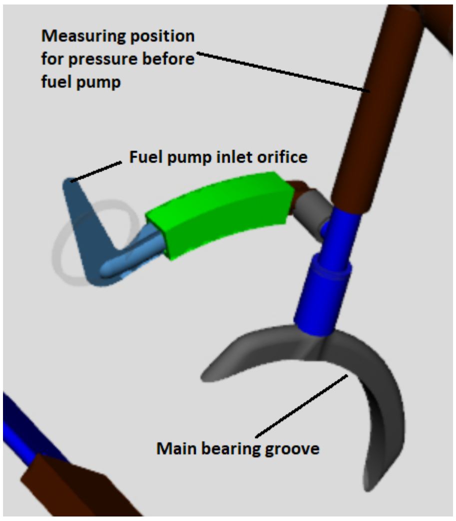

Figure 40: Measuring position for the pressure before the fuel pump

It can be observed that the oil channel that is used for measuring the pressure before the fuel pump is the same one that leads the oil to the main bearing. That suggests that if the flow going to the bearing is not correct, it might affect the pressure at the fuel pump as well. Looking at figure 37 it can be seen that the main bearings aren't the biggest oil consumers, as they should be along with the piston cooling jets according to Rodrigo Aihara, an applications engineer that works with lubrication at GAMMA TECHNOLOGIES. Moreover, since no flow measurements exist for this thesis work, it is difficult to estimate how the bearing flow predicted correlates with the flow in the test rig. However, a reasonable guess that can be drawn is that they should consume more flow than what is seen, and since this isn't the case there is less leakage from them which causes the pressure at the channel leading to the main bearing in figure 40 to increase. As it happens to be the same channel used for measuring the pressure before the fuel pump, this can be the reason why a relatively high error is obtained at this position.
Further investigation into this matter provides further proof that the flow seen in the main bearings is probably considerably lower than in reality. Looking at figure 41, it can be seen that at the point where the two bearing shells meet, there is a chamfer that spreads across the width of the bearing which allows for more leakage out of the bearing.

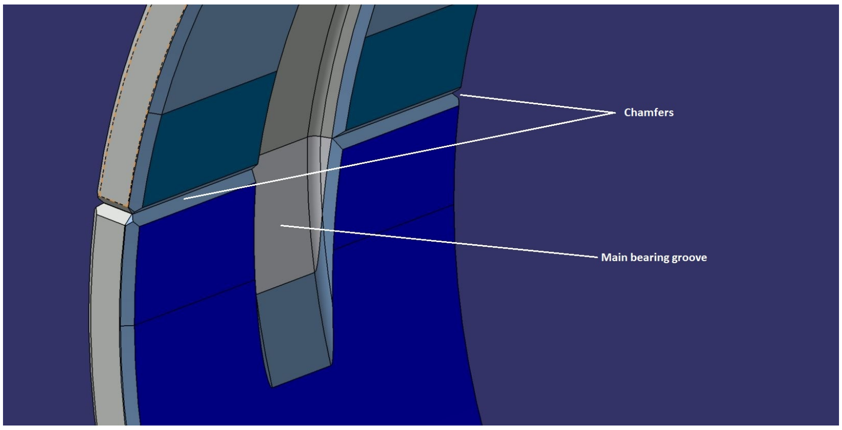

Figure 41: Main bearing chamfer

However, this is unaccounted for when calculating the lemon crush parameters in GT-Suite, specifically with regards to the crush multiplier and the crush multiplier at straddle angle. The reason behind this is that the template provided by the software doesn't take into account bearings with special construction. Therefore, this provides further insight that the leakage is being under-predicted.
Regarding the pressure before the turbo, different reasons can lie behind the fact that the pressure is lower at 60 °C and higher at 130 °C. The decreased diameter results in a flow restriction that raises the pressure at this position. At 60 °C however, the viscosity of the oil is high which causes high pressure losses. At 130 °C, the oil's viscosity decreases considerably which reduces the pressure losses that can be caused by the oil's internal friction. Nevertheless, this doesn't hide the fact that an orifice may just not be the optimal way to describe the turbo. As described earlier, the oil passage inside the turbo is complex, thus a detailed modelling approach might be needed or a pressure drop map, like the one used for the oil cooler and filter. The pressure at the intake cam end and exhaust cam end is higher than the target values at 60 °C. At 130 °C, only the intake cam end pressure is higher. From a model's point of view, the exhaust cam is situated above the intake cam, as shown in figure 42 below.

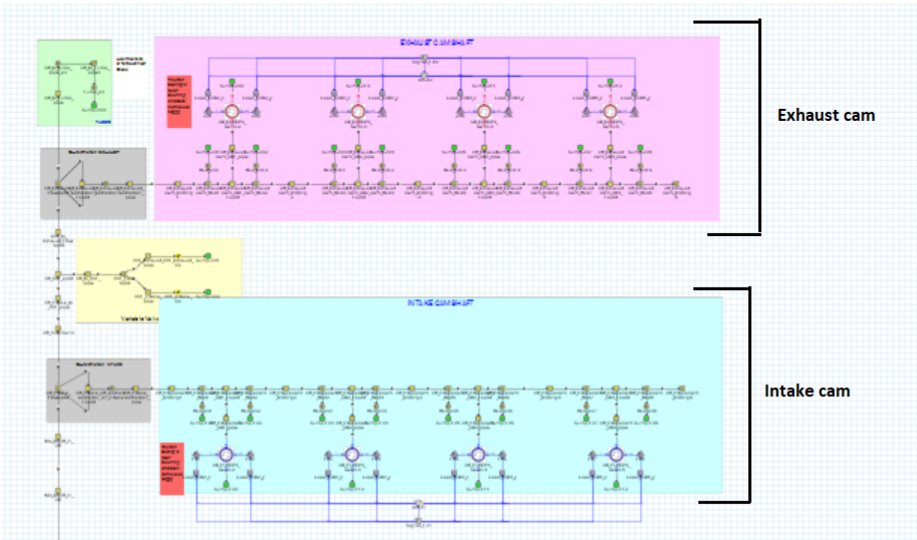

Figure 42: Intake and exhaust cams in GT-Suite

That means that additional pressure losses occur when the flow reaches the exhaust cam, because of the additional channels that the oil has to go through. It is thus reasonable to expect that the pressure at the intake cam end should be higher, assuming that the exhaust and intake cam bearings leakages are similar, which is the case when looking at the system's flow distribution shown in figures 36 and 37. This brings the question of whether the leakages predicted are accurate or not. The camshaft loads and speed are the parameters that directly affect how much flow goes into the cam bearings. Loads are taken from a calculation simulation where the engine was run on part load. They consist of loads that affect the front, middle, and rear parts of the bearings, as seen in figure 43.

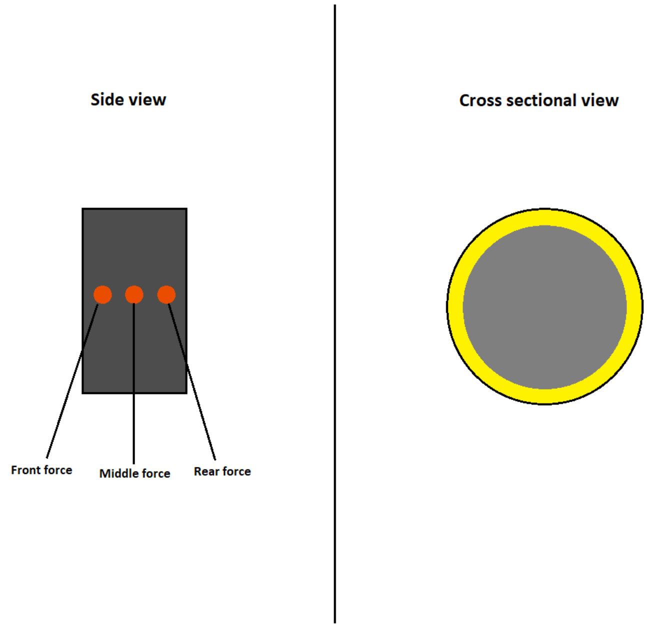

Figure 43: Cam bearing loads

The total force sum is thus the sum of these three forces. An assumption is however made here to be able to translate these forces into x-and y-loads in GT-Suite. The assumption is that the total force is equal to half of the middle load and a quarter of the front and rear load as the following equation shows:

$$F_{tot} = 0.5 \cdot F_{mid} + 0.25 \cdot F_{front} + 0.25 \cdot F_{rear} \tag{17}$$

This equation is assumed to predict the real loads but since its level of accuracy is unknown, it is difficult to know the percentage of error when using these loads in GT-Suite. In other words, if the loads are somewhat inaccurate, the flow into the cam bearings is directly affected, and consequently also the pressure measured at the camshafts' end. A reasonable guess nevertheless is that in the test rig the leakage at the intake cam bearings was higher than at the exhaust cam bearings which evened out the pressure losses that occur from the intake cam inlet to the exhaust cam inlet, thus making the two pressure measured at the cams' end more or less equal to each other, as seen in figure 35.

4.3 Validation results

In this section results for the validation process are shown. The table below shows at which temperatures, engine speeds, and pressure levels the validations are done. It is worth noticing that only the most interesting results are shown here. The rest can be checked in appendix A.

Table 10: Validation parameter space

Engine speed (rpms)	Temperature (°C)	Pressure level (bar)
1000, 3000, 5000	60	High pressure level
1000, 3000, 5000	130	High pressure level
Whole rpms range	90	High pressure level
Whole rpms range	110	High pressure level
Whole rpms range	90	Medium pressure stage 2
750-5500	90	Medium pressure stage 1
750-3500	60	Low pressure level
750-3000	120	Low pressure level

As shown in table 10 below, the model's average relative error during the validation is almost 1% while the maximum relative error lies at 8.51 %. The correlation factor obtained is 99.59%.

Table 11: Model's relative error and correlation

Model's total average relative error in validation (%)	Model's total maximum relative error in validation (%)	Correlation factor in validation(%)
0.99	8.51	99.59

4.3.1 Validation at high pressure mode

Overall similar system behavior is observed in the validation done at 60 and 130 °C and high pressure level as the one seen in the calibration process with the same location of the maximum relative error values. The results for the simulation at 60 C can be found in Appendix A.1.
Taking a look at 130 °C, as seen in figure 44, the first thing to be noticed is that the pump outlet pressure is slightly lower than desired. Regarding the new positions, it can be seen that the pressure at the pump return channel is slightly lower than desired, while the pressure before the cylinder head gasket and VVT is higher than desired.

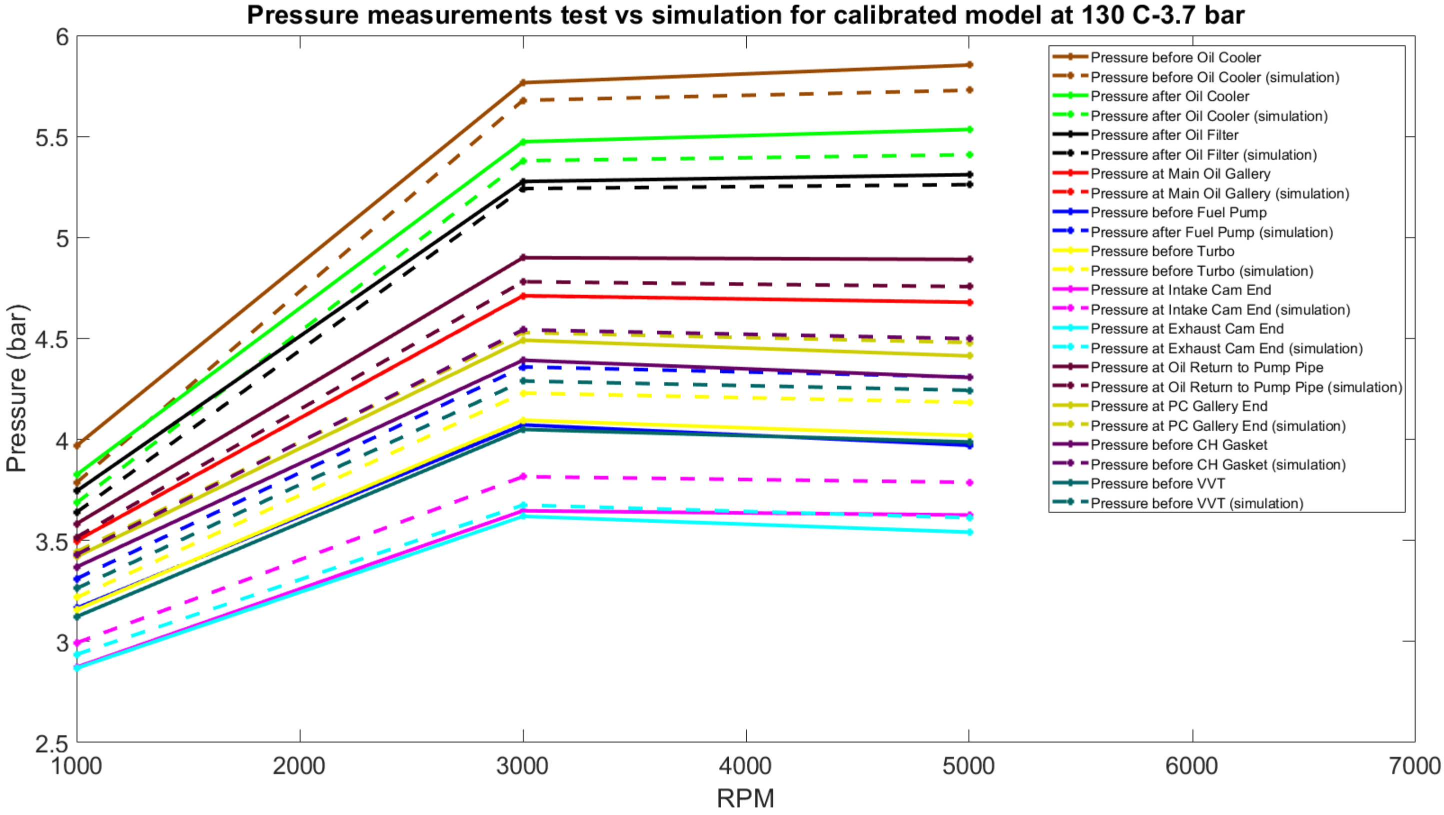

Figure 44: Validation results at 130 °C and high pressure mode

Table 12: Accuracy and Correlation at 130 °C-High pressure

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 130 °C and High pressure	3.05	8.51	99.04

The accuracy results, as shown in table 11, show an average and maximum relative error of 3.05% and 8.51% respectively, with the latter one obtained at the location before the fuel pump. The correlation factor lies at 99.96%.

At 90 °C, the pressure results lie in better agreement with the measurement data than at 60 and 130 °C.

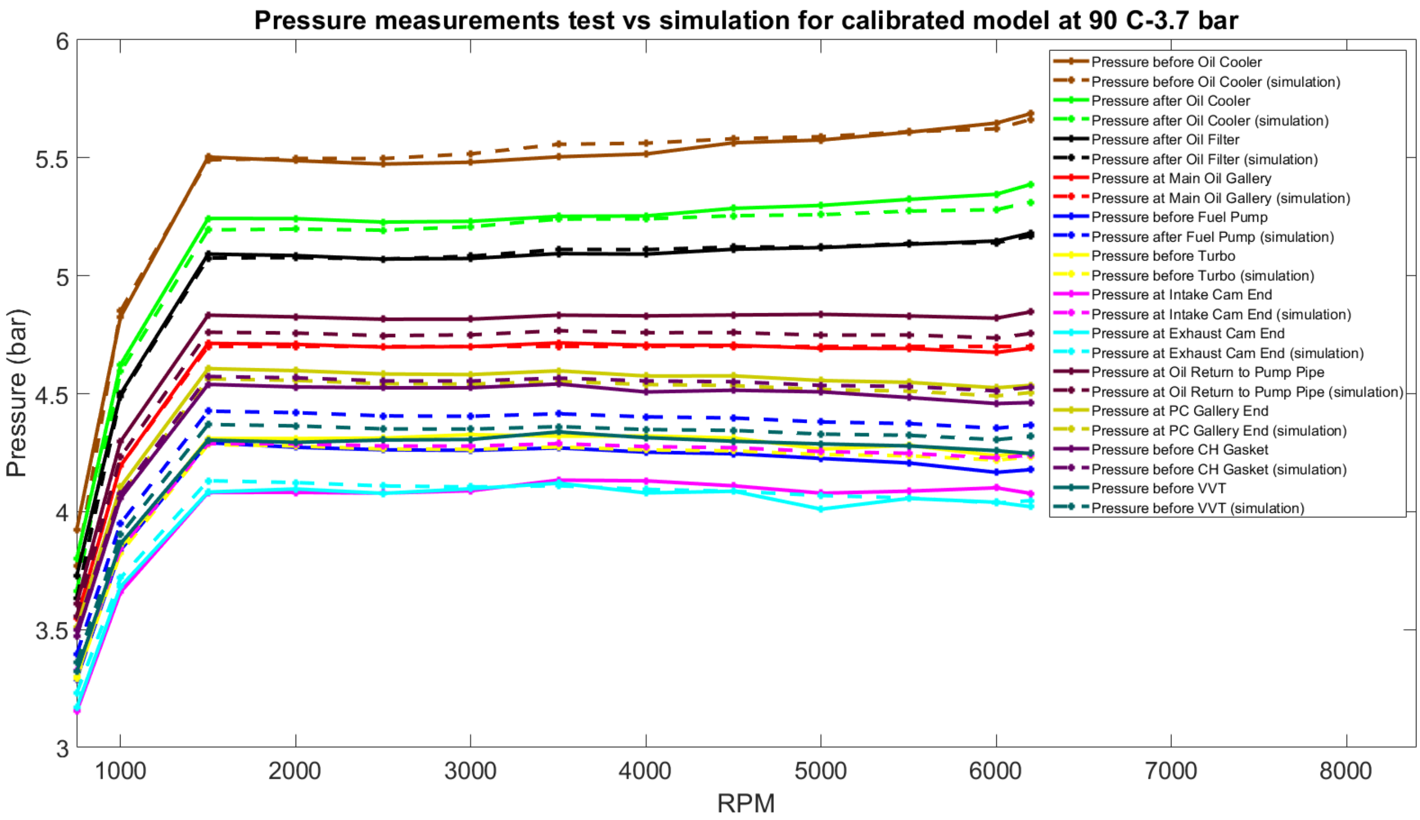

Figure 45: Validation results at 90 °C and high pressure mode

Table 13: Accuracy and Correlation at 90 °C-High pressure

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 90 °C and High pressure	0.45	5.53	99.06

The average relative error lies at 0.45%, with a maximum relative error of 5.53% obtained at the intake cam end, as seen in table 12. The correlation lies at 99.06%.
Going to the oil temperature of 110 °C, more or less the same system behavior is obtained. Results can be seen in Appendix A.2.

4.3.2 Discussion of validation results in high pressure mode

As expected, the same behavior seen in the calibration process is obtained here during the validation at 60 and 130 °C. Engine speed seems hence to play a less important role in the system's behavior when the same pressure level and oil temperature are considered. Concerning the new oil measuring positions that are added, the pump return channel is connected to the pump is the real engine to be able to regulate the pressure at the main gallery. In the model, however, since the pump model doesn't come with a connection that permits the attachment of such a channel, the return channel is modelled with a flow cap at its end that simply represents a dead-end for the flow as seen in figure 46 below.

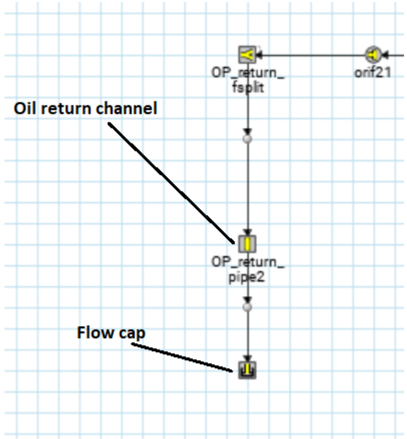

Figure 46: Oil return channel in GT-Suite

An assumption is thus being made which results in an error seen in the pressure results at this position. The nature of the error is the same in that the pressure is lower at all the considered temperatures in the chosen pressure level. The system behavior is slightly different when it comes to the pressure before the cylinder head gasket and the pressure before the VVT. At 60 °C, they are slightly lower, specially when it comes to the pressure before the VVT, but at 110 and 130 °C they are both higher than the target values. At high oil temperatures, the pressure before the head gasket is most likely influenced by the behavior of the intake cam and fuel pump since the head gasket's position is between these two. Taking into consideration that the pressures at the intake cam end and before the fuel pump are over-predicted, the pressure before the head gasket also follows the same pattern. In other words, it can be said that the effect is coming from two positions where the highest errors are obtained. Regarding the pressure before the VVT, a possible explanation to that is that the VVT is described by a map that describes the pressure drop values at 90 °C only. The best accuracy for this position is thus seen at 90 °C. To calculate these values at other temperatures, the software simply performs an extrapolation. As a consequence accuracy is lost when going away from 90 °C.

The lower than desired pressure at the main gallery end obtained at 60 °C is probably an indicator that the flow going into the main bearings is lower than what it should be. Looking at figure 47, the measuring position for this pressure lies right above the feed channel to the main bearing.

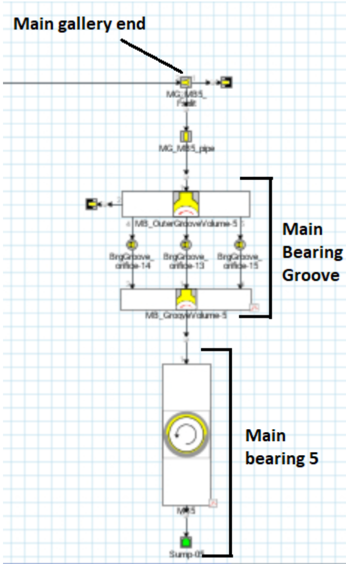

Figure 47: Main gallery end

In other cases, the logical explanation behind this lower pressure would be that the leakage out of the bearing is too high causing a pressure drop. However, considering that the system flow distribution at 60 °C, shown in figure 36 shows that the main bearings are getting lower flow than the big-end bearings, this might be the reason behind the pressure obtained. Has it been higher it would keep the pressure at this point high enough to reach the target value while providing the right flow to the bearing.
Also, one phenomenon that can be noticed is that at low engine speeds, more specifically when the engine speed is 750 rpms at 90 and 110 °C and 1000 rpms at 130 °C, there seems to be a higher relative error at the pump outlet position than at higher engine speeds. The pressure there is lower than the target value as seen in figure 48.

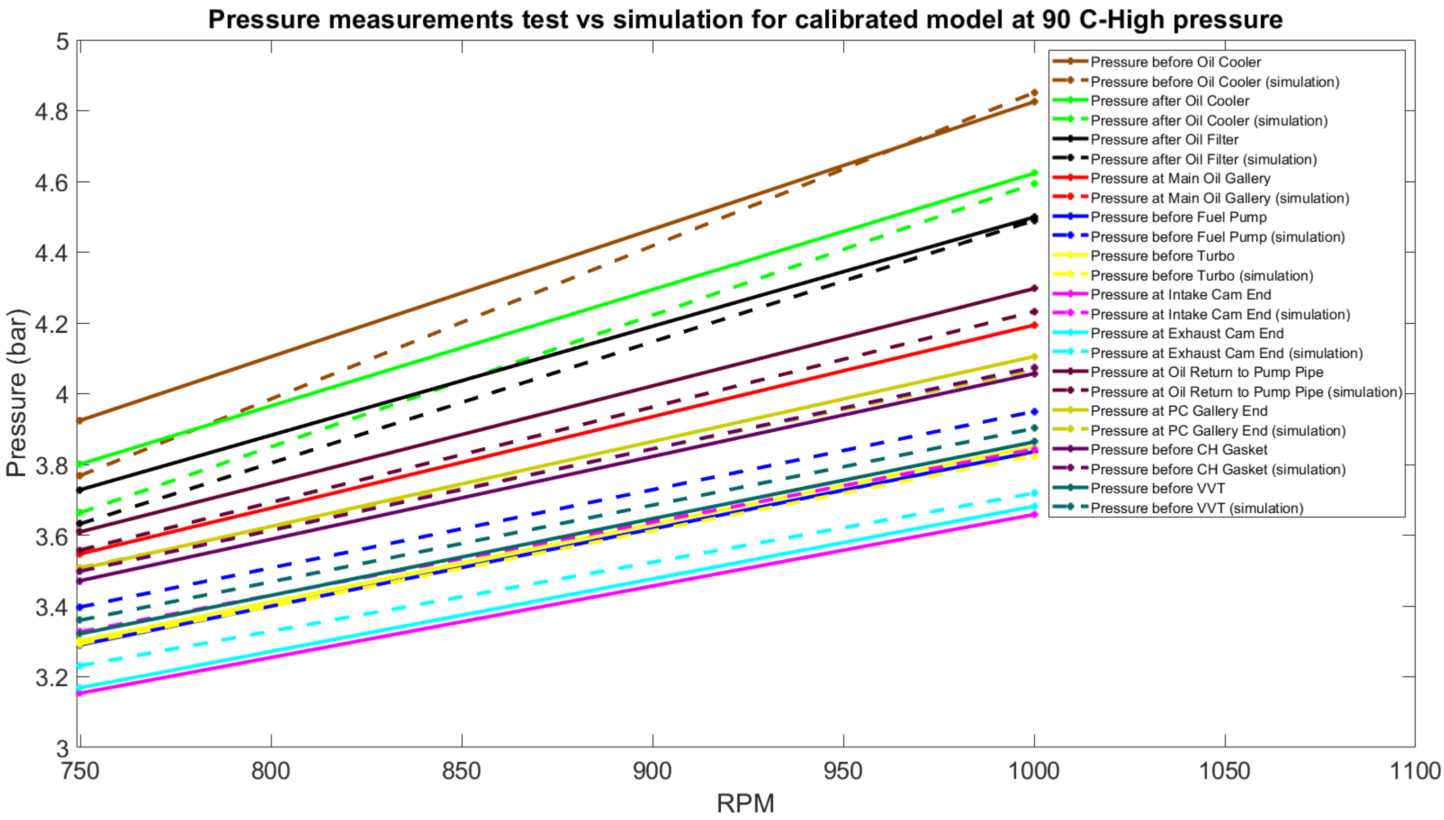

Figure 48: Validation results at 90 °C, high pressure mode and idle speeds

A possible reason can be that since it is observed that the turbo and fuel pump consume almost the same amount of oil regardless of engine speeds, as seen in figures 36 and 37 earlier, their consumption is being too high at idle speeds. At these engine speeds, the pump is less efficient. Thus, it reaches its maximum capacity and isn't able to produce enough flow that otherwise would result in the correct pressure drop between the oil filter and the main gallery, even with the use of the calibrated main gallery inlet orifice.
In general, when the temperature increases, a clear behavior is observed in the system; the pressure at the pump outlet decreases relative to the measured value while the pressure at the main gallery end, before the VVT, before the fuel pump, before the cylinder head gasket, and before the turbo increases relative to the measured value. Lastly, the pressure drop over the oil filter remains relatively constant while the pressure at the intake cam end decreases slightly relative to the measured value but still remains higher at all temperatures.

4.3.3 Validation at medium pressure stage 2 mode

Moving to medium pressure stage 2, which is the second highest pressure level, the validation is performed at 90 °C. The system behavior observed in figure 59 is similar to what is seen at the high pressure level and at the same temperature. This can be checked in appendix A.3. The discrepancies in the measurement data at high engine speeds increase, however, specially positions located farthest from the oil pump.

4.3.4 Validation at medium pressure stage 1 mode

In medium pressure stage 1 mode, the most noticeable behavior is that the pressure after the oil cooler is almost equal to the pressure after the oil filter that is gotten from measurement data, as seen in figure 49. The rest of the pressure values show a similar trend compared to what is seen earlier.

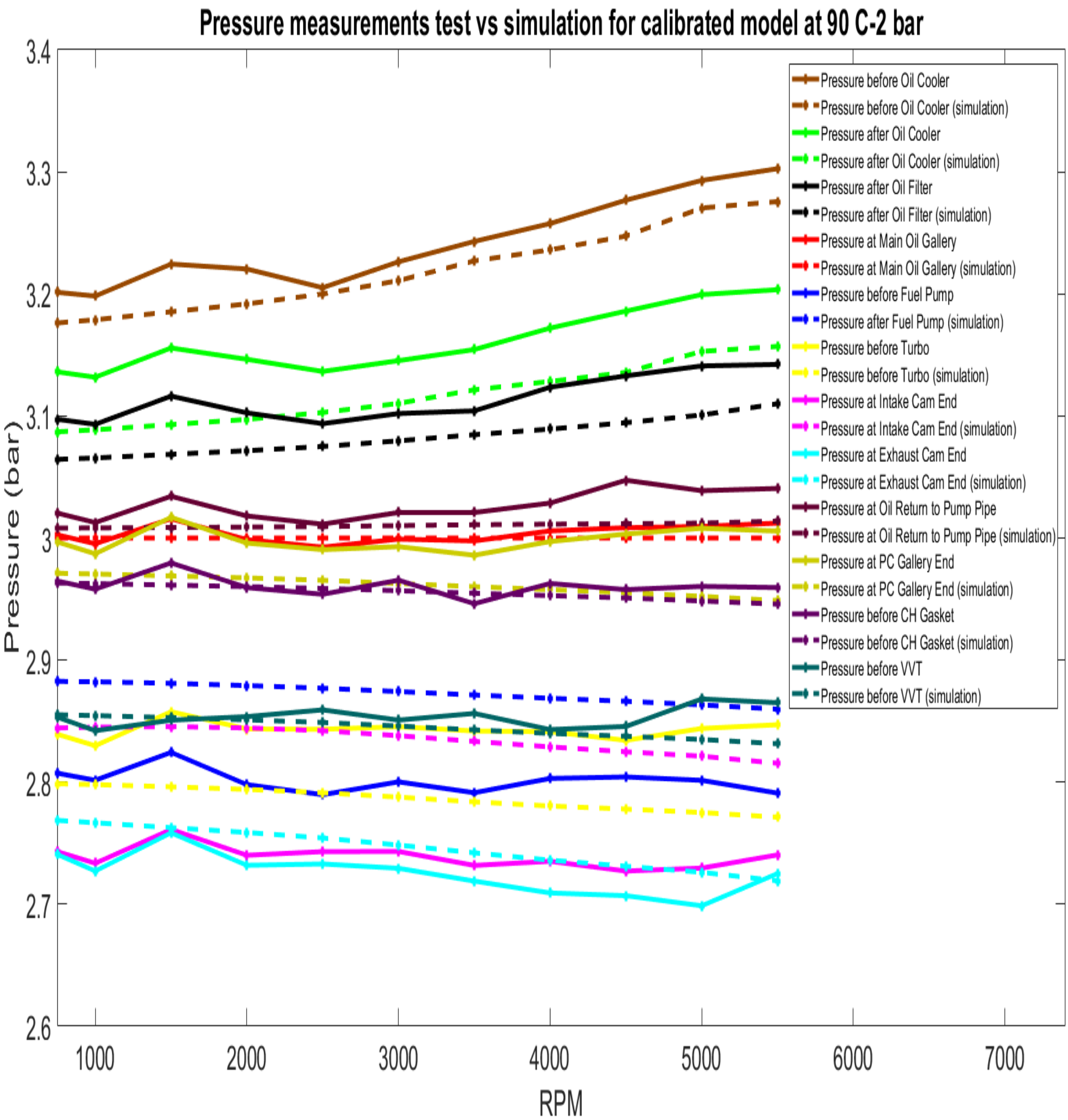

Figure 49: Validation results at 90 °C and medium pressure stage 1 mode

Table 14: Accuracy and Correlation at 90 °C- medium pressure stage 1

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 90 °C and medium pressure stage 1	0.46	4.08	96.63

The average relative error that is received here lies at 0.46% while the maximum relative error is 4.08% at the location of the intake cam end, as seen in table 13. The correlation factor lies at 96.63%.

4.3.5 Discussion of validation results in medium pressure stage 1 mode

It can be noticed that a bigger relative pressure drop over the oil cooler is occurring here at the medium pressure stage 1 compared to what is seen earlier at higher pressure levels. As discussed earlier, the pressure drop map used for the oil cooler in the model is based on a different oil than the one used in the test rig. Having said that, both oils have the same viscosity at the engine operating temperature which is the temperature studied here. Thus, the oil viscosity shouldn't have an effect in this case. To try to understand this phenomenon, it is important to remember that the higher the flow rate that goes through a component, the higher the pressure drop measured. A possible explanation can then be that the flow rate, in this case, is higher than in the test rig. With no flow rate measurements available, it is difficult to know if that is the case. Nevertheless, looking at the flow distribution in figure 50, it can be seen that the two biggest consumers are the fuel pump and turbo. As discussed earlier, the flow at these components should be lower, to say the least. In other words, this can be a reason why the pump is forced to produced more flow, which explains the bigger pressure drop observed.

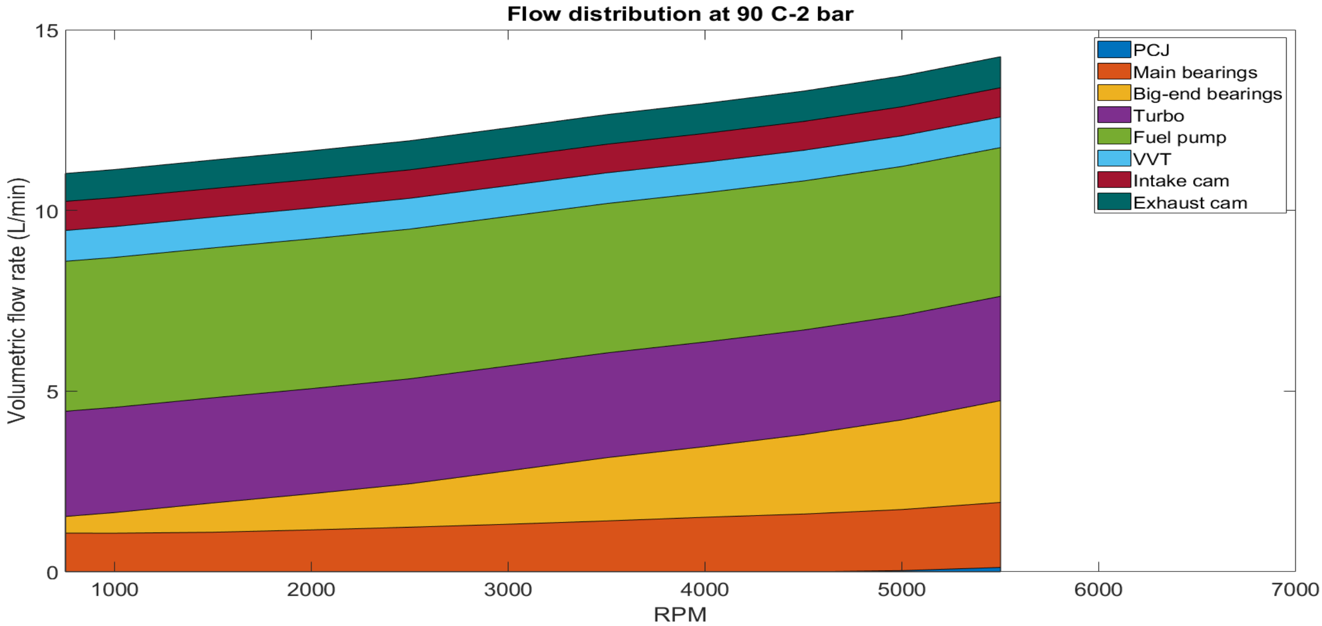

Figure 50: System flow distribution at 90 °C and medium pressure stage 1 mode

It is also possible that there is a lack of accuracy in the pressure drop values provided at a certain flow rate and temperature. In other words, the flow rate produced by the pump and the temperature at the considered pressure level might not exist among the data values used in the component test to generate the pressure drop map. GT-Suite will then have to do an interpolation to find the correct pressure drop values, which results in an increased error.

4.3.6 Validation at low pressure mode

The behavior observed at the low pressure mode, at both 60 and 120 °C, is very similar to what is observed in the calibration results. The results at 60 °C can be seen in figure 51 while the results at 120 °C can be checked in appendix A.4. What is most noticeable however is the pressure fluctuations seen in the measurement data at almost locations.

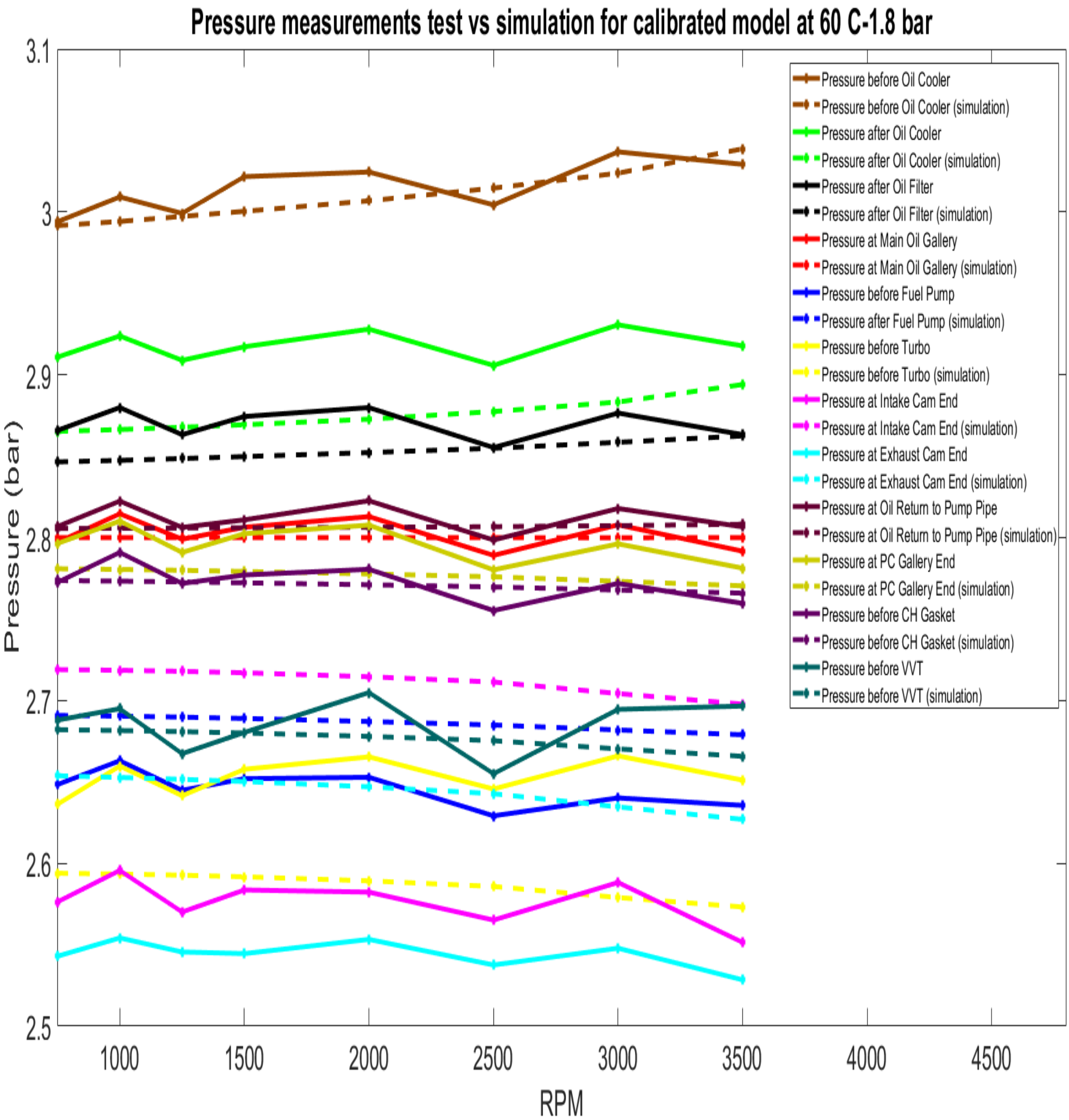

Figure 51: Validation results at 60 °C and low pressure mode

Table 15: Accuracy and Correlation at 60 °C- low pressure

	Average relative error (%)	Maximum relative error (%)	Correlation(%)
Results at 60 °C and low pressure	0.56	5.75	91.58

Table 15 shows that at 60 °C, the average and maximum relative error lie at 0.56% and 5.75& respectively with the location of the maximum error being at the intake cam end. The correlation factor is 91.58%.

4.3.7 Discussion of validation results in low pressure mode

It can be noticed that similar system behavior is being seen at different pressure levels. This is an indicator that pressure level seems to be not as important to consider in the calibration process as oil temperature and engine speed if time is limited and only the most influential factors should be considered. Having said that, oil temperature seems to have the greatest influence on how the system responds in terms of pressure values.
Out of all results, the lowest correlation factor is gotten at this pressure level. This can be mainly attributed to the fluctuations seen in the measurement data. These data can in their turn be attributed to measurement error as mentioned earlier.

4.3.8 Validation at 30 °C- high pressure

In this validation, there are no test-rig measurements at 30 °C to compare against, so this validation is intended to observe the model's behavior outside of the temperature's range 60-130 °C.

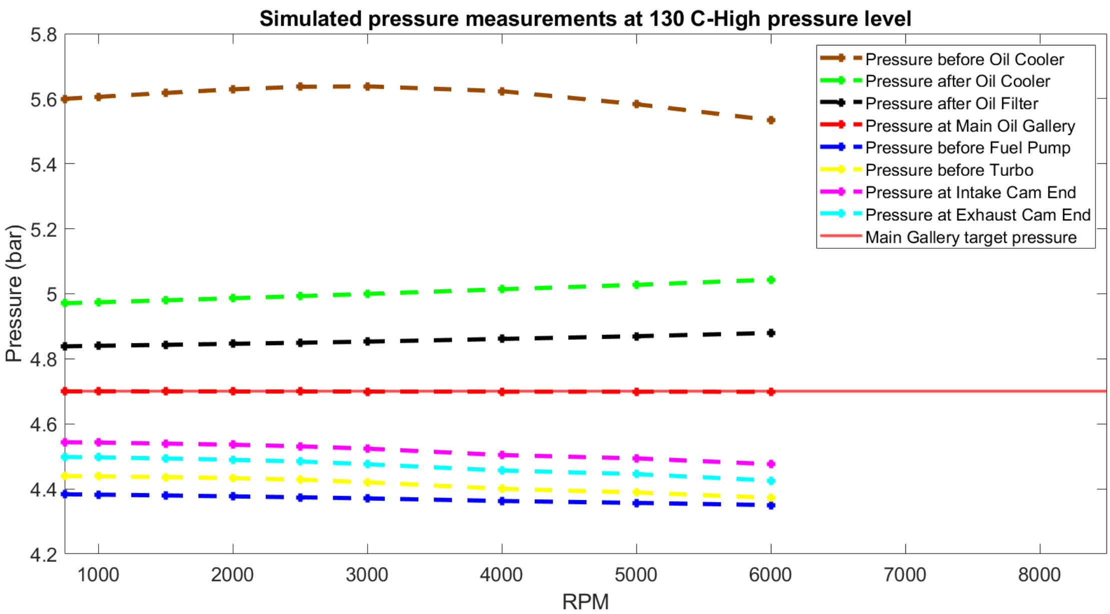

Figure 52: Temperature measurements at 30 °C and high pressure mode

As it can be observed the main gallery high-pressure target is achieved at all rpms. Compared to the closest temperature rig-test measurement available which is 60 °C, where the oil pressure couldn't reach the target at sub 2000 rpms, it can be seen in the 30 °C simulations that even at the lowest rpms the target pressure is achieved. That is mainly due to the viscosity of the oil, which is a function of temperature, the lower the oil's temperature is the higher its viscosity becomes. And in this case at 30 °C, the oil is viscous, significantly more than what it is at 60 °C and the more viscous the oil is the higher its flow resistance becomes and the higher its pressure becomes.
The main gallery target is met throughout all of the rpms range, and the oil's pressures behavior of the various components have similar correlation and behavior as the 60 °C ones. The only exception can be seen in the oil pressure curve before the oil cooler where it's decreasing a bit at the highest rpms. That can be due to the high frictional losses at very high rpms caused by the dynamic sheering of the viscous oil, which takes away from the oil pump's work. When it comes to the other locations such as the cooler and filter, it is observed a very similar and logical pattern when compared to the simulations within the temperature range. The same can be seen regarding the ends of the system such as the exhaust, intake and turbo, where their pressures curves come in similar order as the other simulations, indicating a general and accurate prediction of the oil's behavior outside of the temperature's range.

4.4 Sensitivity study results

In this section, the results of the system sensitivity study are shown. Only the results with the highest error relative to the base simulation are shown.

4.4.1 Bearing clearances and lemon crush parameters

It can be noticed that the system is not sensitive when only one type of bearing clearance is changed. The maximum relative deviation between the studied model and the base model lies between 0.5 and 5% with the average relative deviation being considerably lower. Likewise, similar deviation is obtained when only the lemon crush parameters are varied between their minimum and maximum values.
A considerable difference is first seen when all the bearings are set to their maximal and minimal clearances and lemon crush parameters alternatively. The results are shown in figures 53 and 54 below. It can be seen that when the bearing parameters are set to their maximum values, the pressure at the pump outlet is considerably higher than in the base model. On top of that, the pressures at the intake and exhaust cam end are lower than in the base model. The average relative deviation from the base model lies at 1.77 % while the maximum relative deviation is 6.32% located at the intake cam end.

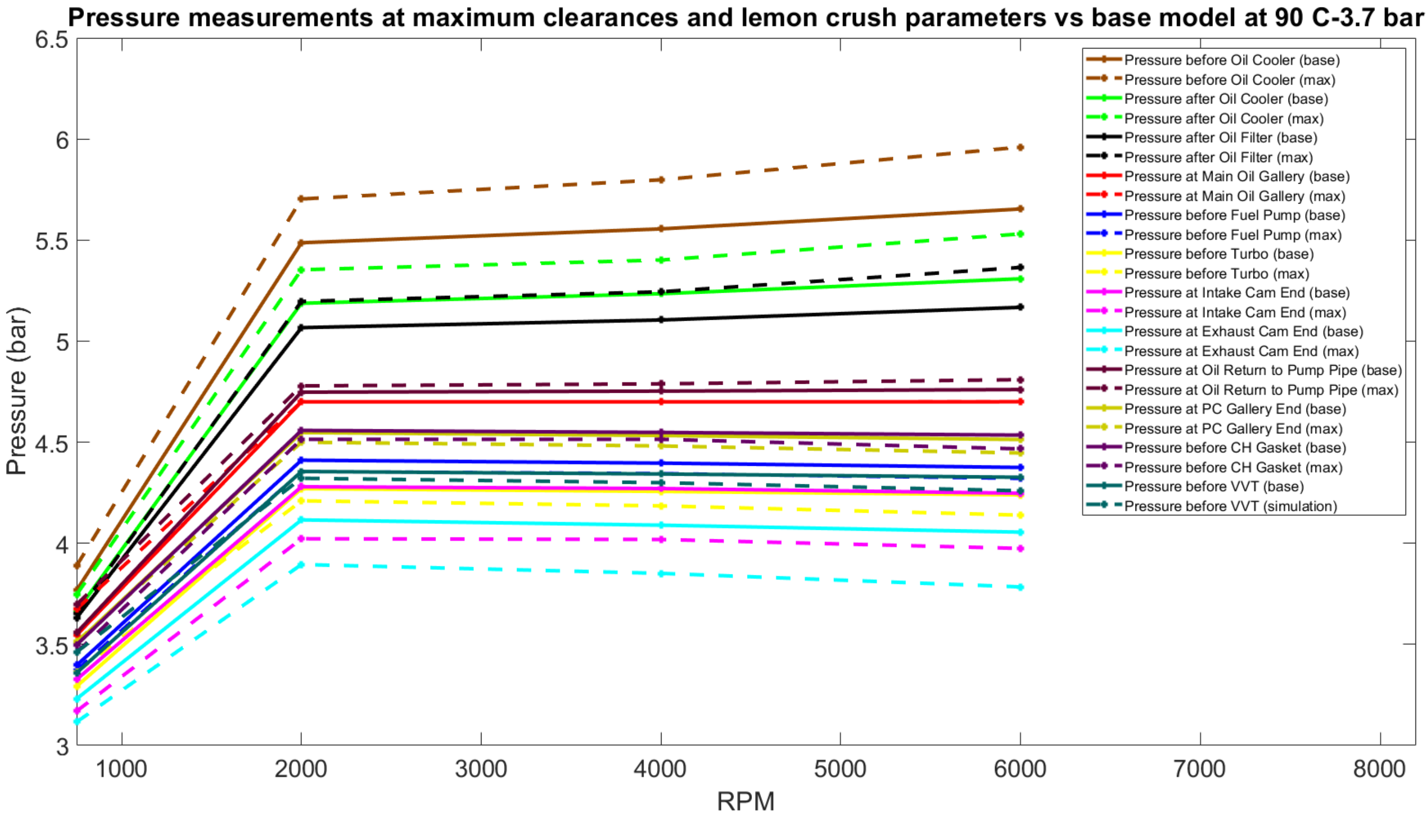

Figure 53: Maximum clearances lemon crush vs base model

When the bearing parameters are set to their minimum values, however, the pressure at the pump outlet is lower while the pressures at the intake and exhaust cam end are higher than in the base model. The average relative deviation is 0.98 % while the maximum relative deviation is 3.62% located at the exhaust cam end.

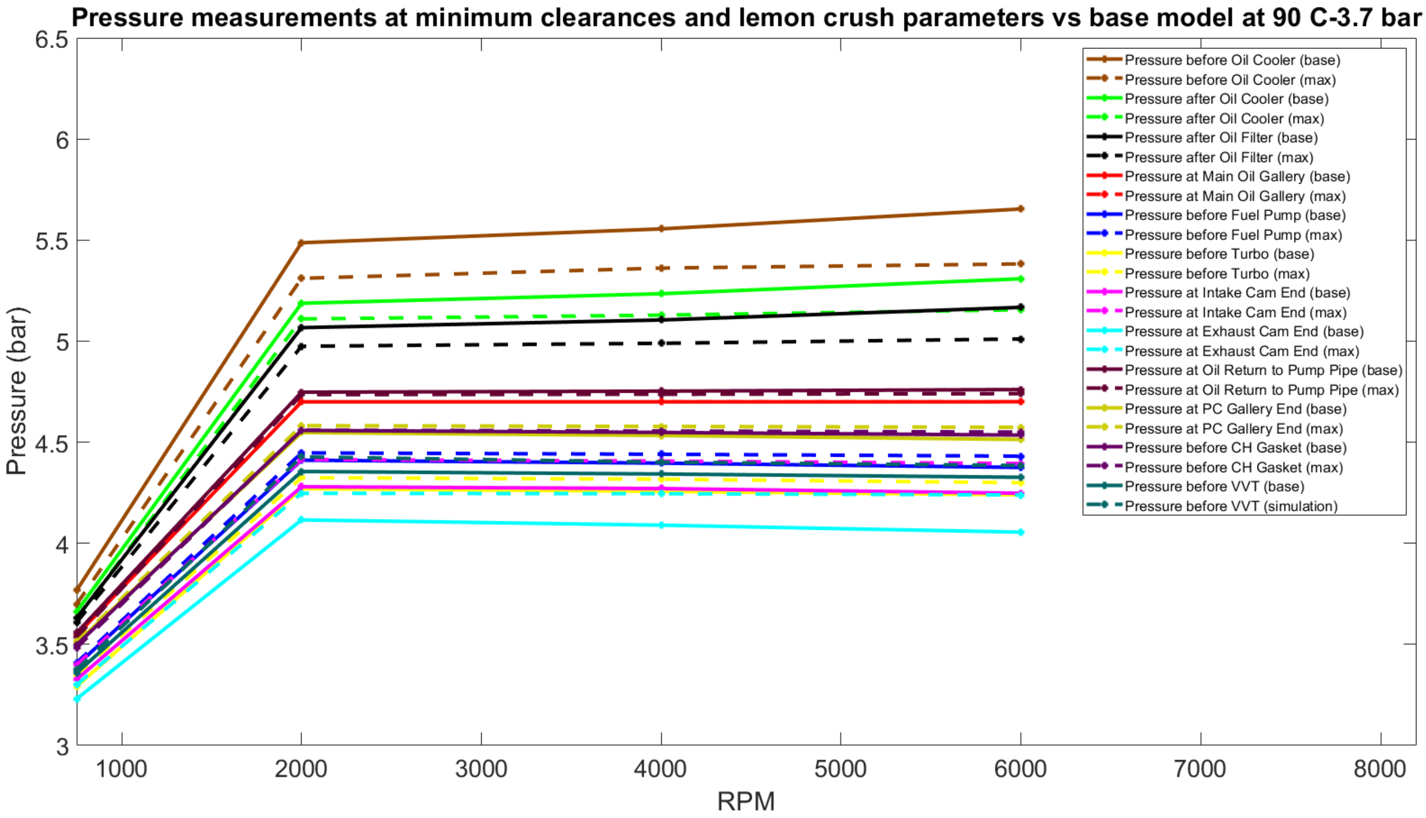

Figure 54: Minimum clearances lemon crush vs base model

4.4.2 Engine load

Concerning the engine load, no significant deviation is observed at both 90 °C and 130 °C. The average deviation lies between 0.2 and 0.35 % while the maximum deviation goes up to 1 %, with the pressure values being slightly higher than the values in the part-load model. Many reasons can be explained as to why this result is seen, but generally speaking, main bearings are mostly sensible to temperature changes and not as much to engine load. Temperature plays a huge role on bearing clearance, which increases proportionally with temperature, along it, oil flow and leakage increase. Another reason is that the oil pump, is a variable displacement pump, which means, as mentioned earlier, that it adjusts it's flow rate to match the required main gallery pressure, so at part-load and 90 °C the pump is not working at full capacity, while at full-load and same temperature, the oil pump will increase its flow to the maximum capacity, thus increasing the main gallery pressure and counteracting the high bearings load.
Moreover, since the fuel pump and turbo are modelled as orifices, consuming a higher than intended amount of oil, influences the actual flow and pressure at the main gallery and the end of it as well, explaining even more the similar values and correlation between part and full load simulated oil pressures.

4.4.3 Aeration

A study was made on GT-Suite models to analyze the influence of oil aeration on the oil pressures.

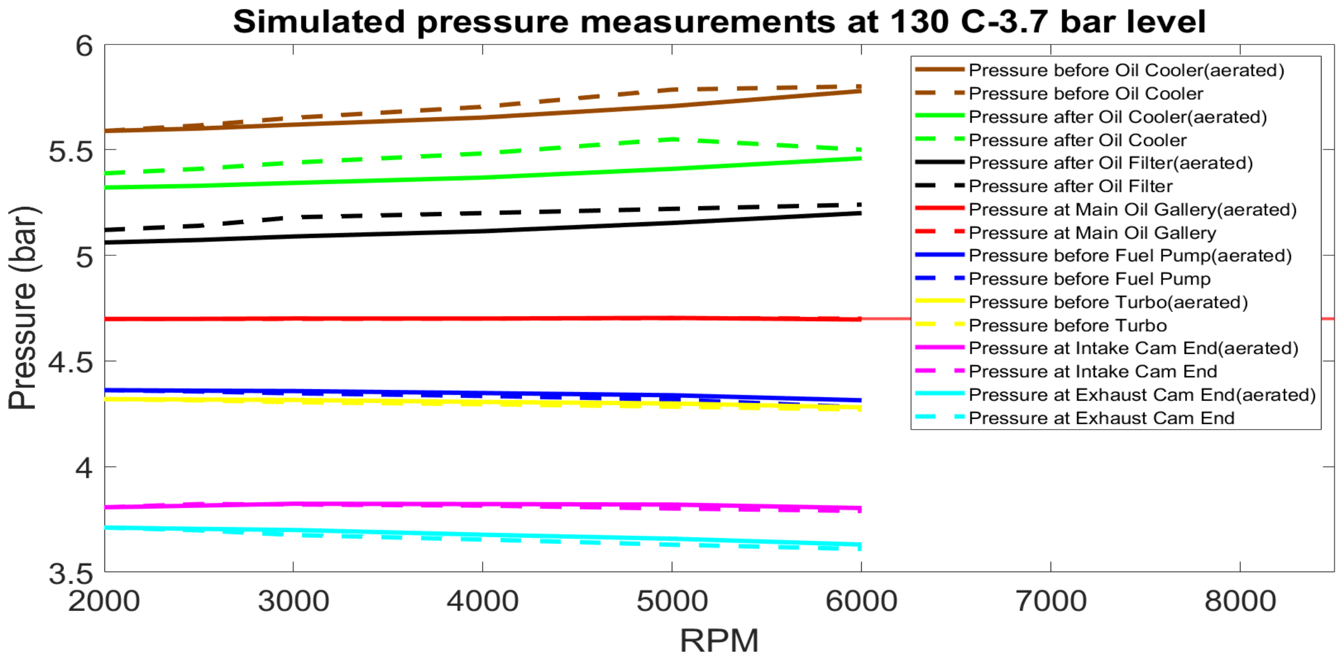

Figure 55: Validation results at 130 °C aerated conditions and high pressure mode

As it can be seen in the figure above, the pressure results of the aerated oil are very similar to the non-aerated oil pressure values at the same temperature of 130 °C. At pre-main gallery locations, such as at the oil cooler and filter, it is observed that the aerated model has slightly lower oil pressure along all the rpm range, something that is expected but is under-predicted by the model. At post-main gallery locations, the oil pressures of both the aerated and non-aerated models are almost equal.
The pressure values correspond to rpms of 2000 and above as no measurements were taken at rpms below 2000. What can be concluded from the graph is that the oil model doesn't seem to be highly sensitive to aeration especially at high rpms, where oil pressures are not really affected. A possible reason for is that the free air in the oil, usually in form of bubbles, will almost entirely disappear when it gets pressurized through the pump and oil channels and will only reappear when pressure drops to near or below atmospheric level which is not the case in most of the studied engine's components with the exception of the oil sump. However at high levels of aeration, especially at high rpms, oil pressure can decrease, something that is not entirely or correctly predicted by the model which could be one of the model's weaknesses. Also by only looking at pressure values, one will not be able to predict other potential effects of aeration, such as higher wear to sensitive oil consumers like bearings and hydraulic lash adjusters which can be affected by aerated oil even when completely dissolved.

4.4.4 Discretization length study

In this sensitivity study, the discretization length has been varied from 0.5 mm up to 70 mm. The four cases are run simultaneously, thus the case that takes the longest time dictates the total simulation run time. In terms of accuracy, the difference is negligible; when lengths of 0.5, 1 and 3 mm are used an increase of 0.01% in accuracy is observed and when lengths between 30 and 70 mm are used a decrease of 0.02% in accuracy is seen.

In terms of simulation run-time, there is a significant difference observed when different lengths are used.

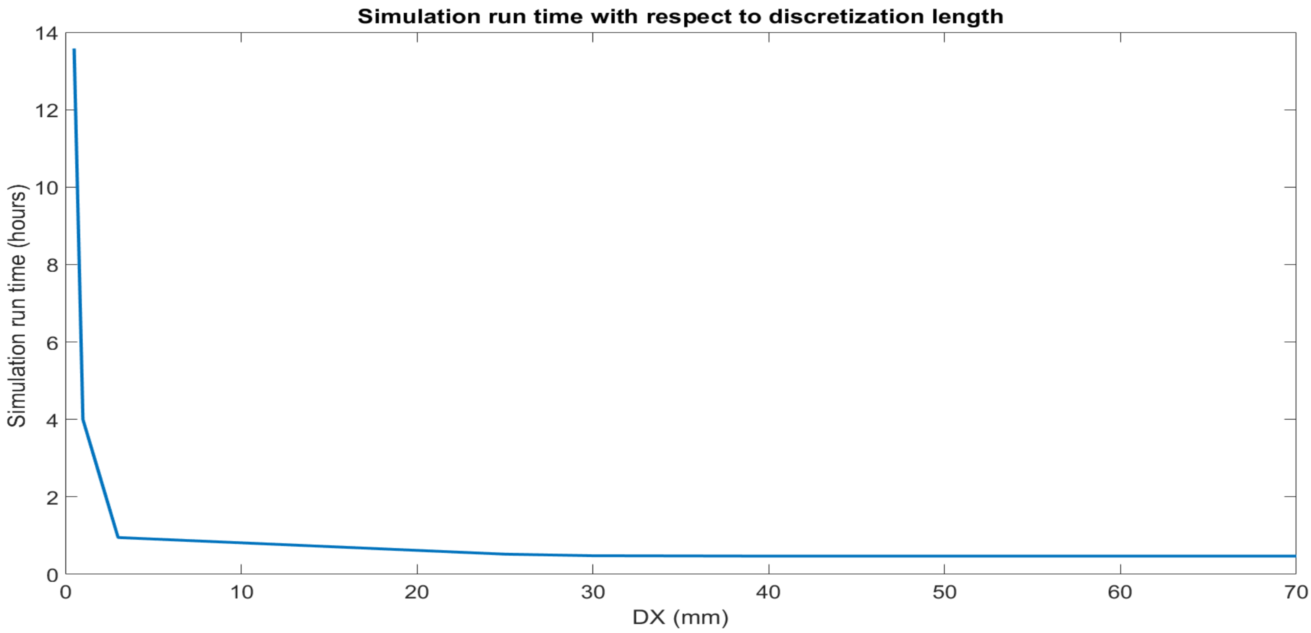

Figure 56: Simulation run time with respect to discretization length

Looking at figure 56 it can be seen that using a length of 0.5 mm, the simulation takes about 14 hours to complete. Moving to 1 mm, there is a massive decrease in run time to about 4 hours. Using a length of 3 mm, the run time becomes one hour. With the standard length of 4 mm used for this thesis work, the run time is 31 minutes. After 30 mm, a decrease in run time is no longer observed.

4.4.5 Discussion of the sensitivity study

As observed in the results, the model hasn't turned out to be sensitive to the variation of bearing clearances in general. That is because the pump adjusts the flow depending on the system flow requirement. When the clearances are increased, the flow rate produced increases in such way that the target pressure is maintained at the main gallery. With all the clearances and lemon crush parameters set to their maximum, the pump is still able to produce enough flow, but the flow rate is now so high that the pressure at the positions before the main gallery increase. The opposite applies when minimum values are used; the pump decreases the total flow rate which causes a decrease in the pressure before the main gallery.
The system seems to be more sensitive to the cam bearings clearance that is being used. Taking into consideration that the pressure is measured at the end of the camshafts which are one of the farthest placed locations in the system relative to the main gallery, the pressure is affected by both the losses that occur in the channels and the bearings leakage. Thus, it can be seen that the pump isn't able to produce enough flow all the way up to the camshafts in a way that would maintain the pressure at the end of these two locations. When minimal clearances are being used, the pressure at the previously named locations is now higher even with the now lower total flow provided by the pump. That is due to the now considerably restricted leakage in all of the eight camshaft bearings. Furthermore, the locations that are located near the main

gallery, like the main gallery end or before the fuel pump, don't seem to be affected much by the change in clearances with regards to pressure. That is because the pressure in the main gallery is still remaining at its target level combined with the fact that the magnitude of the friction losses is diminished as a consequence of the location proximity.
Nevertheless, keeping in mind that the total flow in the system is most probably lower than what it should be, the pump can come closer to its maximum efficiency if the correct flows are to be predicted. In other words, if the sensitivity study is to be carried out with the new flow predictions, the pump might not be able to provide the required flow which can mean larger deviations in the pressures measured. The system might then become more sensitive to what clearances are being used.
An increased engine load leads to increased bearing clearances. Thus, as explained earlier in this section, the pump will then produce a higher flow rate to satisfy the new system requirement. That is probably why no significant deviation is observed. Another possible explanation, is that the rotational inertia of the crankshaft, caused by the crankshaft's mass, overcomes and prevails over the engine load's effect on the bearings' oil film. However, at low pressure modes, the load's effect could be noticeable to some degree. Nevertheless, at full load conditions, the oil pressure will be at its highest level meaning that the scenario of full load and low oil pressure is unrealistic.
Regarding the discretization length, it can be concluded that the choice of the length used barely affects the accuracy. The choice can thus be made based primarily on the computational power available. With that said, a greater length can be chosen in order to cut down on development time, which in its turn translates to sparing resources.

5 Objectives

The work of this thesis has generated a general, improved and inclusive GT-suite oil model that accurately predicts the oil's pressure's behavior at a broad temperature range, wider than what it was calibrated for. The model was successfully made less dependent on calibrating parameters by reducing them from nine to only two while not exceeding the computational run time of one hour. Moreover, a documented calibration method procedure was created. Although for further simulations, flow data is needed this model could be easily used from transient modelling, in this way this thesis work has met all of its objectives.

6 Conclusion and future work

It can be concluded that pressure measurements by themselves are not enough. Flow measurements are crucial for accurate calibration of the model and correct system behavior. Specifically, measurements of the pump outlet flow are needed to perform the calibration in the best possible way. Measurements for the main and crank bearings leakage are also needed as they are one of the biggest oil consumers in the system, thus they influence the system behavior in terms of pressure at different locations and flow distribution. Since these are hard to obtain, a computer software program can be used to calculate and predict the 3D flow into these bearings. When the flow numbers are obtained, lemon crush parameters can be calibrated to reproduce the same flow numbers. A thermal model should ideally be incorporated into the system as well, to improve its behavior and better describe what's happening in a real engine lab.

Orifice diameters are a simple approach to model components that lie at the system's ends. However, they don't reproduce the correct pressure and flow behavior of these components specially when different oil temperatures are considered. Depending on the computational power available, detailed model or pressure drop maps can be opted for.

Moreover, it can be concluded that accurate component tests are needed in case pressure drop maps are expected to accurately predict the pressure losses in the system, as differences in the test implementation in terms of the oil or measuring device used and procedures that are not taken into account in the model can lead to an inaccurate performance.

However, as mentioned before, this work has been done on a one-dimensional model that describes and predicts the behavior of a complex three-dimensional flow. It has thus been expected that the results, no matter how accurate they are, would not be able to fully predict the fluctuations, pressure deviations, and flow values.

Having said that, the calibration process performed can be considered successful, given the available measurement data. The number of parameters has been greatly reduced from 9 to 2 calibration parameters. Moreover, one set of values for these parameters has been used for all temperatures, pressure levels, and engine speeds, while still retaining acceptable accuracy and correlation. Nevertheless, a more accurate flow behavior is needed for the model to be used in transient simulations, test measurements for clogged oil filter conditions generating data that could be used in Gt-suite and for even better accuracy, a detailed aeration modelling should be considered.

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A Appendix

A.1 Validation at 60 °C and high pressure mode

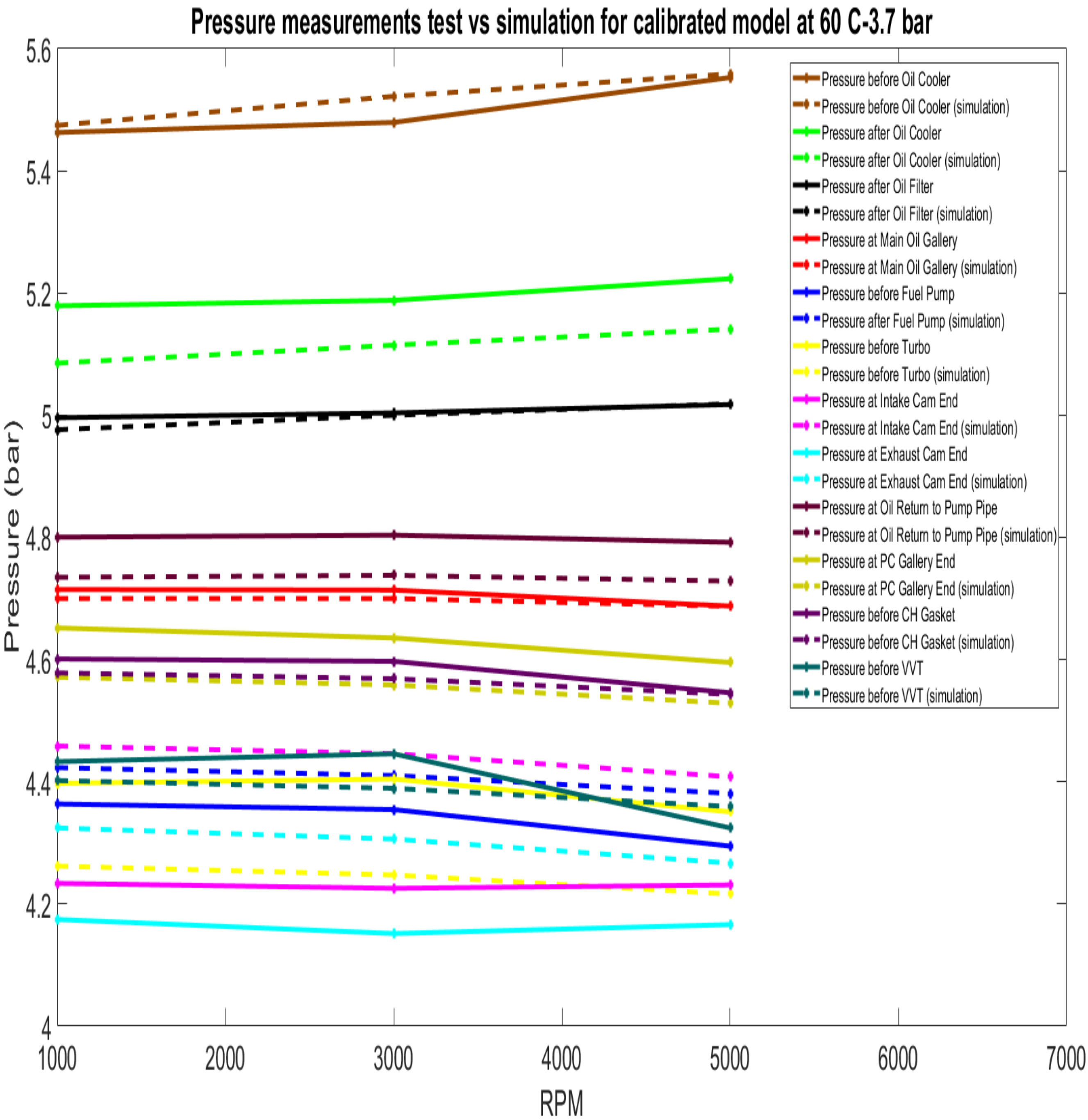

Figure 57: Validation results at 60 °C and high pressure mode

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 60 °C and High pressure	1.63	5.31	97.47

Table 16: Accuracy and Correlation at 60 °C-High pressure

A.2 Validation at 110 °C and high pressure mode

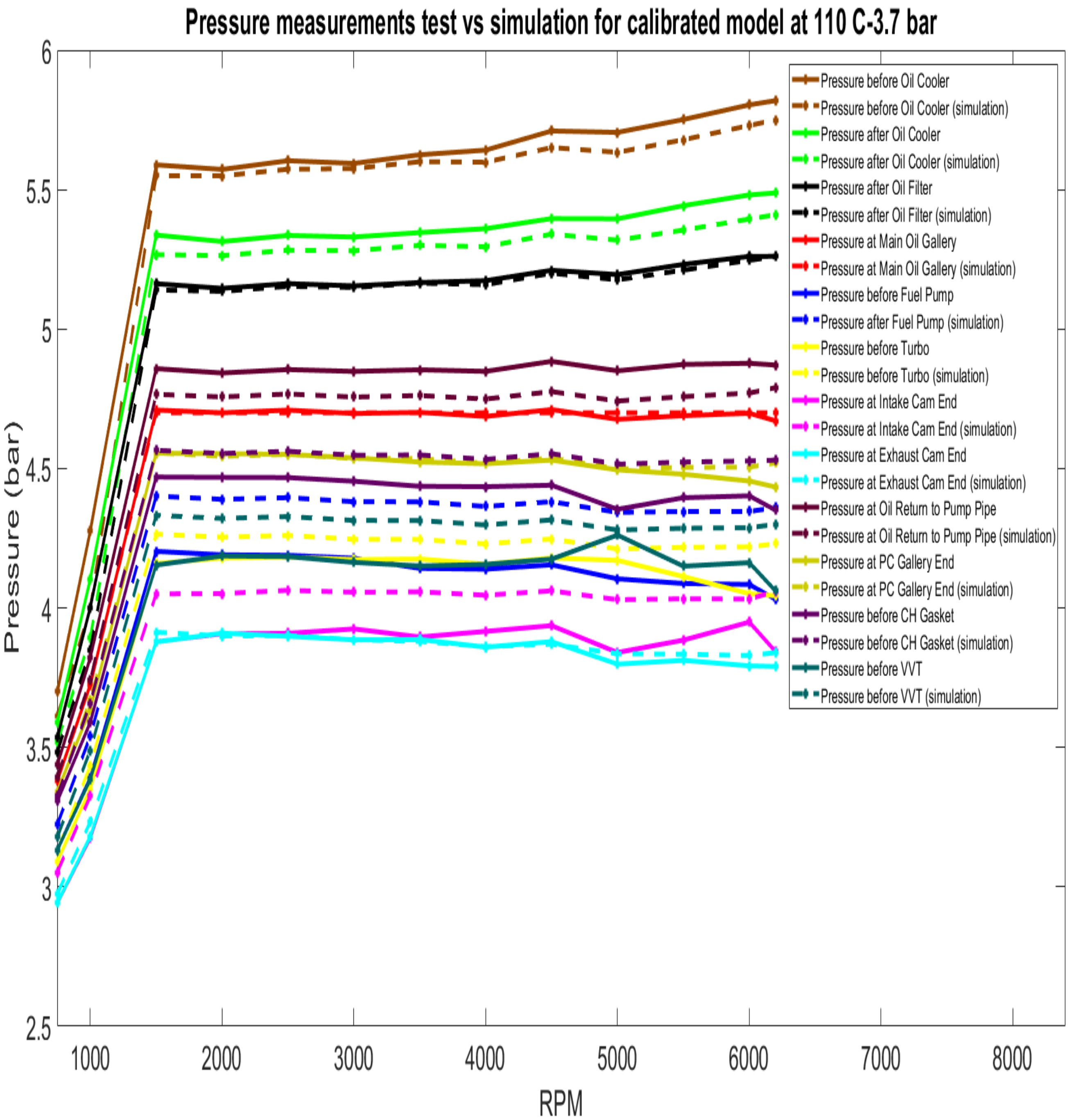

Figure 58: Validation results at 110 °C and high pressure mode

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 110 °C and High pressure	0.65	6.54	98.95

Table 17: Accuracy and Correlation at 110 °C-High pressure

A.3 Validation at 90 °C and medium pressure stage 2 mode

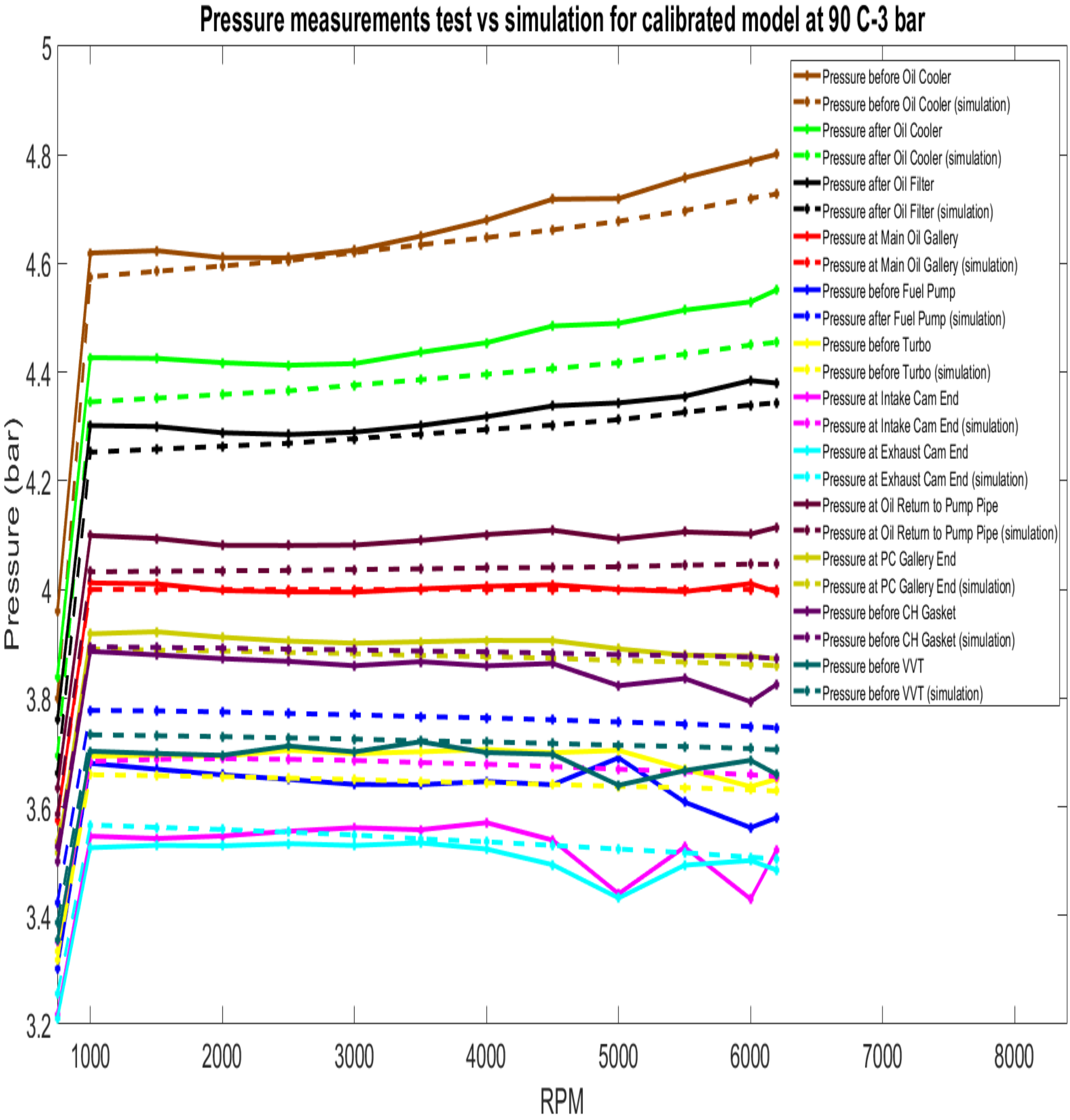

Figure 59: Validation results at 90 °C and medium pressure stage 2 mode

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 90 °C and medium pressure stage 2	0.48	4.17	98.63

Table 18: Accuracy and Correlation at 90 °C- medium pressure stage 2

A.4 Validation at 120 °C and low pressure mode

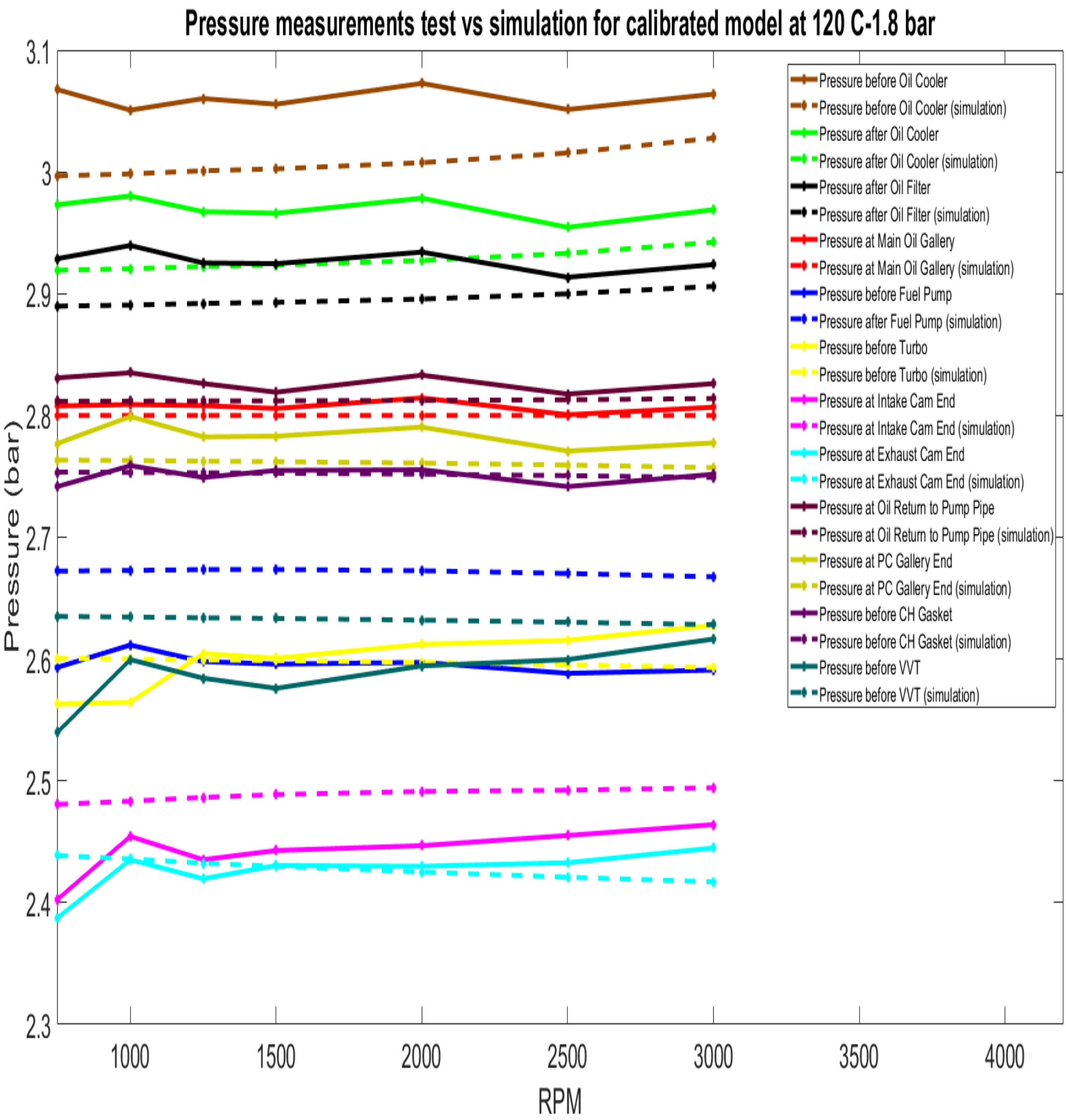

Figure 60: Validation results at 120 °C and low pressure mode

	Average relative error (%)	**Maximum relative error (%)**	**Correlation(%)**
Results at 120 °C and low pressure	0.62	3.75	98.77

Table 19: Accuracy and Correlation at 120 °C - low pressure

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