



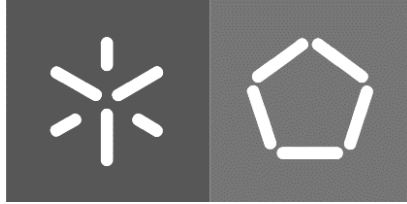
Universidade do Minho

Escola de Engenharia

Francisco Manuel Mendes Gomes

**Enhancement of proportional solenoid valve to
manage variable displacement oil pump.**

Maio de 2024



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to manage variable displacement oil pump.**

Dissertação de Mestrado Integrado em Engenharia
Mecânica

Trabalho efetuado sob a orientação dos

Professor Doutor Jorge José Gomes Martins

Professor Doutor Francisco Carrusca Pimenta de Brito

Maio de 2024

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AGRADECIMENTOS

Ao professor doutor Jorge Martins e ao professor doutor Francisco Brito pela orientação do trabalho, pelos esclarecimentos e pelas correções necessárias.

À Bontaz Portugal pela oportunidade de realizar o estágio e pelas ótimas condições oferecidas para a sua realização.

Ao meu orientador na Bontaz, engenheiro André Santos, pela disponibilidade e ajuda valiosas à minha adaptação e ao desenvolvimento das atividades no âmbito do meu estágio.

A cada membro da equipa de R&D da BCTP que contribuiu com um enorme apoio ao longo do trabalho, seja em discussão de ideias, ajuda técnica, integração e convívio, e por me transmitirem a vontade de aprender sempre mais.

Aos técnicos António Malheiro e Pedro Correia, pela paciência louvável e auxílio na componente prática, imprescindível à realização deste trabalho.

Ao engenheiro Tomé Barreiro, pela importante contribuição no âmbito da conceção do modelo de simulação e compreensão do sistema em estudo.

Aos meus pais, por me darem as condições para chegar até aqui e me ajudarem desde o princípio.

À minha irmã, por me acompanhar e apoiar sempre de perto, independentemente da distância.

Ao meu avô, a quem devo grande parte do meu conhecimento prático, e à minha avó, por me apoiarem desde sempre.

À minha restante família, pelo apoio em todos os momentos.

À Margareida, pela companhia e motivação que tornam o dia a dia mais fácil.

Aos meus amigos, pelo companheirismo e interajuda indispensáveis ao longo destes anos, que hoje fazem também parte de quem eu sou.

STATEMENT OF INTEGRITY

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ABSTRACT

The present work was done within BONTAZ – BCTP Equipamento Automóvel with the aim of optimising a solenoid valve used to manage an automotive variable displacement oil pump (VDOP). The system works by using the solenoid valve to hydraulically regulate the pump to provide less oil flow, lowering its engine torque demand, thus reducing fuel consumption and emissions. The product in question has been facing problems during customer end of line (EoL) tests, regarding oscillation in the engine oil pump's output pressure, resulting in a percentage of the parts being considered not OK (NOK). To avoid reproducing the issue in the newer generation of the solenoid valve, this work serves as a tool to understand the problem faced by the customer and take corrective measures.

Firstly, a revision of the literature was carried out, in order to accurately describe the functioning of oil pumps, and more specifically variable displacement types. Additionally, the functions of different types of solenoid valves and their control systems were enumerated.

Secondly, the methodology followed in the execution of the work was established, starting from a brief description of the system in study, followed by a product and process analysis, where the documents related to production of the solenoid valve are presented. There is an explanation of the functional tests employed on the solenoid valves, as well as the test benches where these were done, along with the software used to analyse the performance curves. Some changes had to be done to the pump test bench, which are also shown.

From a problem-solving point of view, quality tools such as an Ishikawa diagram and the evaluation matrix were applied in order to provide a well-guided approach to the problem and assist in finding the root cause. The tests and analysis done were based on the possibilities presented here, which were gathered from the research and brainstorming within the Bontaz Portugal R&D team.

Furthermore, the present work provided Bontaz with the tools to create a simulation model of a solenoid valve operated variable displacement vane oil pump, which can be used to develop new products and foster internal knowledge on VDOP systems.

Lastly, the experimental results from the different tests are presented and discussed.

KEYWORDS

Ishikawa; Proportional hydraulic solenoid valve; Root cause analysis; Variable displacement oil pump.

RESUMO

O presente trabalho foi desenvolvido na BONTAZ – BCTP Equipamento Automóvel, com o objetivo de otimizar uma electroválvula utilizada para gerir uma bomba de óleo de caudal variável para automóvel. O sistema funciona através da utilização duma electroválvula para regular hidraulicamente a bomba para fornecer menos caudal, reduzindo o seu consumo de binário e, por isso, reduzindo consumo de combustível e emissões. O produto em causa tem enfrentado problemas durante os testes End of Line (EoL) do cliente, relacionados com oscilação da pressão de descarga da bomba, o que resulta numa percentagem das peças serem consideradas não OK (NOK). Para evitar reprodução deste problema na nova geração da electroválvula a ser desenvolvida, este trabalho serve de ferramenta para a compreensão do problema encontrado pelo cliente e para tomar ações corretivas.

Em primeiro lugar, é feita uma revisão à literatura, de forma a descrever o funcionamento de bombas de óleo e, mais especificamente, dos tipos de caudal variável. Adicionalmente, as funções de diferentes tipos de electroválvulas e os seus sistemas de controlo são enumerados.

De seguida, a metodologia seguida na execução do trabalho é estabelecida, começando por uma breve descrição do sistema em estudo, seguida duma análise de produto e projeto, em que os documentos relacionados com a produção da electroválvula são apresentados. São explicados os testes funcionais feitos às electroválvulas, bem como os bancos de ensaio em que estes se realizam e o *software* em que se analisam as curvas de desempenho. Algumas alterações que tiveram de ser feitas ao banco de ensaios são expostas.

Do ponto de vista da resolução do problema, ferramentas de qualidade como um diagrama Ishikawa e a matriz de avaliação foram aplicadas de modo a seguir uma análise cuidada ao problema e dar auxílio na descoberta da causa raiz. Os testes e análises feitas foram baseados nas possibilidades neles apresentadas, que foram recolhidas através de pesquisa e brainstorming no seio da equipa de R&D da Bontaz Portugal.

Além disso, o presente trabalho equipa a Bontaz com as ferramentas necessárias à criação de um modelo de simulação de bombas de óleo de caudal variável equipadas com electroválvula, que poderá ser utilizado para o desenvolvimento de novos produtos e para fomentar conhecimento interno sobre bombas de volume variável.

Por fim, os resultados experimentais dos diferentes testes são apresentados e discutidos.

PALAVRAS-CHAVE

Análise de causa raiz; Bomba de óleo de caudal variável; Eletroválvula hidráulica proporcional; Ishikawa.

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LIST OF SYMBOLS, GREEK LETTERS, AND ACRONYMS

SYMBOL	DEFINITION	UNIT
A_s	Surface area	[m ²]
A_{sec}	Cross-sectional area	[m ²]
B	Magnetic flux density	[T]
D	Solenoid valve piston's large diameter	[m]
D_p	Pipe diameter	[m]
d	Solenoid valve piston's small diameter	[m]
F	Force	[N]
f	Discharge frequency	[Hz]
f_D	Darcy friction factor	
g	Gravitational acceleration	[m/s ²]
H	Magnetic field strength	[A/m]
h	Height	[m]
I	Electric current	[A]
l	Wire length	[m]
N	Number of coils	
p	Pressure	[bar]
p_x	Pressure at point "X"	[bar]
Q	Volumetric flow	[m ³ /s]
R	Electrical resistance	[Ω]
V	Voltage	[V]
v	Velocity	[m/s]
Z	Rotational speed	[rpm]
GREEK LETTER	DEFINITION	UNIT
Δp	Pressure differential	[bar]
μ	Magnetic permeability	[N/A ²]
μ_0	Magnetic permeability of the vacuum	[N/A ²]
ρ	Density	[kg/m ³]

ACRONYM	DEFINITION
BCTP	BONTAZ Portugal
BCUS	BONTAZ United States of America
BEV	Battery Electric Vehicle
CAD	Computer-Aided Design
EGR	Exhaust Gas Recirculation
EoL	End of Line
EU	European Union
FLR	First Level Regulation
HVAC	Heating, Ventilation, and Air Conditioning
NOK	Not Okay
OEM	Original Equipment Manufacturer
PCJ	Piston Cooling Jet
PD	Positive Displacement
PEEK	Polyether Ether Ketone
PHEV	Plug-In Hybrid Electric Vehicle
PID	Proportional-Integral-Derivative
PWM	Pulse Width Modulation
R&D	Research and Development
SLR	Second Level Regulation
SV	Solenoid Valve
USA	United States of America
VDOP	Variable Displacement Oil Pump
VVT	Variable Valve Timing

1. INTRODUCTION

1.1. Framing and Motivation

Although Battery Electric Vehicle (BEV) sales are showing a rapid increase over the last few years, the vast majority of cars on the road still rely on internal combustion engines as a power unit. More specifically, of approximately 1470 million cars on the road worldwide in 2023 [1], only about 26 million of those are electric, and, within these, 18.1 million are BEV (the remaining 7.9 million being Plug-in Hybrid Electric Vehicles, or PHEV). Besides this, 13.8 million of the 26 million total electric vehicles were registered in China, as of 2022 [2] In fact, in 2023, electric vehicle represented only nearly one out of every five new cars sold [3]. Therefore, the internal combustion engine is still very much relevant, representing about 98% of all cars as of today.

Internal combustion engines need lubrication and cooling, so oil pumps are employed in order to maintain appropriate pressure and flow levels of engine oil in important points of lubrication. These pumps are moved (directly or indirectly) by the spinning crankshaft of the engine and can draw up to 2.5 % of engine torque. [4] Traditional fixed displacement oil pumps are over dimensioned for most conditions, ensuring that sufficient lubrication is provided, regardless of conditions. To produce this extra power to run the oil pump and other accessory systems, fuel consumption is increased. In line with both the EU's [5] and the USA's [6] ambitions to become climate-neutral until 2050, regulations concerning CO₂ emissions are becoming ever stricter. As CO₂ is a product of the complete combustion of fuels containing carbon, fuel consumption has to be decreased in order to reduce its emissions [7]. One solution in this direction was the creation of Variable Displacement Oil Pumps (VDOPs). These pumps minimise over dimensioning and allow for less power to be drawn from the engine by adapting their flow output to the needs of the engine under different conditions. VDOPs can either have two stages of regulation or be continuously regulated. Two stage pumps can be managed by mechanical valves, thermal actuators, springs, or ON/OFF solenoid valves, whereas continuously regulated VDOPs are controlled via proportional solenoid valves.

1.2. Objectives

The present work was done within a project of Bontaz, involving a client's request for investigation into a problem which is described further ahead, and as a means to acquire internal knowledge. It concerns the study and optimization of a vane VDOP system equipped with a Bontaz solenoid valve. The solenoid

valve has 3 ports – an inlet, an outlet, and a draining port – and can open in proportion to the voltage supplied to its coil. In this specific project, Bontaz acts as a tier 2 supplier to the OEM, through the pump manufacturer (tier 1 supplier).

The OEM has reported an excessive output pressure oscillation issue with the functioning of the vane VDOP system to the tier 1, who could not eliminate it by changing pump parameters alone, so Bontaz was given the task of looking into the issue.

Having said this, the goals of this project are to:

- Creation of state of art for oil pumps, fostering internal knowledge concerning oil pump systems;
- Equip Bontaz Portugal with the necessary conditions to test VDOP systems with integrated proportional solenoid valve (test fixtures, procedure);
- Analyse tests previously done at Bontaz USA;
- Replicate the customer's problem;
- Utilise quality tools – Root Cause Analysis – and define corrective measures;
- Run tests in order to evaluate possible root causes;
- Analyse tests to find the root cause (if possible);
- Provide data for creation of a simulation model of solenoid valve controlled VDOP systems;
- Serve as a “Lessons Learned” for future problem solving.

1.3. Company presentation

Bontaz started out in 1965 as a small turning workshop on a family farm in Marnaz (Haute-Savoie, France). Initially manned solely by Yves Bontaz, the company manufactured precision-turned parts on a subcontract basis for local businesses, until it established business relationships with French automakers and parts manufacturers. In a few decades, through production of engine and brake subassemblies, establishing itself as a Tier 1 OEM supplier and opening a total of 24 plants in 11 countries, including 4 R&D centres, Bontaz became and solidified itself as the world leader in automotive hydraulic functions, producing 165 million parts per year.

The bulk of the production is based on piston cooling jets and solenoid valves, the latter mainly for oil pump and nozzle control, as well as parking brake and coolant valves.

Nowadays, the company is set on keeping up with the changes in mobility and diversifying into high-potential markets such as electric vehicles, fuel cell engines and light mobility. In line with the company's

aim to accompany the changes in mobility, there is a current focus on BEV product development, such as oil cooling and battery pack ventilation.

Alongside the will to develop new products for zero emissions means of transportation, Bontaz aims to achieve carbon neutrality by 2050 [8].

BONTAZ – BCTP Equipamento Automóvel is an R&D site which opened in 2018, in Lanheses, Viana do Castelo district, and currently employs a workforce of around 100 people. The subsidiary comprises several departments, such as Industrialisation, Mechanics, Production, Injection, and R&D, among others.

In support of the R&D department, BCTP is equipped with prototype assembly lines, containing both manual and electrical presses, a leak testing machine, and hydraulic test benches for oil pumps or solenoid valve/piston cooling jet products and water valves. Alongside these devices, there are basic measurement devices such as callipers, micrometres, comparators, as well as more complex tools - profilometer, vertical measuring columns, tensile and compression testing equipment, 3D coordinate measuring machine, x-ray and tomography scanner, and a profile projector.

2. OIL PUMPS

2.1. Evolution and Components of Lubrication Systems

Internal combustion engines have several moving parts that require lubrication in order to prevent wear or, at worst, friction welding. In older, simpler engines, usually with side valves (or flathead) in which the camshaft is located in the crankcase, sufficient oil could reach these parts through splash lubrication via a spade in the rotating conrod cap (see Figure 1) which would hit the oil in the sump, causing it to splash, hence the name. This, however, might provoke the formation of excessive foam and overheat the oil in engines that work at higher rotational speeds.

As engines evolved to use overhead camshafts and were subjected to greater demands, lubrication systems had to keep up and be able to send the oil upward under pressure, integrating oil pumps, still mixed with splash lubrication.

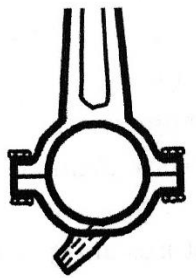


Figure 1: Conrod with spade for splash lubrication. [7]

Nowadays, however, most engines, especially in cars, do not resort to splash lubrication whatsoever, using an oil pump (with a filter) to deliver pressurised oil to all the required points, as can be seen in Figure 2. Having said this, the purpose of oil pumps in an internal combustion engine is to force the circulation of the necessary oil for lubrication and cooling of the engine. Additionally, the film formed by oil around the components helps to improve the seal around the pistons, preventing compression leaks [7].

To avoid potentially damaging pressure peaks, the system is equipped with a safety valve downstream of the pump. Such pressure peaks usually occur mainly when the oil is cold and thus at a higher viscosity level than it would be in steady-state conditions [9].

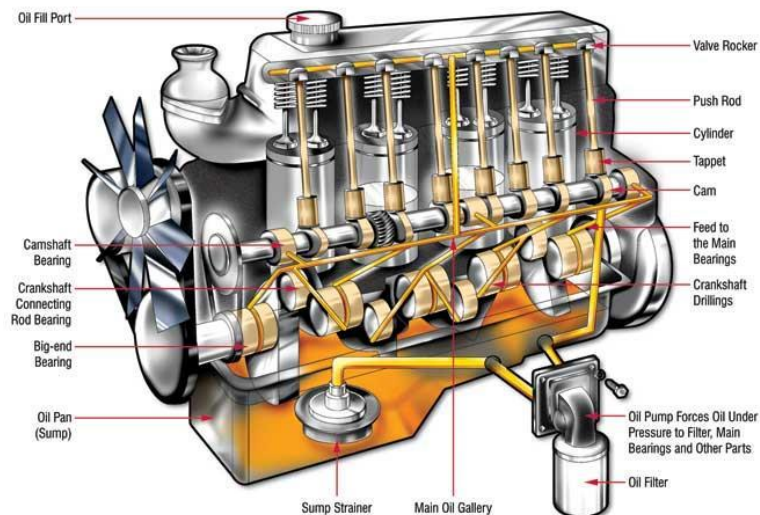


Figure 2: Lubricating oil circuit (without dedicated heat exchanger) [10].

As seen above, the oil stored in the pan, or sump, is sucked by the oil pump, then being forced under pressure through the filter. Downstream of the filter, there is a main gallery, or ramp, from which oil is routed to the different lubrication points, including the crank- and camshaft journals, valvetrain, among others; and other functions, such as regulating camshaft position.

The oil pump can be moved directly through the movement of the crankshaft, or via chain or belt drive, or even with gears, connected to the camshaft. When a transmission system is used instead of direct drive, the transmission ratio can assume different values, depending on project specifications. Apart from these systems, there are also electric motor driven pumps, but they are still not widely used in internal combustion engines [7]. This is because the energy required to drive such systems would have to be supplied by the alternator, which has an associated energy conversion efficiency, making it less efficient than driving the pump directly, while at the same time raising both system complexity and weight. Another reason is the fact that oil supply to the engine is only necessary when the engine is working, meaning that there is no need for an external source of power to the pump during periods when the engine is off. Although most internal combustion engine vehicles use the wet sump system depicted in Figure 2, some vehicles may resort to a different solution, known as dry sump. The idea is to have one pump (called the scavenger pump) for moving the oil that drops from the engine onto the sump to an exterior tank, and another to provide oil from the tank to the engine. These two pumps may be connected together in modules, reducing the overall space taken up by this design, constituting what is known as a tandem pump. A diagram of this system can be seen in Figure 3.

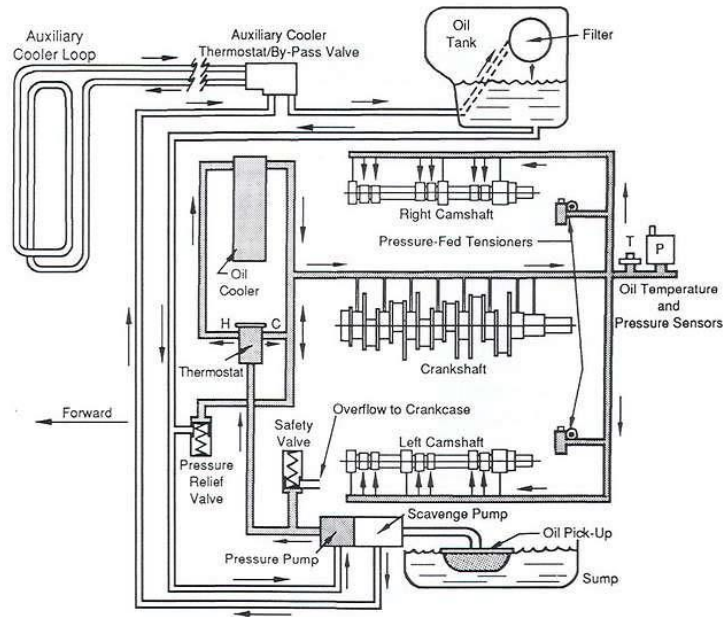


Figure 3: Dry sump lubrication system diagram [11].

The main reasons why dry sumps are used is because certain applications, such as race cars and off-road vehicles, are subjected to high cornering speeds or road slopes steep enough to cause the oil to slosh around in the sump from the centrifugal force or gravity, respectively, leaving the oil pick-up zone without oil, causing the engine to starve. Additionally, dry sump systems allow manufacturers to reduce oil pan height, making the engine lower, thus permitting a lower centre of gravity, and the oil to be relocated elsewhere to accommodate for any weight imbalances [7].

Besides the drive system, pumps can differ in their working principles and geometry.

2.2. Concepts Concerning Oil Pumps

Oil pumps can be categorised using several criteria. Having already analysed the way they are driven, one can separate them according to their working principle (positive displacement or centrifugal), and within those, in terms to their mechanism (reciprocating or rotary), and output (fixed or variable displacement). More distinctions can be made, namely relating to their field of application, design complexity, number of moving parts, etc.

2.2.1. Positive Displacement *versus* Centrifugal Pumps

Oil pumps should be chosen in accordance with the desired pressure and flow values, depending on the system and pump curves. Besides that, it is necessary to take into consideration the working fluid, namely its viscosity.

There are two big groups of pumps: positive displacement (PD), and centrifugal or dynamic pumps. These two types have different means of working and the distinction is evident upon looking at their typical pressure/flow curves, displayed in Figure 4.

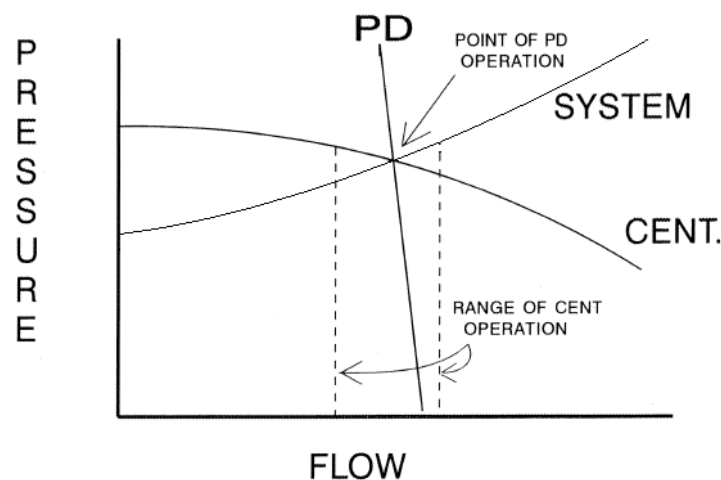


Figure 4: Examples of Positive Displacement (PD) vs Centrifugal pump curves [12]. (adapted)

Centrifugal pumps are also referred to as dynamic pumps because they use centrifugal force to pump fluids to a higher level of pressure. The fluid enters the pump through the centre of the rotating impeller, which imparts velocity to it. Then, it is discharged through a variable cross-section volute, creating pressure. If a system requires a pressurized network with approximately constant pressure at a given flow rate, this kind of pump tends to be the best option.

PD pumps create suction and discharge by displacing a volume of fluid. These pumps suck fluids via an expanding cavity, which creates a pressure drop, sucking in the working fluid to fill the space. Then, the

volume of the cavity where the fluid is contained is decreased through the geometric characteristics of the pump housing and its internal components, increasing pressure, which forces the fluid out. This means that for a given pump speed the discharged volume is approximately constant throughout the pressure range, as can be denoted in Figure 4 by the nearly-vertical PD pump curve. This allows for a high degree of flow control.

Thus, systems which require the same flow throughout a pressure range are best equipped with a PD pump. This happens, for example, in systems where viscosity varies greatly, where pressure will change as well.

A relief valve is mandatory in a PD pump system. If flow is blocked downstream when using a centrifugal pump, the maximum pressure achieved in the system will be the pump's shutoff head (pressure at zero flow). On the contrary, when the pump in question is of the PD type, the peak pressure can be several times the design operating limit.

Most PD pumps are self-priming, depending on the effectiveness of the seals provided by the close tolerances [13], and tend to be preferred when fluid supply is irregular [14].

Another key difference between these pump types is that the PD pumps are practically immune to both changes in fluid properties and in the system, and can operate at high mechanical efficiency over wide ranges of fluid viscosity, while centrifugal pumps' efficiency decreases with increasing fluid viscosity. This is demonstrated in Figure 5 [12], [15].

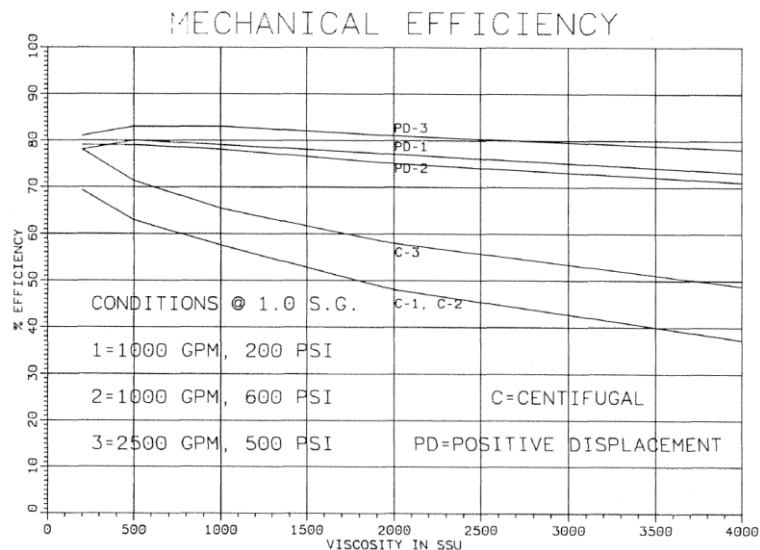


Figure 5: Example of efficiency/viscosity curves for PD and centrifugal pumps [12].

To summarize, the key differences between PD and centrifugal pumps can be observed in Table 1.

Table 1: Key differences between Positive Displacement and Centrifugal (or Dynamic) pumps [15].

	Positive displacement	Centrifugal
Definition	Increases pressure by delivering a fixed fluid volume in a determined space	Increases pressure by increasing fluid velocity through its rotary blades
Types	Reciprocating (piston, plunger, etc.); Rotary (gear, gerotor, vanes, etc.)	Radial; axial
Characteristics	▪ Constant flow ▪ Variable differential pressure ▪ Relatively insensitive to fluid properties or system changes ▪ Self-priming ▪ Not self-limiting	▪ Variable flow ▪ Constant differential pressure ▪ Sensitive to fluid properties and system changes ▪ Not self-priming ▪ Self-limiting

In the case of internal combustion engine oil pumps, for the reasons listed in this section, PD pumps are employed, and for this reason they will be described below.

2.2.2. Reciprocating *versus* Rotary Oil Pumps

PD pumps can be divided into two groups – reciprocating and rotary pumps. Reciprocating pumps can use pistons, plungers, or diaphragms. This style of pump typically requires anti pulsation devices, such as volume bottles, orifices, or pulsation bottles [15]. This effect can also be mitigated via the use of more chambers, timed sequentially to minimise overall pulsation [12]. Nevertheless, these pumps have repeatable and predictable action, meaning that they are ideal for applications which require metering or dosing. Alterations in stroke or length make it possible to regulate the quantity of pumped fluid [14].

When it comes to rotary pumps, they are more commonly used in internal combustion engines, as they allow for fewer moving parts (they do not require inlet and outlet valves) in a smaller, lighter package. Rotary pumps also cause pulsating flow, but it is typically not as severe as that in reciprocating pumps. Within rotary PD pumps, vane, external gear, lobe, internal gear, gerotor, and screw pumps will be described, but there are more examples which are not as relevant to the application in question [7], [9]. Depending on the number of rotating elements, they can be categorised as single- or multiple-rotor pumps.

2.2.3. Fixed *versus* Variable Displacement Oil Pumps

VDOPs emerged from the constant need to improve fuel consumption, since fixed displacement pumps are designed to ensure the supply of flow levels above the engine's needs in all conditions (ensuring proper lubrication in any conditions) [7], as shown in Figure 6. This over dimensioning results in excessive power consumption in off-peak operation, which obviously implies larger fuel consumption and emissions levels.

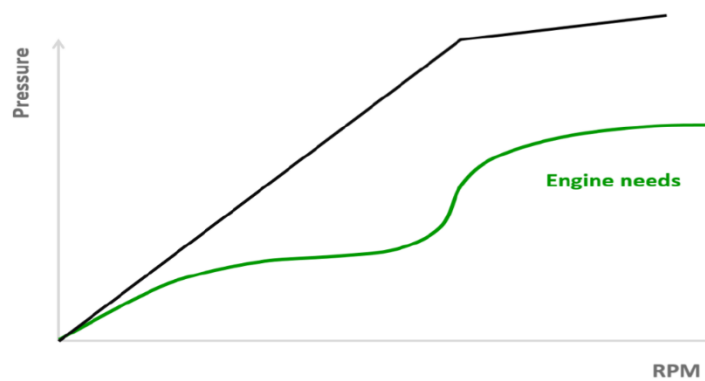


Figure 6: Approximation of engine oil pressure needs (green) vs fixed displacement oil pump output (black) [16].

The graphic representation in Figure 6 makes it clear that the engine's needs are not constant throughout its speed range, meaning that it varies as a function of the revolutions per minute (rpm), the load applied to it, oil temperature, and other factors. These parameters induce the formation of the two *plateau*-like zones visible in Figure 6. These *plateaus* originated the idea for two stage VDOPs, a type of pump regulation that allowed the oil pressure to get closer to the needs of the engine, by having two different operating pressure levels: FLR (First Level Regulation) and SLR (Second Level Regulation) – see Figure 7 - mimicking the *plateaus* in the graphic in Figure 6.

Two-stage VDOPs can be managed in several different ways, usually with on/off solenoid valves (SV), mechanical valves, thermal actuator valves, a double stiffness spring, or two springs.

Nonetheless, these two-stage systems still require significant over dimensioning in certain conditions, so a more elaborate solution was devised to fix the issue, which fell on continuous regulation oil pumps. These are piloted by a proportional solenoid valve, which can adjust to the needs of the engine using a PID controller and being fed a Pulse Width Modulation (PWM) signal. This way, the oil pressure at the exit of the pump may be lower throughout the whole rpm range, because it does not need to be excessive in order to ensure proper lubrication, and electrical power is saved in the control of the SV (via PWM), resulting in decreased fuel consumption and emissions [16].

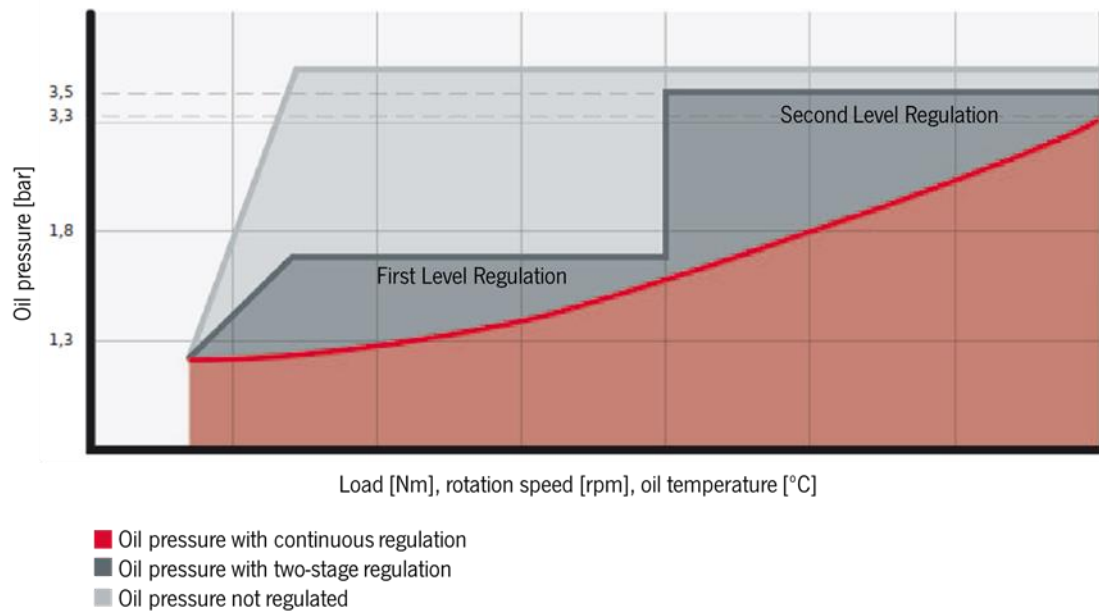


Figure 7: Oil pressure in function of main parameters using different pump regulation systems [17]. (adapted)

It is important to mention that all pumps are equipped with a safety valve, as mentioned in 2.2.1, so that the pressure supplied to the engine is never dangerously over the required limit, however, the pump still consumes excessive power to pump the oil, whose pressure is then limited by the valve. This heats up the oil, causing it to age faster. Hence, variable displacement pumps end up being doubly beneficial, by avoiding both drawing excessive power, and by preventing overheating and the associated early aging of the oil [18].

2.3. Types of Oil Pumps

Having explained the several distinctions between concepts of oil pumps, the following chapter showcases pump designs which are typically used, exploring both reciprocating pumps and, with more detail, rotary types.

2.3.1. Reciprocating Positive Displacement Pumps

Piston and plunger pumps are examples of reciprocating type pumps, and can be radial, axial, or even bent-axis axial, meaning the pistons and their respective rods can be oriented in different directions. A typical horizontal piston pump would work as illustrated in Figure 8.

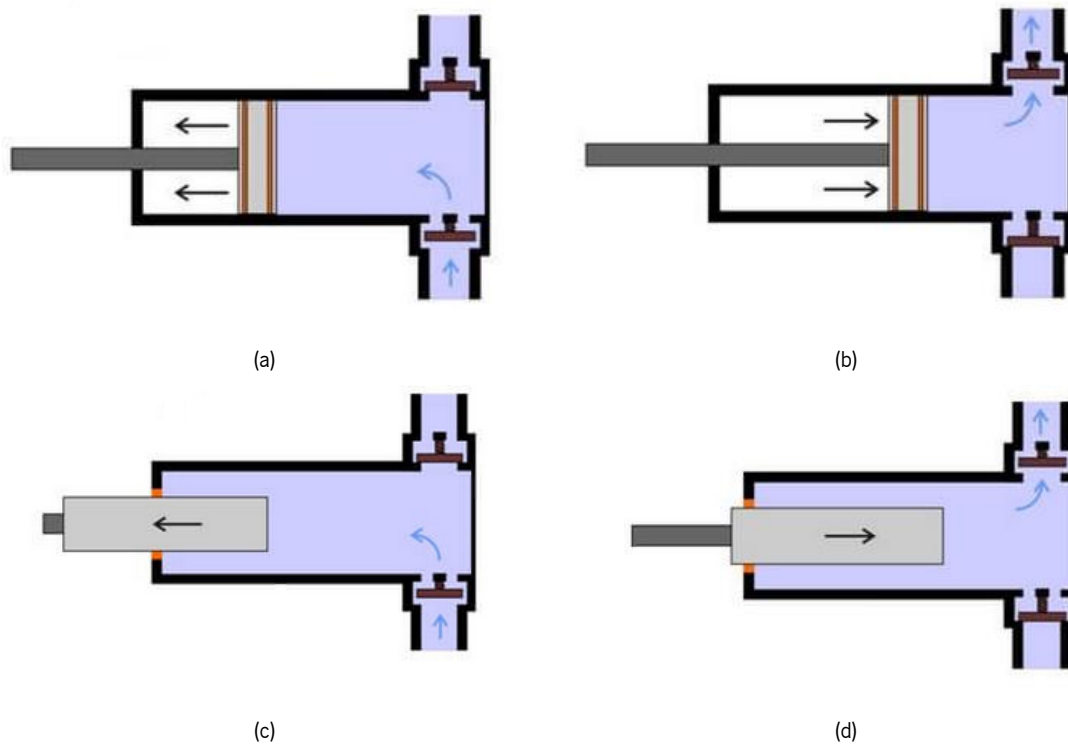


Figure 8: Outline of the operating principle of Piston pumps - (a) and (b) - and plunger pumps - (c) and (d) [14].

In Figure 8 (a) and (c), it can be seen that, as the piston or plunger recedes, the volume in the chamber increases, creating suction, which causes the intake valve to open and let oil enter the pump. Next, in Figure 8 (b) and (d), it is shown that the piston or plunger moves toward the fluid, pressurising it and forcing it outward, through the outtake valve. The same can be done in the opposite side of the piston (where the connecting rod is located), by incorporating more valves, making a double acting pump with a single piston.

The main difference between piston and plunger pumps lies in the fact that, in piston pumps, the sealing ring is located around the outside of the piston, rubbing against the inside wall of the pump housing, whereas in plunger pumps, the sealing ring is inserted on the pump housing wall, rubbing against the plunger's outside wall.

Plunger pumps are able to achieve higher pressures than piston ones. Because the seal is placed at the entrance to the chamber, the plunger fits loosely inside the cylinder, and moves without touching the wall [19]. This design allows for decreased mass and size, which means the speed of plunger pumps can also be higher [20]. The difference in seal locations means that plunger pumps can produce higher pressures [21].

These kinds of pumps can reach up to 95 % volumetric and mechanical efficiencies [22]. Radial and duplex axial pumps are exemplified in Figure 9.

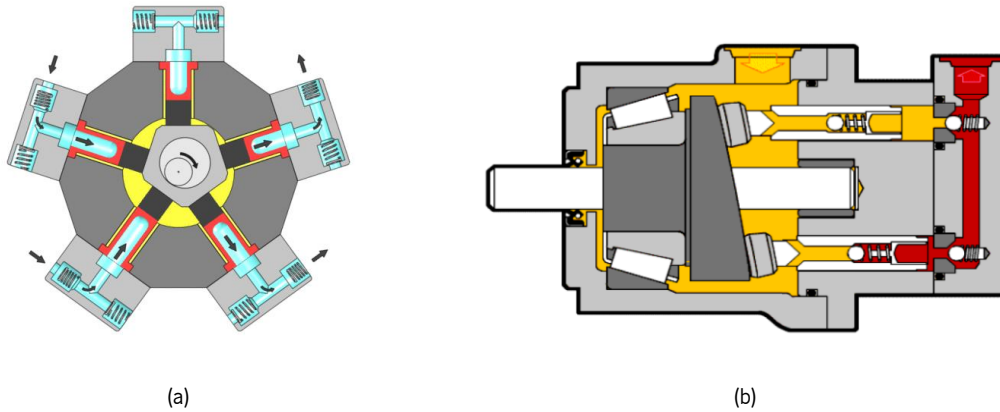


Figure 9: (a) Radial piston pump [23]. (b) Axial duplex pump [24].

When it comes to diaphragm pumps, these are also reciprocating pumps, but instead of a piston, or plunger, they use one or two membranes (in the case of double diaphragm pumps) and valves to control intake and outtake. Movement of the membrane can be piloted mechanically, via solenoid directly, or through a solenoid valve (typically, using compressed air).

To better explain its functioning, an example of a plunger-operated (mechanical) diaphragm pump is shown in Figure 10.

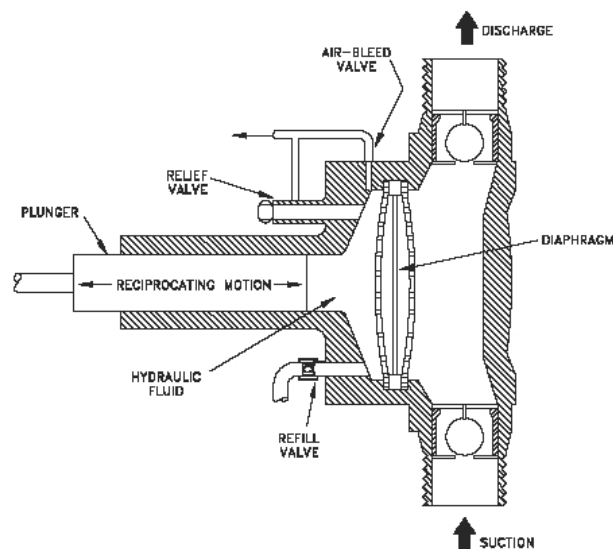


Figure 10: Plunger-driven diaphragm pump [25].

When the plunger recedes, the pressure inside the left side chamber (drive chamber) decreases, sucking in hydraulic fluid. When the plunger advances, an increase in pressure in the compression chamber happens, which will force the pumped fluid out through the discharge valve.

These pumps offer little friction, they can be sensitive to low pressures, and are self-priming [25]. Also, they can pump a variety of fluids, even containing suspended solids, depending on the materials used. However, as they are reciprocating pumps, they require frequent maintenance [26].

2.3.2. External Gear Pumps

External gear pumps contain two gears in mesh inside a case, being a type of multiple-rotor pump. One of the gears, which is driven by the input shaft, moves the other, but each gear has its own shaft. This design is widespread in internal combustion engines, because of the reduced number of components and, consequently, its affordability and reliability. A diagram showing the basic functioning of these pumps is presented in Figure 11.

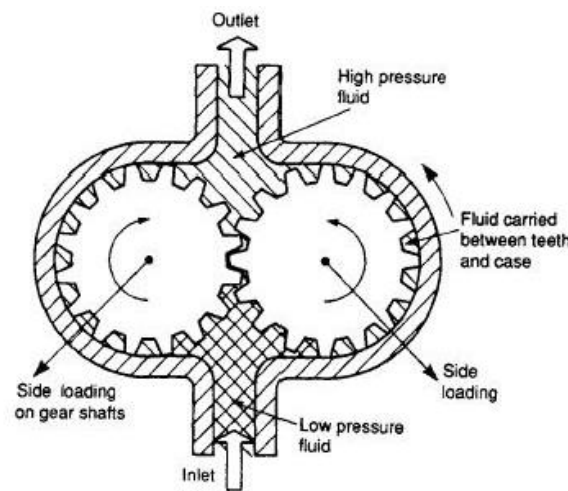


Figure 11: External gear pump diagram [27].

As illustrated, oil is pressurized by the movement of the gears, which drag it along the walls of the pump case and out through the other side, a process that can be reversed simply by changing the direction of rotation. The meshing of the gear teeth impedes the oil from leaking back onto the suction side. Nevertheless, they create a nearly constant loss through running clearances between gear faces, tooth crests, and the housing, meaning that running low speeds and flows may be inefficient and that these pumps should be operating close to their maximum speeds and outputs [28].

Different gear types can be used in this kind of pumps and their design is not normalised, being the individual choice of each manufacturer, according to their own needs [29].

In summary, external gear pumps are compact, well-suited for high viscosity fluids, and easily manufactured in different materials to match fluid properties. [26] They should not be used when dry, having to be primed in order to ensure lubrication to the pump's own moving parts. This also means that abrasive fluids or entrained solids might damage the pump [14].

2.3.2.1. External Gear VDOP

In the case of the external gear pumps, displacement variability is provided by moving the shaft of the driven gear (not powered by the engine), thus reducing the pumped volume, which varies proportionally according to the length of the meshing area of the gears [30]. An example can be seen in Figure 12.

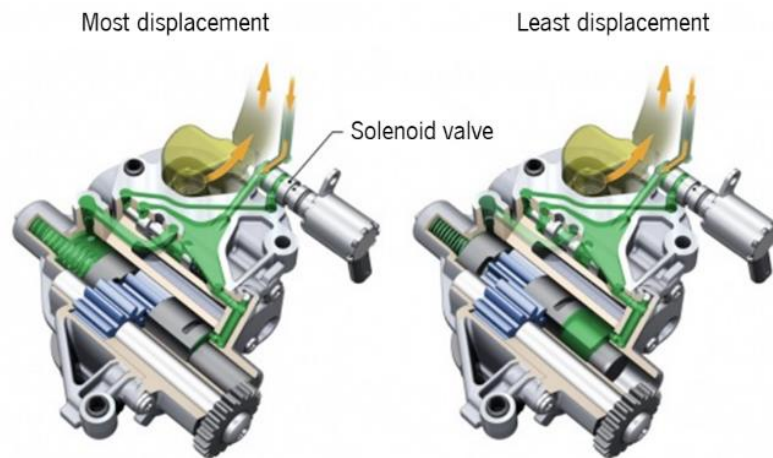


Figure 12: External gear VDOP [30].

2.3.3. Lobe Pumps

Lobe pumps are a type of multiple-rotor rotary pump whose operation is quite similar to that of external gear pumps and can also be reversed simply by changing the direction of rotation. In comparison, one big advantage with this design is that the lobes do not come in contact with each other, be it a two, three or more lobe pump, avoiding contamination and fluid shear, as well as reducing wear on the pump. A diagram of this type of pump is represented in Figure 13.

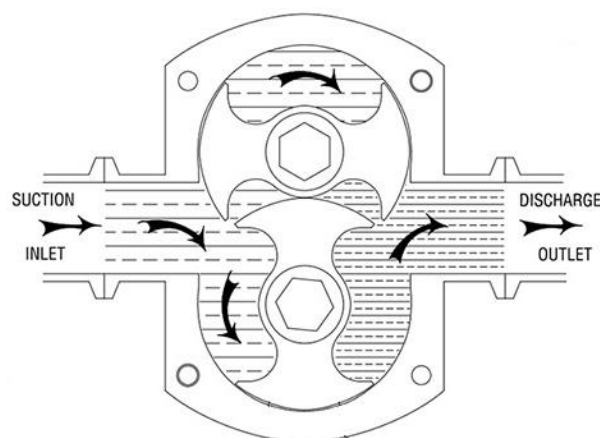


Figure 13: Lobe pump diagram [31]. (adapted)

These need tight dimensional and geometric tolerances, but not as strict as those with the rotating elements in contact. Rotation timing between the shafts is maintained by gears in the gearbox of the pump, where the support bearings are also located. For this reason, pressure is limited by the bearing location and shaft deflection.

Because the lobes do not touch, fluid contamination is kept to a minimum [32].

2.3.4. Screw Pumps

They move the fluid via a rotating, helical threaded shaft (screw), causing it to move axially. For a diagram of a screw pump, see Figure 14.

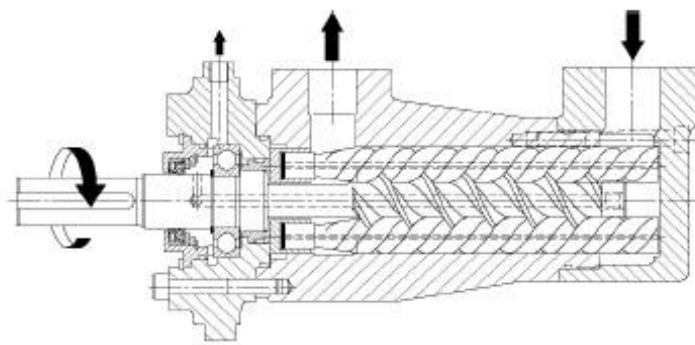


Figure 14: Single screw pump diagram [33].

This kind of pump is typically used for hydraulic lifts and submarines, as well as cutting oils and lubricants, thanks to its low noise and reduced pulsation [33].

2.3.5. Internal Gear Pumps

In this kind of pump, the oil is also pressurised due to the rotation of two gears, but in a different layout, with one inside of the other. The outside gear has internal teeth to mesh with the inside gear's external teeth. Either gear can be the drive gear or the driven gear, depending on the pump. One example is shown in Figure 15.

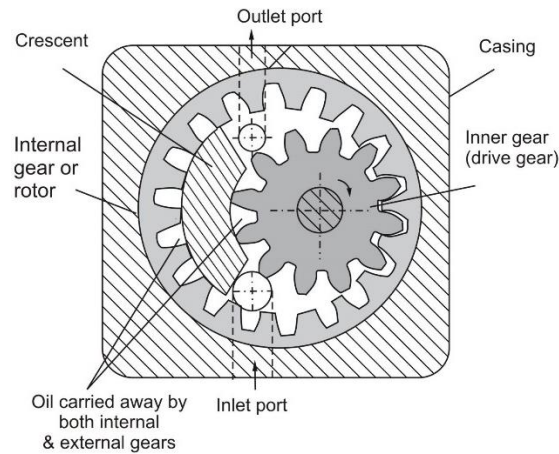


Figure 15: Internal gear pump scheme [34].

Space between the gears increases from the meshing zone towards the direction of rotation, creating a pressure drop which sucks in the oil from the inlet port. This oil is then conducted through either side of a crescent shaped filler piece separating the low- and high-pressure zones. Tolerances in this area must be tight to keep both zones sealed. After passing by the crescent, the oil is pressurised using the opposite effect that initially provided suction, that is, by diminishing the space between gears, forcing the oil out through the outlet port.

2.3.6. Gerotor Pumps

Gerotor pumps have a working principle analogous to the one described above, comprising an internal and an external gear in constant contact, with the external gear containing one more tooth than the internal gear – also called gerotor - providing displacement of volume between them. Within vehicle applications, besides internal combustion engines, they can also be placed in some automatic gearboxes, or in hydraulic locking differentials and power steering systems [35].

However, instead of a crescent sealing and separating the high- and low-pressure zones, the contact of the teeth of the internal rotor with the external gear is what guarantees the seal. To better understand the concept, a scheme of a gerotor pump is exposed in Figure 16.

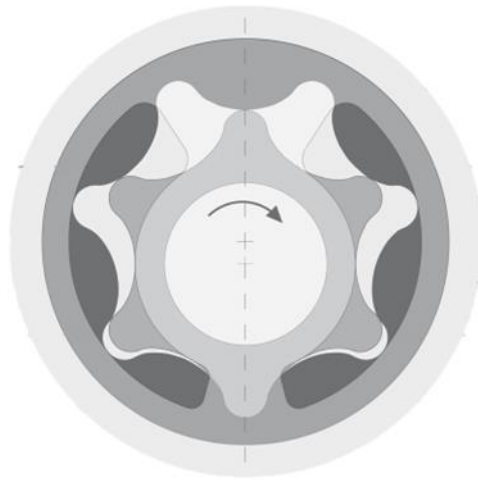


Figure 16: Gerotor pump scheme [36].

Typically, the teeth of the internal gear have an epitrochoidal, hypotrochoidal, or cycloidal profile, among others, and the outer gear is made to match with circular arc profiles [37]. This way, the space that was otherwise occupied by the crescent (in an internal gear pump) can now be filled with oil, meaning that, when comparing a gerotor pump with an internal gear pump with the same teeth width, outer diameter of the internal gear and root diameter of the external gear, the volume displaced per rotation is bigger in the case of the gerotor pump [38].

2.3.6.1. Gerotor VDOP

Gerotor VDOPs work by moving the minimum area zone shown in Figure 16. In these VDOPs, the outer gear sits inside a stator, which is held in place by a spring. An example of a gerotor VDOP is exhibited in Figure 17.

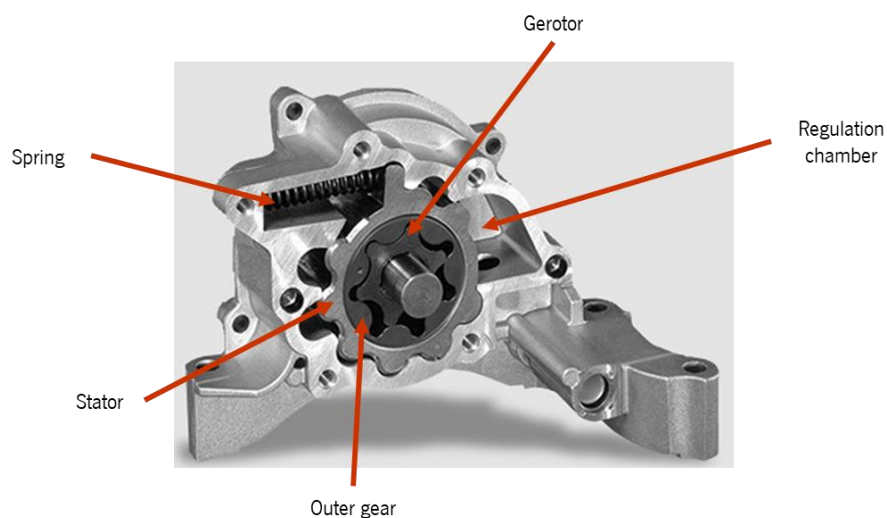


Figure 17: Gerotor VDOP [39].

To reduce displacement, oil pressure is fed to a regulation chamber outside of the stator, pushing it. When the force applied by oil on the stator overcomes the force of the spring, the stator rotates and in turn, the eccentric axis of the outer gear rotates about the central axis of the gerotor, changing the position of the minimum area zone relative to the inlet and outlet ports of the pump. This way, the suction and discharge zones's alignment with their respective ports is altered, reducing flow [40].

2.3.7. Vane Pumps

Vane pumps are a type of single-rotor pump, composed of an external case containing an outer ring, an eccentric shaft, and two or more sliding vanes, inserted in cavities in the eccentric shaft. An outline of this kind of pump is illustrated in Figure 18.

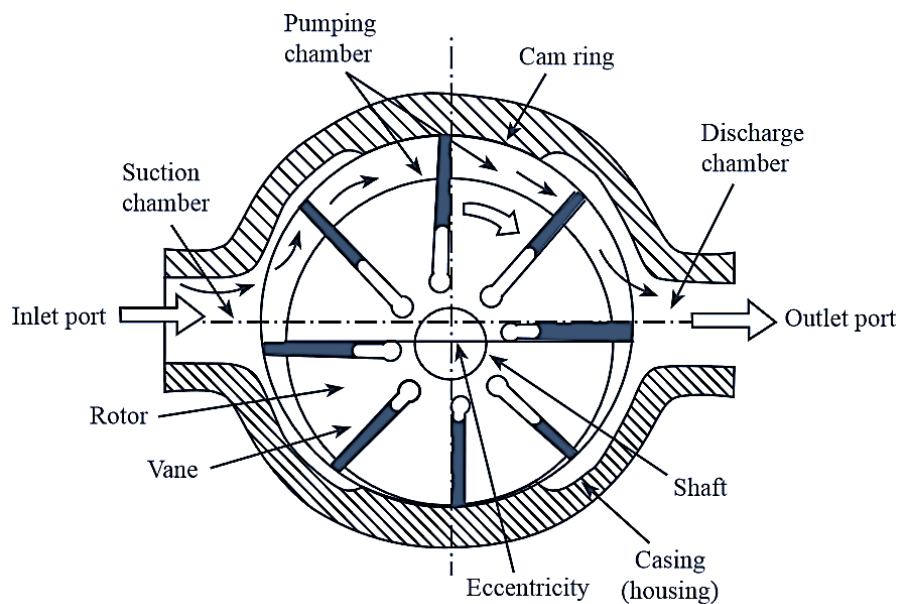


Figure 18: Vane pump diagram [41].

As the eccentric shaft rotates, the vanes slide across it, due to the contact between the tip of the vanes themselves with the outer ring. To guarantee the vanes are always in contact with the outer ring, the system might rely on centrifugal force caused by the rotation, on a centring ring, on springs, or on pressurised oil taken from the discharge port to the inner base of the vanes, pushing them outward [29]. To avoid damaging the inner surface of the ring (usually made of steel), the vanes are normally made from materials with a lower hardness, varying according to their application. Rotor materials can also vary for the same reasons [42], [43].

Because the rotor is eccentric, the distance between a fixed point on its surface and internal perimeter of the outer ring varies as it rotates. Hence, the vanes create several cavities between them, whose volume

is shorter near the point where the eccentric's axis of rotation is closer to the outer ring (left side in Figure 18), and larger on the opposite side (right side in Figure 18).

The oil enters the pump through the inlet hole, being sucked by the pressure drop caused by the increase of the cavities' volumes. Then, it is moved around the pump by the rotating vanes and, at the exit port, the opposite effect occurs: the cavities decrease in volume, pressurizing the oil, which is forced out of the pump. This process can be reversed, changing the direction of pumping [43]. The flow occurring at the exit of vane pumps should be mostly low pulsation flow, but pulsation might be a problem for pumps with a low number of vanes (one or two) at low rotation speeds [43].

There are several design variations for vane pumps, with an important variant being pendulum pumps. The design is very similar to regular vane pumps, except that instead of sliding vanes, the pump contains pendulum-shaped sliders supported on an outer rotor, sliding in grooves located in the internal rotor. One example can be seen in Figure 19.

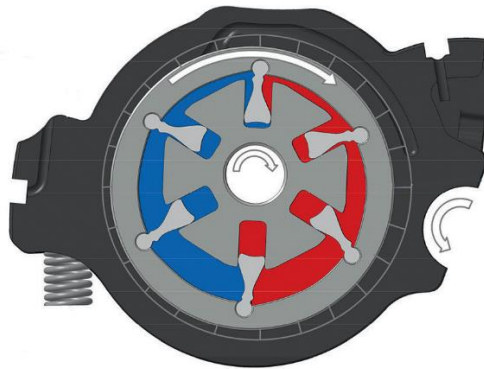


Figure 19: Sliding pendulum type vane pump [44]. (adapted)

2.3.7.1. Vane VDOP

When it comes to vane pumps, to regulate the displacement, the eccentricity of the rotor might be reduced through rotation of the external ring around a fixed pivot point. This is the case of the example in Figure 20.

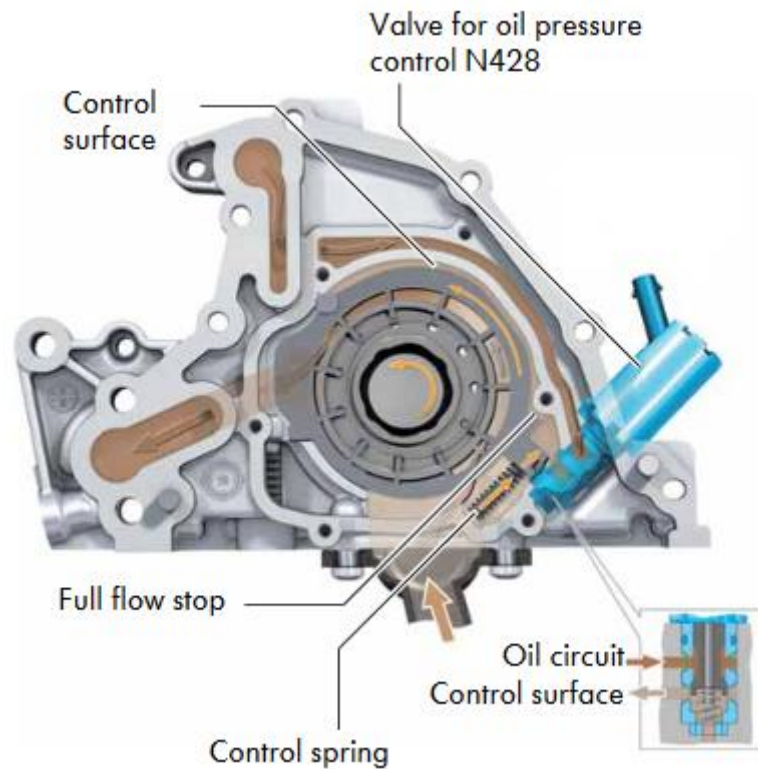


Figure 20: Unregulated vane VDOP (solenoid valve not activated) [45].

Between the outer ring and the pump casing is a volume called the regulation chamber. As shown in Figure 20, a spring keeps the external ring in its unregulated position until its force is exceeded by the pressure in the chamber. When the force exerted by the pressure on the outer ring is greater than the opposing spring force, it starts to move, compressing the spring, as seen in Figure 21. This adjusts the pump so that the outer ring's centre travels closer to the inner ring's centre. In doing so, the difference in volume between the chambers formed by the rotating vanes becomes smaller, thus creating less suction and less pressure on exit.

Enhancement of proportional solenoid valve to manage variable displacement oil pump

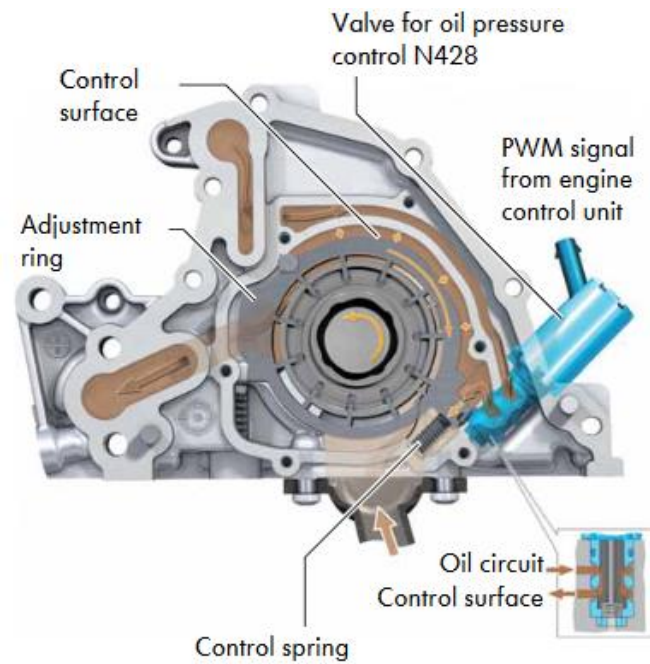


Figure 21: Regulated vane VDOP (solenoid valve activated) [45].

The necessary pressure to regulate the pump might be approximately constant or increase as the outer ring rotates [16].

Alternatively, vane VDOPs can use a translational mechanism, where the displacement is adjusted through the translation of the outer ring, rather than its rotation, but following the same principles. (Figure 22).



Figure 22: Vane translation VDOP model [46].

3. SOLENOID VALVES

Solenoid valves (SV), often abbreviated as just “solenoids”, are a type of electromechanical valve with a wide array of possible functions of fluid (either gas or liquid) control, of which on/off and flow regulating valves are examples.

Since their first appearance, perhaps the greatest turning point towards solenoid valve sustainability in the market was the introduction of plastic overmouldings in the 1950s, making them more efficient, reliable, and resistant to corrosion and other chemical interactions [47].

Ever since, these valves have proved their usefulness throughout the several different industries, as they control pneumatic and hydraulic circuits which can be used in operating tools, from handheld to large industrial equipment; HVAC systems; domestic applications, such as automated garden watering lines; and automotive systems, among others. Besides this, SV can be used as part of a control circuit to pilot other, larger valves, in a function analogous to that of a relay.

Nowadays, solenoid valve applications are proliferating as new mechanisms are developed, and the introduction of new composites and high performance polymers such as PEEK provide a large diversity of possible improvements.

3.1. Automotive Solenoid Valve Applications

Focusing on the automotive industry, solenoid valves see a significant range of applications. These applications are spread out across various systems in a car, operating with different working fluids. For example, in a car, there can be solenoid valves controlling the heating and cooling systems - flow of HVAC refrigerant, coolant for cabin heating and cooling systems – emissions reduction systems - Exhaust Gas Recirculation (EGR), Adblue thermal management – braking systems – compression brakes and exhaust brake – and even Variable Valve Timing systems (VVT), among others. However, as the focus of this work is on an oil regulation SV, two important examples of oil valve function provided by Bontaz are explained in more detail - piston cooling jet and VDOP management valves.

3.1.1. Piston Cooling Jet (PCJ) Control

The piston cooling jets (example in Figure 23) are a fundamental part when it comes to cooling and lubricating internal combustion engines and play a very significant role in oil consumption, as they can direct around 40 % of the pumped oil flow [48] towards the pistons (see Figure 24) in order to lower their temperature and ensure the proper running conditions are maintained.

Enhancement of proportional solenoid valve to manage variable displacement oil pump



Figure 23: Piston cooling jet [49].

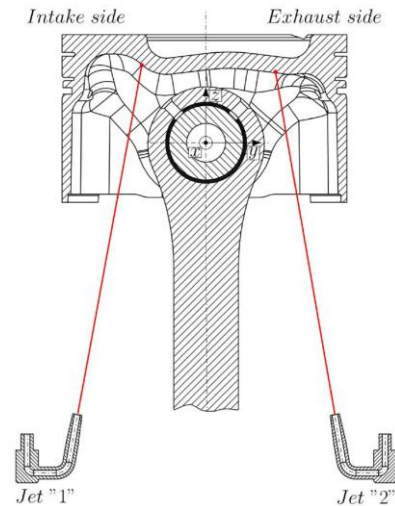


Figure 24: Diagram of piston cooling jet oil spray towards the base of a piston [50].

If left fully open, the added cooling will delay achieving the steady state conditions upon cold start, so the oil flow should have regulation. In addition to this, by continuously managing oil flow to the PCJ, piston temperature can be optimized for efficiency and performance, and lubrication systems can be downsized, reducing power consumption, and thus reducing emissions. An example of a solenoid valve directly integrated in the lubrication system to control PCJ oil flow is shown in Figure 25:

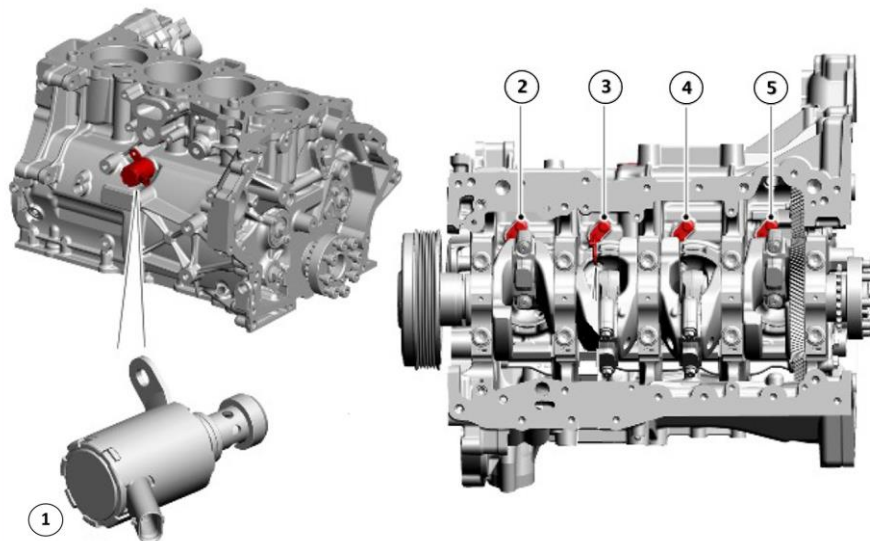


Figure 25: Integrated solenoid valve for PCJ oil flow control [51]. 1 – Solenoid valve for PCJ control. 2 to 5 – PCJs.

3.1.2. VDOP Management

As explained in 2.2.3, VDOPs can be managed by SV, with this system being at the forefront of emissions reductions when it comes to oil pumps. For this, manufacturers such as Bontaz provide solutions in the

form of proportional solenoid valves, which can continuously adapt to the supplied oil pressure from the engine, regulating the pump as necessary.

The system has already been shown in Figure 20 and Figure 21 respectively, where a vane VDOP is shown unregulated while the SV is off, limiting the pressure of the oil arriving to the pump's regulation chamber; and regulated (SV on), allowing oil under higher pressure to arrive at the regulation chamber, which rotates the outer ring and reduces pressure changes within cavities.

3.2. Working Principle

Since the component under study is a solenoid valve (SV), which works via an electromagnetic mechanism, the theoretical foundations behind the operation of this mechanism should be explained.

3.2.1. Definition of Magnetic Field

Firstly, the concept of magnetic field must be introduced. A magnetic field is a space in which a magnetic force is exerted. It happens around a magnetised material or a conductor carrying electrical current. Its location is indicated by the presence of magnetic lines of force (simply called flux lines). Magnetic fields enter and exit materials through zones denominated as poles, which come in pairs, with flux lines being oriented from the North to the South poles [52]. The exemplifying scheme in Figure 26 shows the flux lines around a typical bar magnet.

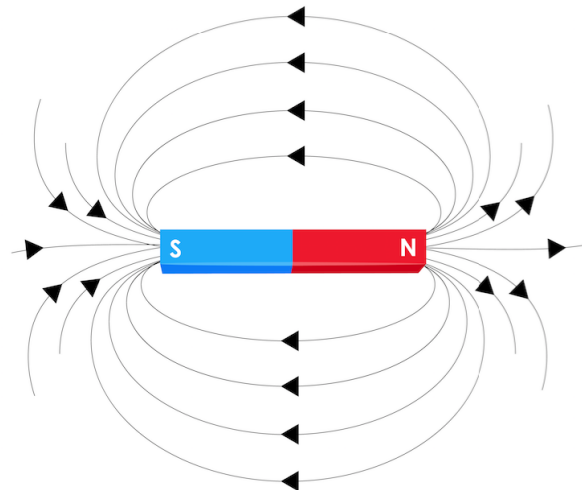


Figure 26: Magnetic flux lines around typical bar magnet [53].

The magnetic field is represented by $\vec{B}$.

3.2.1.1. Solenoid

A solenoid is a coil of conductive wire. For a solenoid to have a magnetic field, a current has to be applied to it. Once charged, the magnetic field, $\vec{B}$, of the solenoid will behave as shown in Figure 27:

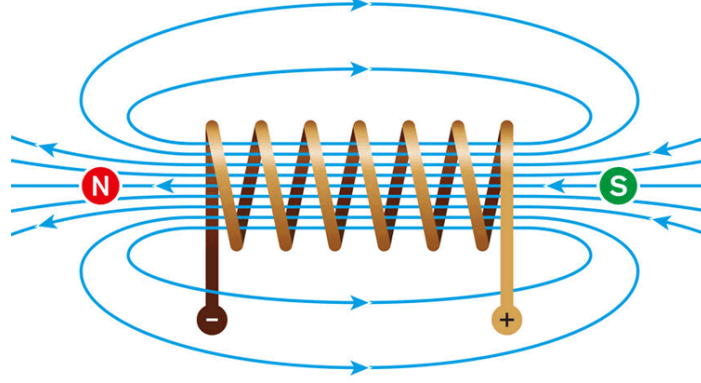


Figure 27: Magnetic field around a solenoid [54].

The strength of the exerted magnetic field is represented by $\vec{H}$, in Ampere/metre which is given by equation 1.

$$\vec{H} = \frac{N \cdot I}{l} \quad 1$$

Where N is the number of coils, I is the current flowing through the solenoid and l is the length of the wire.

From here, the magnetic flux density, B , expressed in Tesla, can be determined, through equation 2:

$$\vec{B} = \mu_0 * \vec{H} \quad 2$$

Where μ_0 is the permeability of the vacuum, in Newton/Ampere² since there is no material around or inside the coil.

3.2.1.2. Electromagnet

Solenoid valves work thanks to the use of electromagnets, where a ferromagnetic metal core is placed in the middle of a solenoid, as exemplified in Figure 28.

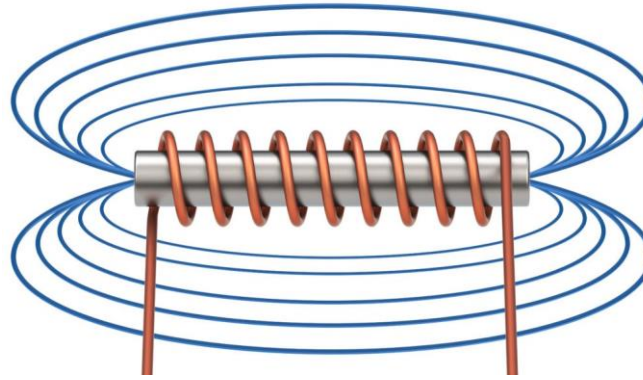


Figure 28: Diagram of an electromagnet [55].

Because there is now a metal core, whose permeability is much higher than that of the vacuum, the magnetic flux density created by the electromagnet is given by replacing μ_0 in equation 2 with the much higher permeability of the core material, μ . In the end, the magnetic field of the electromagnet is much stronger than that of just a solenoid alone [47].

3.2.2. Ferromagnetic Materials

As previously explained, the materials used for solenoid valves' magnetic cores are ferromagnetic. To understand magnetic materials' properties, it is necessary to introduce the concept of magnetic moment, which is the ratio between the maximum torque exerted on a magnet, current-carrying coil or moving charge situated in its magnetic field and the strength of that field.

This property is tied to electron pairing – every electron produces a magnetic field; however, paired electrons have opposite spin and orbits, nullifying each others' magnetic moment. Therefore, atoms with all electrons paired have zero net magnetic moment.

The magnetic properties of materials change the way they react to and affect magnetic fields around them, as shown in Figure 29.

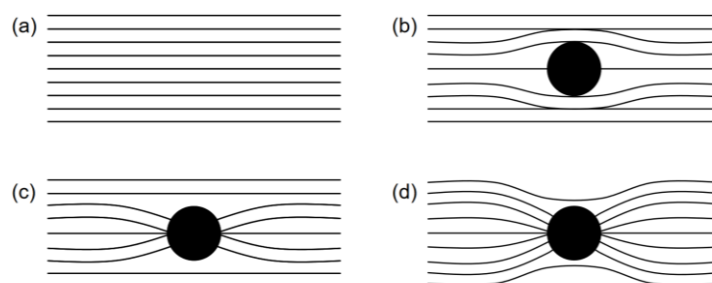


Figure 29: Effect of magnetic materials on a uniform magnetic field (a). Diamagnetic (b), Paramagnetic (c), and Ferromagnetic (d) materials behave differently [56].

3.2.2.1. Diamagnetic materials

In general, materials with all their electrons paired are diamagnetic, although there are exceptions. When subjected to an external magnetic field, they produce a weak opposing magnetic field themselves, repelling the source of the external field (see Figure 29 (b)). If the external magnetic field within which the diamagnetic material is located is strong, then the strength of the field in the air surrounding it will be stronger than the material's own internal magnetic field. Almost every element in the periodic table is diamagnetic, including copper, silver, and gold.

3.2.2.2. Paramagnetic materials

Paramagnetic materials are slightly attracted by an external magnetic field (see Figure 29 (c)) and do not retain properties when it is removed. This behaviour is a consequence of unpaired electrons aligning their paths under the influence of the external field. Some examples of these materials are aluminium, magnesium, molybdenum, and lithium.

3.2.2.3. Ferromagnetic materials

Ferromagnetic materials, such as iron, cobalt, or nickel, are strongly attracted to magnetic fields (see Figure 29 (d)) and retain their magnetic properties if the external field is removed. Their atoms have a net magnetic moment different from zero since they have some unpaired electrons.

They are composed of magnetic domains within which atoms' magnetic moments are aligned parallel to each other, all the while being different in distinct domains (see Figure 30 (a)) [57]. In an unmagnetised part, these regions' magnetic moments point in distinct directions. Thus, the magnetic force of each domain is relatively strong, but the total for the (unmagnetised) material is zero.

When subjected to an external magnetic field, the domains' magnetic moments align (see Figure 30 (b)), and the material emanates its own field. This illustrates how a ferromagnetic material piece can become a magnet. The domains can also be aligned by passing an electric current through the material. When every domain is aligned, the material is said to be magnetically saturated.

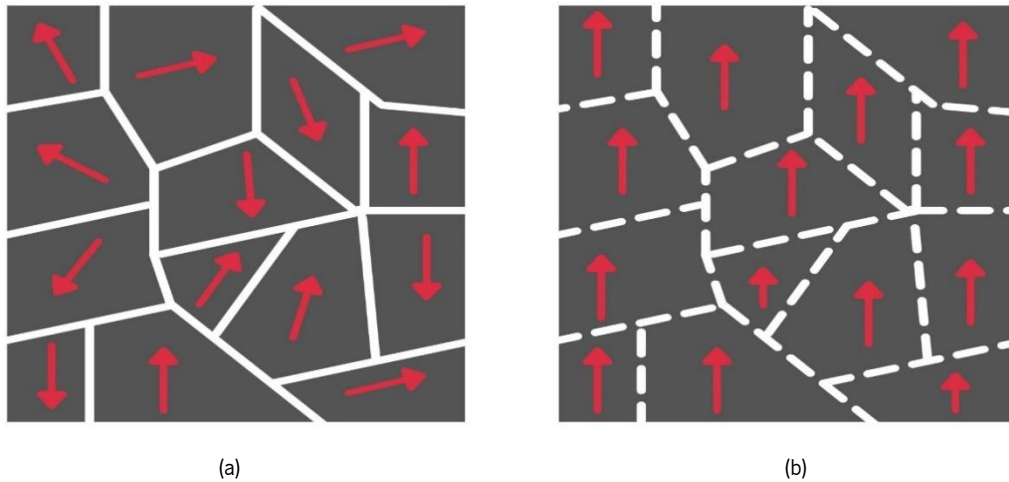


Figure 30: Magnetic domains in an unmagnetised ferromagnetic material (a) and in a ferromagnetic material subjected to an external magnetic field (b) [58].

3.3. Construction and Functioning

An assembled SV usually consists of two subassemblies – an electrical and a mechanical one. The electrical subassembly comprises an electromagnet with both a fixed and a mobile core, in which the mobile core, as it is attracted by the fixed core, moves, actuating the desired mechanical system. When power ceases to be supplied to the electromagnet, usually a spring ensures that the mobile core returns to the initial position.

Typical components of the electrical subassembly of an SV are listed below, and identified in Figure 35:

- Shell;
- Top magnetic washer;
- Connector;
- Overmoulding;
- Spring;
- Coil;
- Fixed core;
- Cartridge;
- Mobile core;
- Needle;
- Plate.

As shown through Figure 31, the shell, top-side magnetic washer, mobile core, fixed core, and the plate are used to encapsulate the magnetic field (when the SV is activated), serving as a solid means of

propagation for the magnetic field lines, which also bridge the air gaps between some of the components. The fixed and mobile cores are in opposite poles of the magnetic field, and, therefore, are subjected to magnetic forces, attracting them to one another. These forces move the mobile core closer to the fixed core, closing the air gap between them, and actuating the valve.

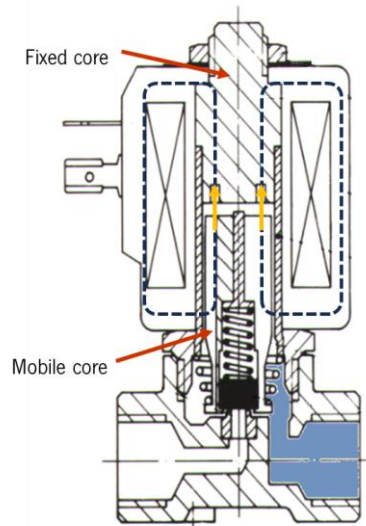


Figure 31: Magnetic circuit representation on a cross section of the electronic subassembly of a SV [59] (adapted). Dotted blue lines represent the magnetic circuit. Yellow arrows represent direction of motion of the mobile core.

On the other hand, the components in the mechanic subassembly of an SV can vary widely depending on the application, but usually consist of:

- Valve body;
- Piston;
- Spring;
- Stopper.

The body is responsible for guiding the fluid from and to the desired canals, with the piston regulating flow (and, necessarily, pressure). The spring acts to return the piston, needle, and mobile core to their original position when the valve is not activated and offers resistance to their advance when activated. The stopper is a metal plate whose function is to hold the spring and allow SLR calibration (see 3.3.3.2), according to the required specification.

To illustrate an example of how a solenoid valve could function, an SV cross section diagram can be seen in Figure 32.

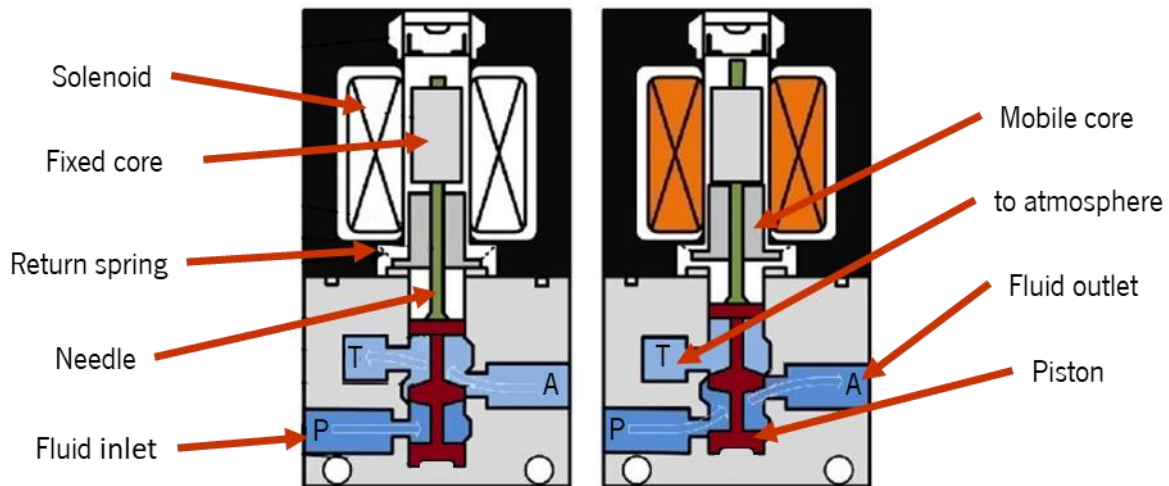


Figure 32: Cross section diagram of a 3/2 on/off solenoid valve. Left – SV off; Right – SV on [60]. (adapted)

While the solenoid valve is off (left side), the mobile core is forced downward by the return spring, and in turn holds the needle and piston (green) down. In this state, the SV is closed, so there is no passage from port P to port A. However, there is passage from port A to port T. In this case, imagining that port A leads to a reservoir and knowing that port T leads to atmospheric pressure, flow from port A to T could be called “draining” flow.

On the right side, where the solenoid is on, the mobile core is pulled towards the fixed core, which is acting as an electromagnet, beating the force of the return spring, and pulling the mobile core upward. As the piston moves with the core, passage from port P to A is opened and flow from port A to T is stopped. The solenoid valve is, therefore, open when energized. Following the previous line of thought, if fluid flows from the inlet, P, to the reservoir, downstream of port A, this influx can be described as “filling” flow, as it fills the reservoir.

Apart from the design shown in Figure 32, there are many different configurations, depending on the valve’s function and the environment in which it operates. Other geometries will be explained further below.

3.3.1. Normally Open *versus* Normally Closed

In the example represented in Figure 32, it is shown that the SV is closed when the solenoid valve is not energized, which means that it is a normally closed valve. In cases where the solenoid valve stays open when not energized and closes when activated, it is said that the SV is normally open.

3.3.2. Nomenclature and Schematic Representations

In a pneumatic or hydraulic scheme, valves are distinguished by:

- Number of ports and positions/states they can be in:

For a valve with x ports and y positions, the designation is x/y valve.

- Normally open or normally closed:

As previously explained, a valve whose resting (deactivated) state is open is said to be normally open, and the opposite for normally closed.

- Activation and Return Mechanism:

Valves can be activated and returned in several different ways. For the case in study, the relevant types are solenoid activated, spring returned valves. Besides this, however, they can be pneumatically/hydraulically, and manually activated and returned.

3.3.2.1. On/off Solenoid Valves Scheme

Considering the on/off valve pictured in Figure 32, because it has 3 ports and 2 possible positions, it is called 3/2 solenoid valve. Its representation, obtained using the Fluidsim software, is as depicted in Figure 33.

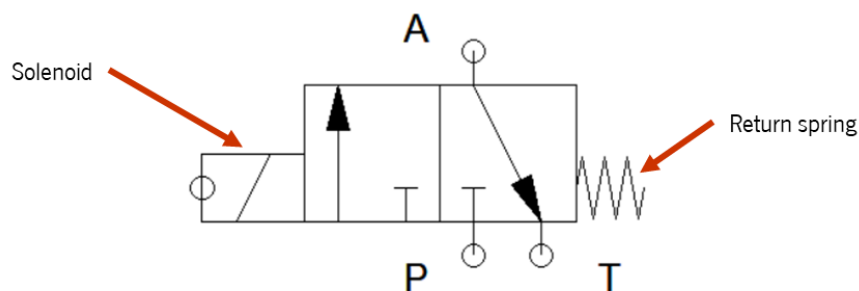


Figure 33: 3/2 on/off SV hydraulic scheme.

The valve is shown in its resting position, with the spring forcing it to stay closed. When the solenoid receives current, it moves the valve scheme to the right, compressing the spring.

3.3.2.2. Proportional Solenoid Valves Scheme

Considering the same example, but assuming that it has a proportional behaviour as current is gradually applied, the valve is still a 3/2 SV, but there is now a middle ground between the two positions. Such a behaviour is represented through a third position, as seen in Figure 34.

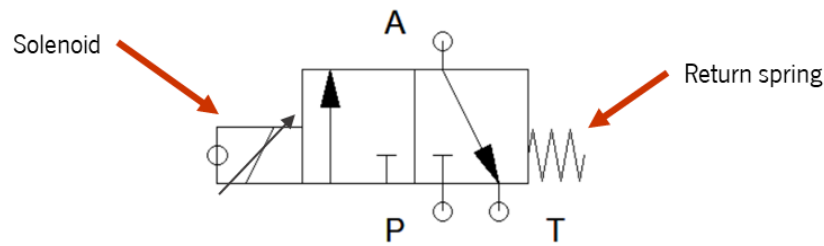


Figure 34: Proportional 3/2 SV hydraulic scheme.

The middle position represents the state of the SV when it is getting between 0 % and 100 % current, with all the ports being in communication. When full current is achieved, the SV is at the leftmost position.

3.3.3. On/off *versus* Proportional

As was explained before, 2-stage VDOPs can be managed by an on/off SV, whereas continuously variable VDOPs are controlled by proportional SV. On/off and proportional valves are not just different in their hydraulic scheme, but also when it comes to their geometry. Figure 35 presents an example of a Bontaz on/off SV.

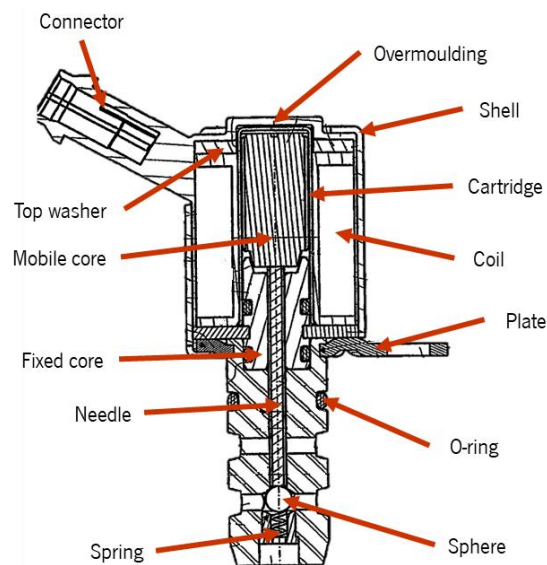


Figure 35: Cross section view of a 3/2 on/off solenoid valve [61]. (adapted)

From an electrical subassembly point of view, this design is analogous to the one presented in Figure 32. The mechanical subassembly, however, differs. The spring separates the mobile core from the fixed core when the SV is off. Instead of a piston, the needle holds a sphere in place, blocking flow between ports P and A. When the solenoid is energized, the mobile core is pulled upward by the magnetic force, compressing the spring. The pressure coming from port P pushes the sphere and needle upward, blocking flow from A to T (and allowing flow from P to A).

It is shown in this design that there is no possibility of only partially opening flow from P to A. It is either fully open or fully closed.

On the other hand, proportional valves are used when there is the need to create a pressure drop from P to A port, but without fully closing flow. This is made possible by a mechanism such as the one demonstrated Figure 36.

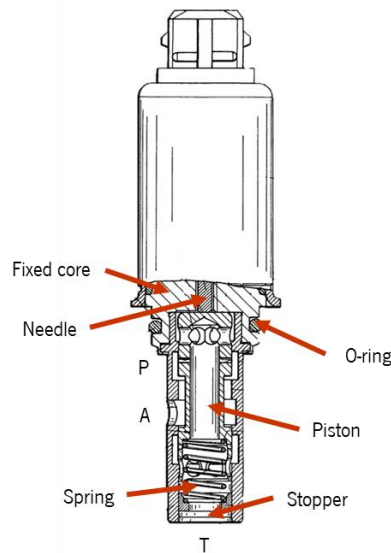


Figure 36: Cross section view of a 3/2 proportional solenoid valve [62]. (adapted)

This specific design pertains to an SV whose electrical subassembly is located outside of the engine, meaning that there cannot be leaks from the hydraulic circuit to the outside of the valve. Due to this, o-rings are used to contain the oil. If the SV was to be used inside the engine, the sealing would only be necessary on the connection between the interior of the engine and the outside environment – usually the electric connector.

In this design, initially, there is a piston blocking flow from ports P to A, but allowing it from ports A to T (normally closed SV). As the solenoid receives current, the mobile core is pulled downward, towards the fixed core, pushing the needle and the piston downward, and compressing the spring against the stopper. If the solenoid is fully energized, the piston will allow for unrestricted flow from ports P to A. If it is partially energized, the piston moves less, and flow from P to A will happen, but will suffer a greater pressure drop than in the first case.

This type of system can be applied in any system that benefits from continuous regulation when compared to on/off.

In clarification of what has been described, Figure 37 shows the behavioural distinction between on/off and proportional SV.

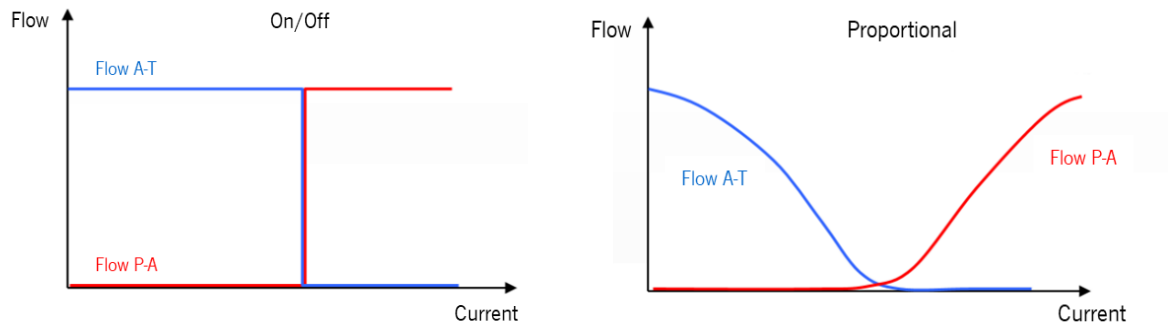


Figure 37: On/off and proportional solenoid valves flow versus current graphs.

3.3.3.1. Underlap, Zero Lap, and Overlap in Proportional Solenoid Valves

Following the design shown in Figure 36, there could be different concepts adopted when it comes to deciding piston height in relation to the diameter of the ports, which impacts the way the SV behaves in transient response.

The possible scenarios are:

- Underlap: height < port opening;
- Zero lap: height = port opening;
- Overlap: height > port opening.

Zero lap is impossible in practice, since there are always tolerances for the machined sizes of the parts and exact measures cannot be obtained, so there is always a slight under- or overlap, even if it is unwanted.

What this means from a functional point of view is clarified through Figure 38.

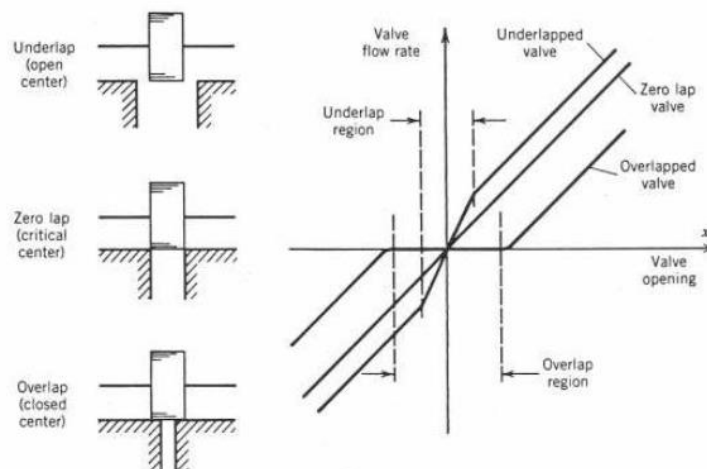


Figure 38: Underlap versus zero lap versus overlap effects on flow rate through valves in function of valve opening [63].

As shown, with zero lap, flow is directly proportional to the valve opening at a fixed slope. However, with underlap, there is a larger slope in the beginning. During that short period, both P to A and A to T flow can happen.

When there is overlap, the opening of the A port is delayed, so flow only begins a while after the piston has passed the port. In this situation, before P to A flow is possible, A to T flow has already been closed (the whole A port is covered by the piston).

3.3.3.2. Proportional Solenoid Valves with Second Level Regulation (SLR)

In case a proportional SV fails to activate, there is the possibility of opening it through a failsafe mode, usually called SLR. The SLR calibration takes its name from the Second Level Regulation of two-stage VDOPs, since the pressure at which it limits pump oil is equivalent to that of the traditional SLR, pictured in Figure 7, and consists of opening the passage of oil to the regulation chamber mechanically, rather than electrically.

Three ways of going about this mechanical regulation are in the pump itself (direct regulation), the engine, or the SV (indirect regulation), the latter being the object of focus in this study.

SLR mode in SVs take advantage of equation 3, where it is obvious that, maintaining pressure, the greater the area, the greater the force applied on it.

$$F = pA_s \quad 3$$

For force to be applied, different diameters are used on the piston (providing different surface areas), and the body of the valve also has to be shaped in order to facilitate this effect. This is the case when pistons such as the one shown in Figure 36 are used. A hand drawn exaggeration of the shape of the body and piston was drawn in Figure 39.

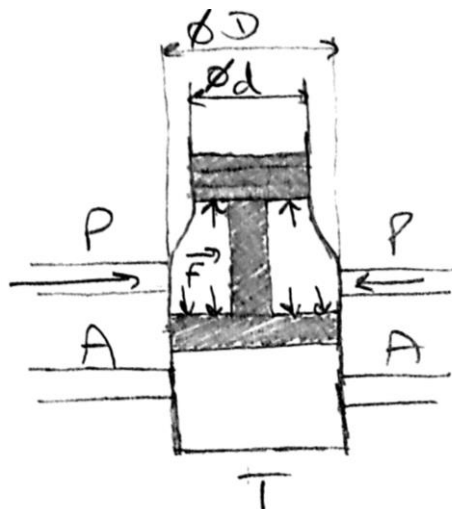


Figure 39: Hand drawn scheme of proportional valve body and piston for SLR mode (exaggerated). Piston coloured for clarification.

In this case, the body of the SV is narrower on top than at the bottom - $d < D$ - and the same happens on the piston. This way, the geometry of the latter is simplified. The same principle from equation 3 applies.

The calibration of when the valve opens – meaning how much pressure is necessary to cause the failsafe mode to open the valve – is done by pushing the solenoid valve's stopper against the spring. This compresses the spring, increasing the preload, meaning that a greater force acting on the piston is required to push it down.

3.4. Control System

For on/off SV, the current supply is either at the nominal value (on) or zero (off).

For proportional solenoid valves, current is applied as necessary, within the range from zero to the nominal current value. As such, there has to be an electronic closed-loop controller which regulates how much current passes through the solenoid, in function of a setpoint (defined by a control unit) and the measured pressure value at the exit of the pump (or at the oil main gallery).

At a given temperature, the electrical resistance of the solenoid is constant. Therefore, in order to limit current, the voltage applied on the circuit has to be changed (as per equation 4 – Ohm's Law), which is how the controller manages the SV.

$$R = \frac{V}{I} \quad 4$$

3.4.1. Voltage Adjustment – PWM Signal

The manner in which the voltage in the circuit is managed is by using Pulse Width Modulation (PWM). PWM is a method of controlling analogue circuits with a microcontroller's digital outputs [64], by varying the pulse width of a fixed frequency of a rectangular wave [65], with the value of the frequency depending on the application. A typical value for solenoid valve solutions is 250 Hz [48]. The pulse width sent out by the controller over time determines the average voltage "felt" by the load (in this case, a solenoid). An example is shown in Figure 40.

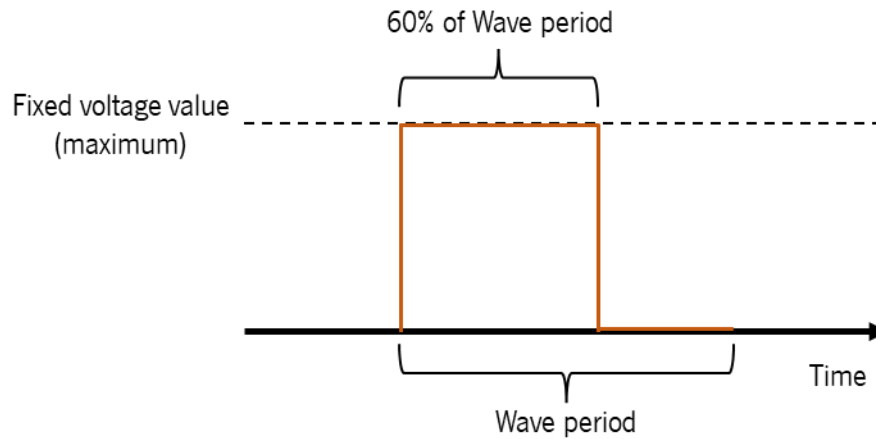


Figure 40: Exemplification of PWM output signal at 60 % (single wave represented).

In this case, the load would receive an average voltage of 60 %, which corresponds to the portion of the wave's period during which the output was on.

The microcontroller calculates the percentage of time it has to send “on” pulses via a PID system.

3.4.2. PID Controller

A PID controller is a system which allows for a better following of the setpoint and better transient response than a typical on/off controller [66].

This type of control system has three adjustable actions: Proportional (P), Integrative (I), and Derivative (D). Each of these terms can be adjusted by attributing a certain value to them (gain), increasing or diminishing their influence on the output value as necessary.

Very often only PI controllers are used, usually by setting the derivative term to zero (nullifying its functionality).

4. METHODOLOGY

In this chapter is a description of the methodology followed in conducting the study of the system at hand, with description of the object of study, a product and process analysis and depiction of the facilities in which the work has been carried out.

4.1. Object of Study

This project exists within Bontaz for one of its direct clients, which is a tier 1 supplier for an OEM. In this instance, Bontaz is a tier 2 supplier to the OEM, meaning the tier 1, Bontaz's direct customer, supplies

the OEM with their product already fitted with a Bontaz solenoid valve. The object of study consists of a vane VDOP system equipped with a Bontaz 3/2 proportional solenoid valve.

The assembly is under study following a complaint from the OEM to the tier 1 supplier, who has relayed the problem to Bontaz after failing to solve it within the pump. The problem consists on pressure oscillations which cause alarms in the OEM's fault detection equipment, possibly masking other potential problems.

An investigation into this problem had already been started at Bontaz France and Bontaz USA (BCUS), so a pump and test fixtures were already available for testing the system. In order to provide support from Bontaz Portugal (BCTP), it was requested that the available equipment was sent to Portugal.

In the beginning of the development of the present work, communication with the customer was put on hold due to an agreement made between the tier 1 and the OEM, which allowed around 99 % of otherwise NOK parts to be accepted. This was not a solution to the problem, but a workaround, and so Bontaz decided to keep investigating.

4.1.1. Customer's Pump

The pump in study is a vane VDOP. It regulates flow output via rotation of the outer ring and is piloted by a proportional solenoid valve which adjusts the pressure of oil that reaches the regulation chamber. The pump is shown as it arrived in Figure 41.

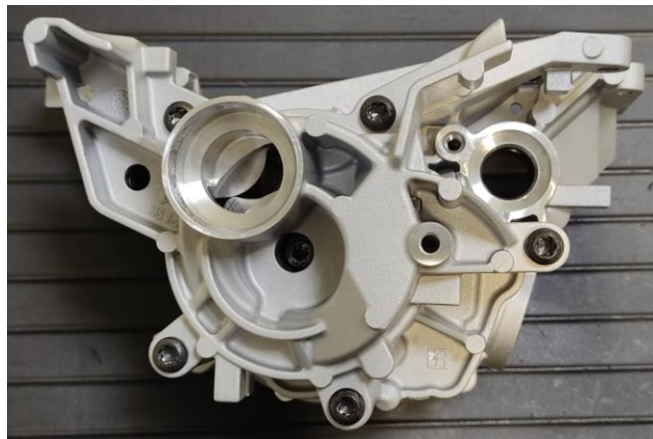


Figure 41: Object of Study: Vane VDOP.

It is noticeable from analysing the pump on the outside that it has two different modules, one of them being a gerotor pump (which is visible through its intake port in Figure 41).

From this analysis, and according to the previous research, despite no indication from the customer, it is possible to conclude that the pump in question is applied to a dry sump system, and combines a pressure

pump, which supplies oil for the engine – the vane module – and a scavenge pump, which sucks oil from the sump and discharges it in the tank – the gerotor module - as per the example given in Figure 3.

The customer provided a CAD assembly file for the pump. From there, it was discovered that the SV intake pressure port was connected to the gerotor module in order to provide oil for lubricating its outer gear.

This CAD file does not, however, include any interior parts, so a deeper study of the pump was required to properly understand its functioning.

4.1.1.1. Pump X-Ray

Initially, to see inside the assembly and take measurements while avoiding having to dismantle the pump, the x-ray projection machine (Figure 42) was used to obtain images of its internal components.



Figure 42: RX Solutions Easytom CT scanner performing x-ray projection on the pump.

This machine has two main different functionalities – x-ray imaging and tomography. X-ray imaging results in still images of the interior of whatever is placed inside the viewing zone, whereas tomography yields a 3D model of the part or assembly, which can then be manipulated in dedicated software, enabling the user to view cross-sections of the object. The tomography route is more time consuming, so, when unsure of how clear the images will be, it is a good idea to experiment with x-ray imaging first. Placed on a rotating table, the part can be moved to provide the desired view, closer or further from the projector, along with the tuning of several parameters to get the clearest possible image.

From the first acquired perspective, it is absolutely confirmed that the pump in question is a double module pump, as two separate assemblies are seen inside the housing. It is, however, not possible to see the shape of the VDOP's stator, nor the profile of the gears, so another x-ray image was taken from

the rear of the pump (transmission gear side), where there should only be an aluminium wall impeding vision towards the vane pump module.

Despite having chosen this second orientation for the pump, the difficulty in getting details from the x-rays persists. This happens because dense materials, such as steel, appear darker in these x-ray images, as they absorb more radiation. Therefore, it is more difficult to see through assemblies with multiple dense components in line, as it happens in the pump. From here it was concluded that a tomographic analysis of the pump would not be justified.

Even though the obtained images were not as clear as desired, they are useful, as performing this analysis contributed to Bontaz knowledge regarding x-ray imaging of large assemblies and allowed perception of the pump's mechanisms to a certain degree. It also showed that the vane pump uses eight vanes, and the position of the regulation mechanism.

4.1.1.2. Components' Measurements

After disassembling the pump and its subassemblies, the components were measured. Due to the relatively complex shapes of some components, different measurement techniques were employed.

For the simpler dimensions, measurements were done using a calliper, however, the rotor and the outer ring, for example, require the use of other instruments.

Using the tomographic analysis feature of the CT scanner shown in Figure 42, a 3D model of the rotor and shaft subassembly was exported. This was instrumental in acquiring the radii of the circular profiles on the inner side of the vane grooves, as well as the positioning of their respective centres in relation to the centre of the shaft.

In the case of the outer ring, a Mitutoyo profile projector was used instead. The projector is equipped with its own distance and radius calculator, making it extremely useful for accurately determining profile dimensions.

4.1.1.3. Full CAD Assembly of the Pump

After acquiring all the necessary values, the internal components of the vane module were modelled and assembled through the PTC Creo Parametric software.

From this 3D model of the pump, several parameters can be extracted or manipulated, paving the way for a better understanding of variable displacement pump mechanisms and the creation of new tools, such as simulation models. Besides this, it may serve as an example to explain the function of oil pump solenoid valves to new employees, among others.

4.1.2. Bontaz Solenoid Valve – 146320

The solenoid valve under analysis is a 3/2 proportional model, pictured in Figure 43. Its function is to control the flow of oil to the pump's regulation chamber, making it regulate more or less, according to the ECU's commands (current provided to the solenoid).



Figure 43: Object of Study: 146320 3/2 proportional solenoid valve.

Required dimensions and technical specifications are declared in the customer drawing. Within these, there is electrical data such as:

- PWM input signal frequency: 250 Hz;
- Nominal voltage: 12 V;
- Coil resistance: 9.9Ω @ 20°C .

As well as the hydraulic data of the SV:

- Failsafe mode (SLR) calibration: $p_P = 6.2 \text{ bar} \rightarrow p_A = 1 \text{ bar}$;
- P-A flow: $> 1 \text{ L/min}$ @ 0.8 A, with $p_P = 1 \text{ bar}$;
- A-T flow (SV off): $> 2 \text{ L/min}$, with $p_A = 0.2 \text{ bar}$.

Additionally, the pass-through connector's maximum leakage limit is specified as being $1 \text{ cm}^3/\text{min}$ @ 0.5 bar.

For a better understanding of the SV, the valve's hydraulic scheme is displayed in Figure 44.

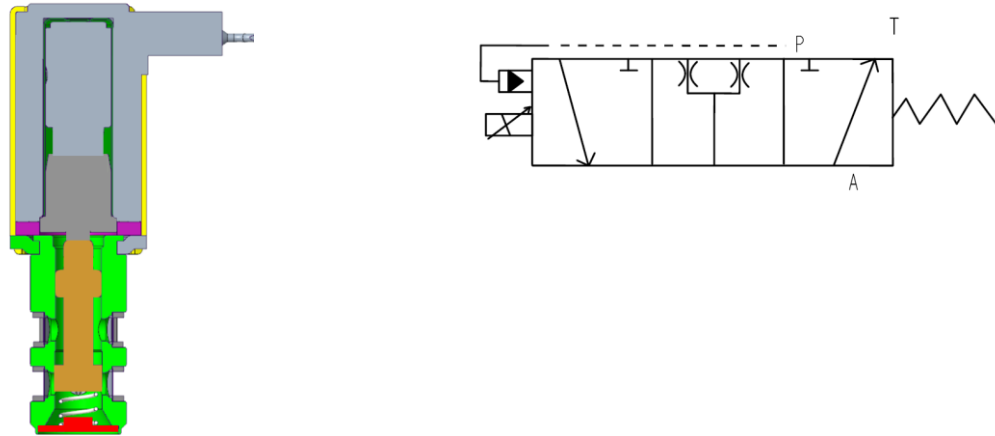


Figure 44: (a) Cross-section of the 146320 SV (simplified); (b) Hydraulic scheme of the 146320 SV.

Components are analogous to those of Figure 36, with the main difference between the designs being the lack of o-rings, because this SV is located inside the engine. In this case, the o-ring is located around the connector (orange o-ring shown in Figure 43).

The hydraulic scheme provides additional information about the valve:

- Proportionality – represented by a diagonal arrow on the solenoid symbol;
- SLR function - represented by the dotted line going from port P to the valve's hydraulic pilot;
- Underlap state – represented by the constrictions on the flow lines in the middle position.

4.2. Product and Process

4.2.1. Product Overview and Production Register

As with every other part manufactured by Bontaz, the SV in study must have a defined Bill Of Materials (BOM) and assembly flowchart, which are present in the product overview, and production register documents, respectively. This way, there is an overview of the product and of the parts used to produce it, and the assembly steps are well documented and easy to access. In order to better understand the product, a simplified BOM was elaborated in Table 2:

Table 2: Bill of Materials of the 146320 Solenoid Valve.

146320						
Sub assemblies					P/N BTZ	Designation
146320 - Electrovanne huile	146313 - Partie elétrique	146310 - Bobine	146315 - Faisceau	146309 - Brin	146319	Brin
					192207	Contacteur
				146317 - Brin	146316	Brin
					192207	Contacteur
					146078	Joint
					192206	Joint
				146308	Gaine	
				MAT802	Faisceau	
			146311 - Carcasse bobinée		146318	Carcasse
					192118	Contacteur
				MAT917	Fil de cuivre	
				MAT824	Bobine	
					105111	Rondelle magnétique
					125428	Noyau magnétique mobile
					146312	Manchon
					146314	Plaque
					146322	Enveloppe
					146323	Aiguille
					160487	Douille
		146324 Partie mécanique				101213
					146321	Corps électrovanne
					146329	Tiroir
					240104	Ressort
					154546	Filtre
					240108	Rondelle magnétique

The bill of materials is organised in such a way that its constituents, from left to right, become simpler, starting from the full solenoid valve, and going down to each individual component. Although not relevant to this study, the mass and material of each component is also characterised in this list.

As previously explained, there are the two main subassemblies – the electrical 146313 and the mechanical 146324 - and, within the electrical one, there are other, smaller groups. Components such as the filters and the magnetic half-washers are not included in either main subassembly and are, therefore, at the bottom of the list.

The assembly flowchart that is used when assembling the solenoid valve, whose steps are followed either in prototype assembly (the present study) or in series production is also shown in Table 3:

Table 3: Assembly Flowchart of the 146320 Solenoid Valve.

STEP	FABRICATION	MOVE	STORE	CONTROL	SPECIAL	OPERATION DESCRIPTION DESCRIÇÃO DA OPERAÇÃO	AFFECTED CONTROL CHARACTERISTICS CARACTERÍSTICAS CONTROLO AFETADAS	INDEX ÍNDICE COMPONENTE
1					15	Move Subset Elec ASSY from storage (146313)		C
2					15	Move Subset Meca ASSY from storage (146324)		B
3					15	Move half washer from storage (240108) x2		D
4						Washing (white room entry)		
5						FRONT CRIMPING PREPARATION		
6						FRONT CRIMPING + VACCUM	Crimping	
7					25	Dielectrical test + Surge test + Resistance control + Datamatrix marking		
8					30	Adjustement + Final functional checking + Marking	Height	
9					35	Move Filters from storage (154546)		C
10						ASSEMBLY		
11					45	General Visual Checking	Components missed + Visual aspect ...	
12					50	PACKAGING	General aspect of the packaging	
13						Storage		
14						Move to BONTAZ US PLANT		
15					55	General Visual Checking (GP12)	Components missed + Visual aspect ...	
16						Storage		

The flowchart contains five different operations – fabrication, movement, storage, control, and a special category for other items. For clarification, the “move” category refers to internal shipments of the parts from one Bontaz Centre to another, and the “special” category, in this situation, is used for movements of components within the assembly line.

The flowchart begins with the acquisition of the already complete electrical and mechanical subassemblies. Similarly to the solenoid valve, those subgroups also need to have production registers, which are shown in Annexes 1 and 2, respectively.

For a better comprehension of the assembly process, it will be described in more detail.

4.2.2. Assembly of the Solenoid Valve

Building the 146320 SV involves a series of defined steps, already shown above. On the prototype assembly lines, since production runs are small (usually ≤ 200 prototypes), assembly is done by hand and not automated at all. In Figure 45, components have been laid out on a prototype line table:

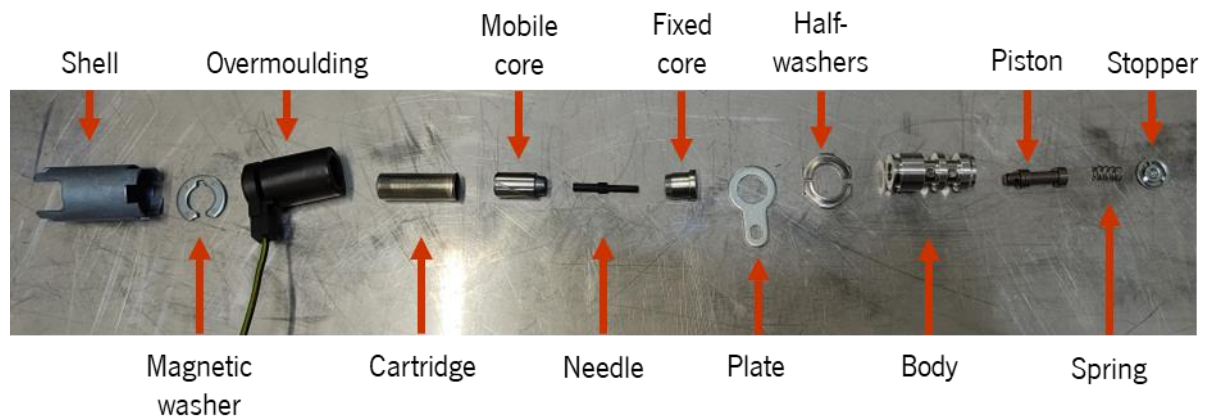


Figure 45: 146320 SV components lined up for assembly (prototype line).

To assemble the SV, the following steps are done:

- Electrical subassembly (146313):
 - Place magnetic washer in the coil overmoulding;
 - Place overmoulding with magnetic washer inside the shell;
 - Insert cartridge, mobile core, needle, and fixed core in the overmoulding;

At this point, a crimping operation is performed (Figure 46), using an electric press, to secure the components inside the shell during the next steps.

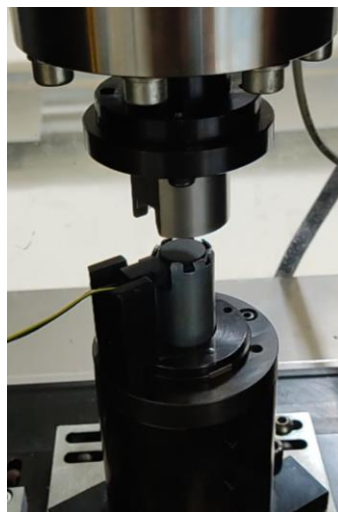


Figure 46: Crimping of the electrical subassembly (by pressing on the flaps atop the shell)

Following the crimping:

- Place the plate at the bottom of the electrical subassembly;
- Insert the half-washers on the body;
- Mechanical subassembly (146324):
 - Place the body with half-washers on the plate;
 - Crimp the shell a second time to secure the body (Figure 47);
 - Insert piston, spring and stopper in the body;
 - Crimp the stopper (Figure 48).



Figure 47: Solenoid valve after second crimping operation.

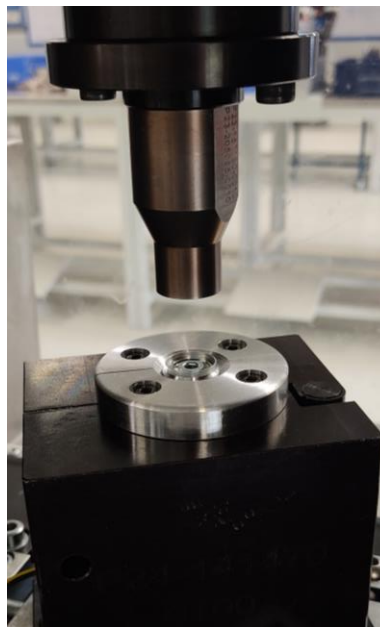


Figure 48: Crimping of the stopper (by pressing on the edge of the body).

After this third and last crimping operation, the valve will be as shown in Figure 49:



Figure 49: Solenoid valve after last crimping operation.

From here, the only parts left to be assembled are the filters at the P and A ports, however, the A port filter can only be installed after SLR calibration of the valve, explained in 4.4.2.

4.3. Test Equipment and Facilities

In order to verify correct functioning of the prototypes produced at BCTP, the site is equipped with test benches designed to accommodate testing, according with the specific requirements of each product. Among this equipment are the prototype test benches, which in this case are used to verify SLR, P-A flow, and A-T flow, and the pump test benches, whose function is to replicate test conditions on the customers' side and observe oil pump operation. Measured variables can be monitored live and be stored for further analysis, using dedicated software.

4.3.1. Prototype Test Benches

The prototype test benches consist of a working area (Figure 50) containing:

- Oil reservoir;
- Oil supply hose with flow meter;
- Electric current supply for solenoid valves;
- Oil pressure sensors and temperature probe;

Enhancement of proportional solenoid valve to manage variable displacement oil pump

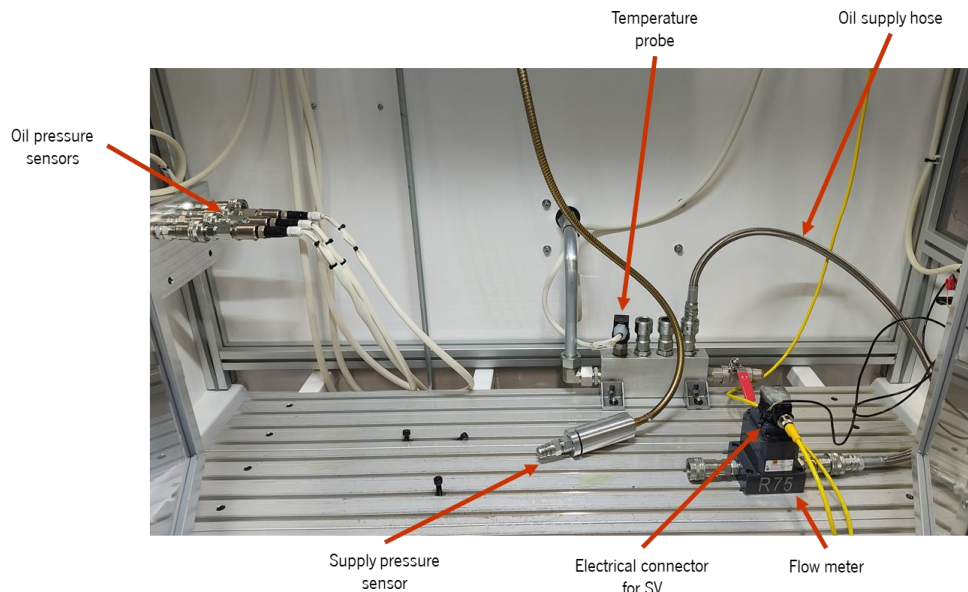


Figure 50: Prototype bench working area.

For the SLR and flow tests, a fixture in which the SV, supply hose and sensors can be connected is required. In this case, a standard Bontaz fixture was used (Figure 51), which means it was not necessary to develop a new, purpose-made one.

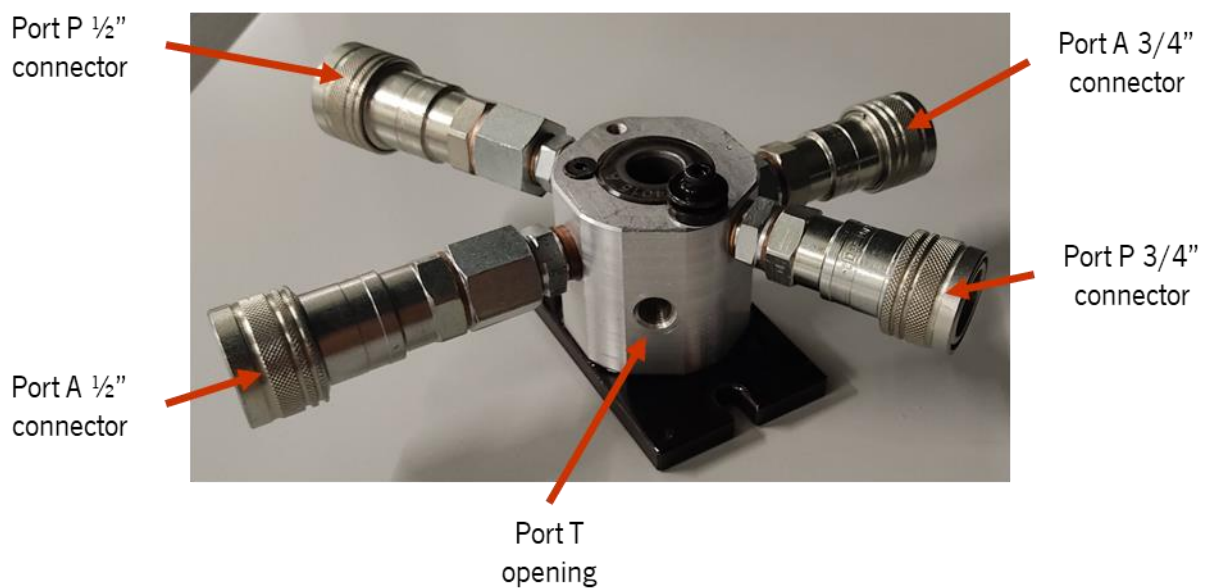


Figure 51: Standard fixture used in bench testing.

Ports P and A have two possible connection lines each, whose female connectors are closed by default, until connected with a male connection line. Port T is left open, as it represents a tank at atmospheric pressure.

The control interface of the benches (Figure 52) provides two different ways to operate the tests – automatic or manual – and allows the user to change parameters such as oil temperature and supply oil pressure control PID values.

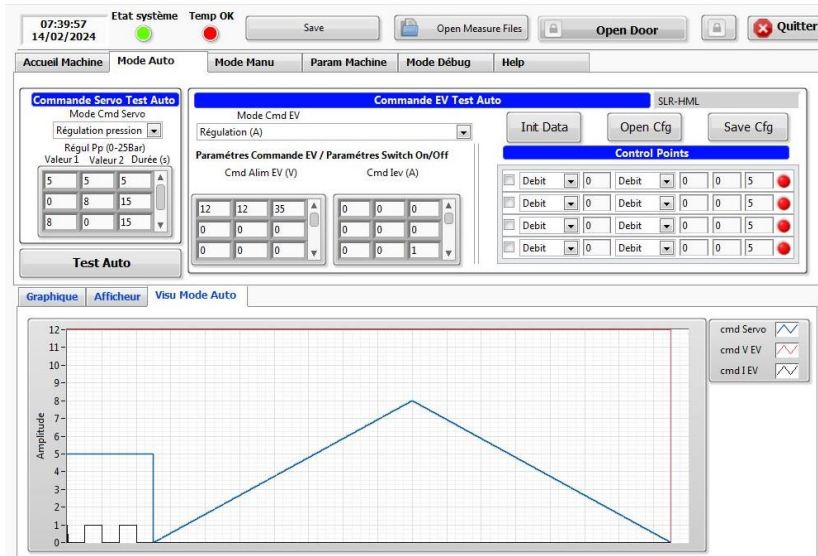


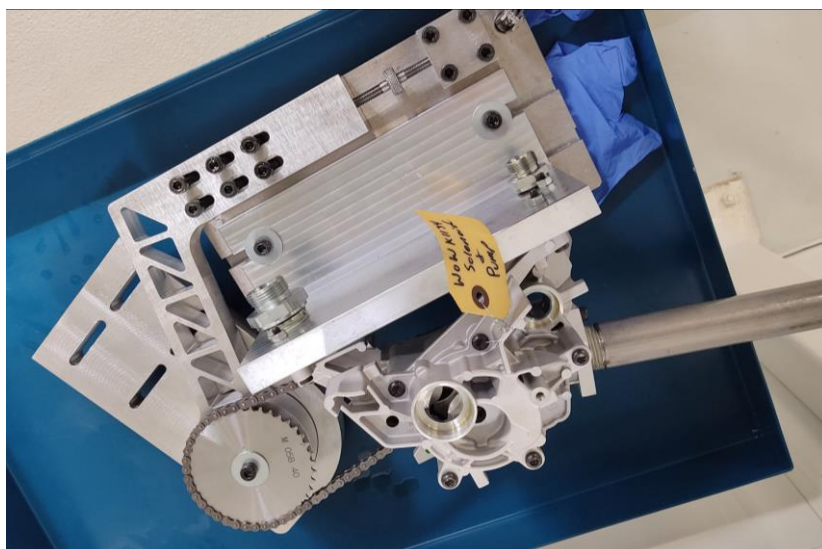
Figure 52: Prototype test bench interface in automatic testing mode (with loaded test file).

4.3.2. Pump Test Bench

In order to use the pump bench, a standalone fixture needs to be made specifically for each project.

4.3.2.1. Fixture

As previously explained, the fixture had been made by the team which first worked on this project, but it was necessary to reassemble it and fix the pump. The resulting assembly is shown in Figure 53 (a).



(a)

(b)

Figure 53: Fixture used for pump testing with pump assembled (a) and fixture shaft assembled onto the bench motor's shaft (b).

The fixture has four functions:

- Hold the pump in place;
- Connect bench hoses to the pump;
- Attach drive gear to the bench motor's shaft and drive the pump;
- Allow regulation of chain tension.

The pump is attached to the fixture via five bolts, in three locations. There are gaskets in each support point between the pump and the fixture to avoid leaks. The bench hoses connect directly onto the male hydraulic connectors on top of the fixture.

The connection to the bench's shaft is done by tightening the fixture's shaft around the motor shaft with a bolt (Figure 53 (b)). The drive gear has 40 teeth and the driven gear has 20 teeth, meaning the pump rotates at twice the speed of the bench motor.

To regulate chain tension, the six bolts on the regulating arm have to be loosened. Then, the adjustment screw is rotated in the desired direction, pushing or pulling the pump in relation to the fixed drive gear. After this adjustment, the six bolts are tightened once again, securing the pump at the set distance.

4.3.2.2. Pump Bench Setup

The intent when using the pump bench is to simulate engine operating conditions, so there have to be restrictions to the flow of oil similar to those present inside the engine. As explained in 2.1, oil flows from the pump to the filter, then to a main gallery, and finally towards the necessary lubrication points, through small channels.

To simulate the pressure drop caused by the oil filter, there is a valve which can be partially closed (henceforth called "vanne 1"). To simulate the backpressure caused by the small cross-section of the engine's lubrication channels, there is another similar valve downstream, henceforth called "vanne 2". To decide how much each vanne should restrict flow, the customer provides pressure values from their own testing, which can then be correlated to those obtained at Bontaz. Typically, opening values operated are of 80 % for vanne 1, and 60 % for vanne 2 as an internal standard.

Having said this, the pump test bench's working area comprises the following elements:

- Oil reservoir;
- Built-in pressure sensors (P1 and P2) and temperature probe;
- Electric current supply for solenoid valves;

- Two pneumatic valves (henceforth called “vannes” for clarity);
- Flow meter.

The setup can be better understood in the hydraulic diagram provided in Figure 54.

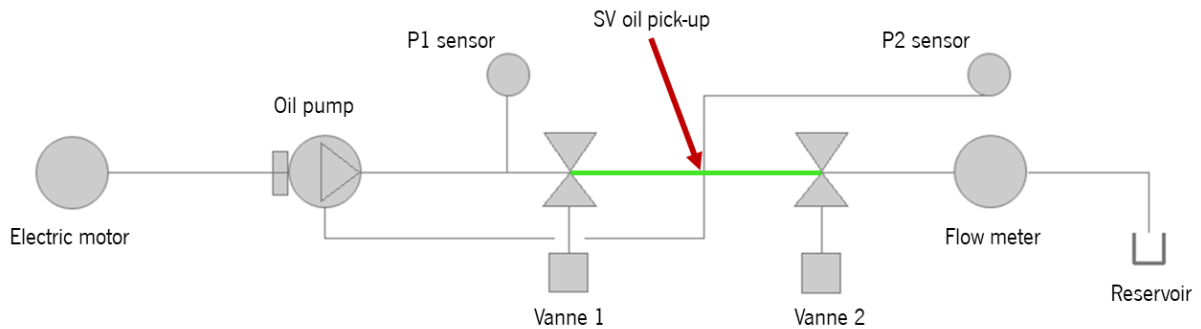


Figure 54: Hydraulic diagram of the pump bench - oil pick-up point (channel that leads to the solenoid valve) indicated. Green line represents the zone that simulates engine main oil gallery.

The pump is driven by the bench's electric motor, drawing oil from the reservoir (not shown). The P1 sensor measures output pressure. The line between vanne 1 and 2 (in green) acts as the main gallery, from which there is an oil pick-up to the solenoid valve, in order to regulate the VDOP. Sensor P2 measures pressure in this gallery. The remaining oil passes through a flow meter and is returned to the reservoir. The pump and the solenoid valve are placed in the working area, supported on the pump fixture, shown in Figure 55.

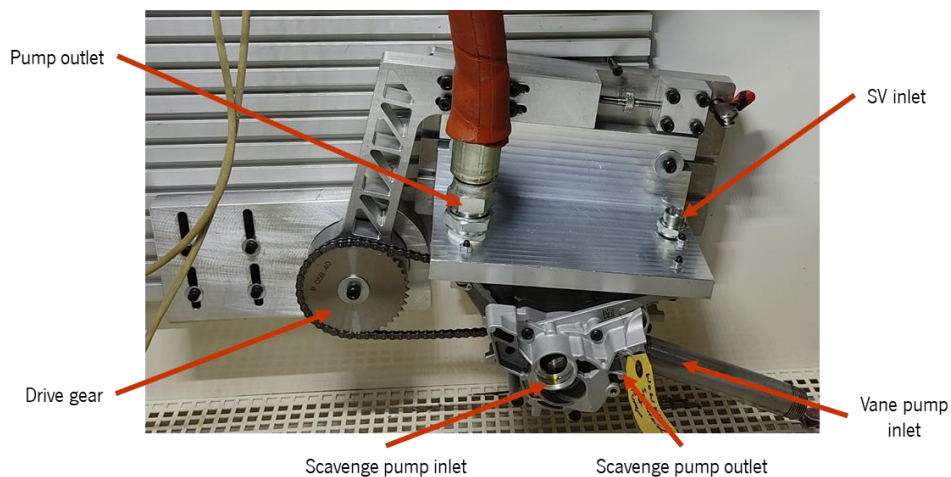


Figure 55: Pump bench working area. Pump attached to fixture and mounted in the bench.

Pressure at the entrance of the solenoid valve is equal to P2 minus the pressure drop caused by the line that carries the oil towards the valve.

The bench where the fixture was made to be attached was slightly different from the one at BCTP. As such, initial testing showed some problems. The problems and the solutions devised to solve each of them are schematised in Table 4.

Table 4: Problems found during initial pump bench tests.

Problem found	Solution devised
Fixture grinding against shaft;	Placed washers on mounting bolts behind fixture
Drive gear not centred on bench shaft	Placed a tighter insert in the centre of the gear
Drive chain misaligned	Designed a washer to bring the drive gear into alignment with the driven gear
Driven gear slipping on pump shaft	Rebored hole diameter, retightened and applied structural adhesive
Sensors' response time too slow and sensitivity too low	Installed quicker, more sensitive sensors

Firstly, the fixture ground against the shaft's (Figure 53 (b)) wider section zone, causing instability. The mark left on the fixture's bearing can be seen in Figure 56 (a). Placing washers behind the fixture was enough to provide clearance from the spinning shaft, solving the issue.

Then, the drive gear had to be properly centred, but its inner diameter was significantly larger than the bolt which fixed it to the shaft, so the diameter was reduced by placing an insert in the centre – see Figure 56 (b).

In order to align both gears to ensure correct chain positioning, the drive gear had to be brought away from the shaft 7 mm. By machining a washer with this thickness Figure 56 (c) and placing it between the drive gear and the shaft, the drive gear as attached in the correct position.

Lastly, as the driven gear slipped on the pump's shaft, the surface of the hole suffered some damage. This was corrected by reboring the hole. To ensure slippage would not reoccur, a structural adhesive was added - Figure 56 (d) - which fixed the problem.

Enhancement of proportional solenoid valve to manage variable displacement oil pump

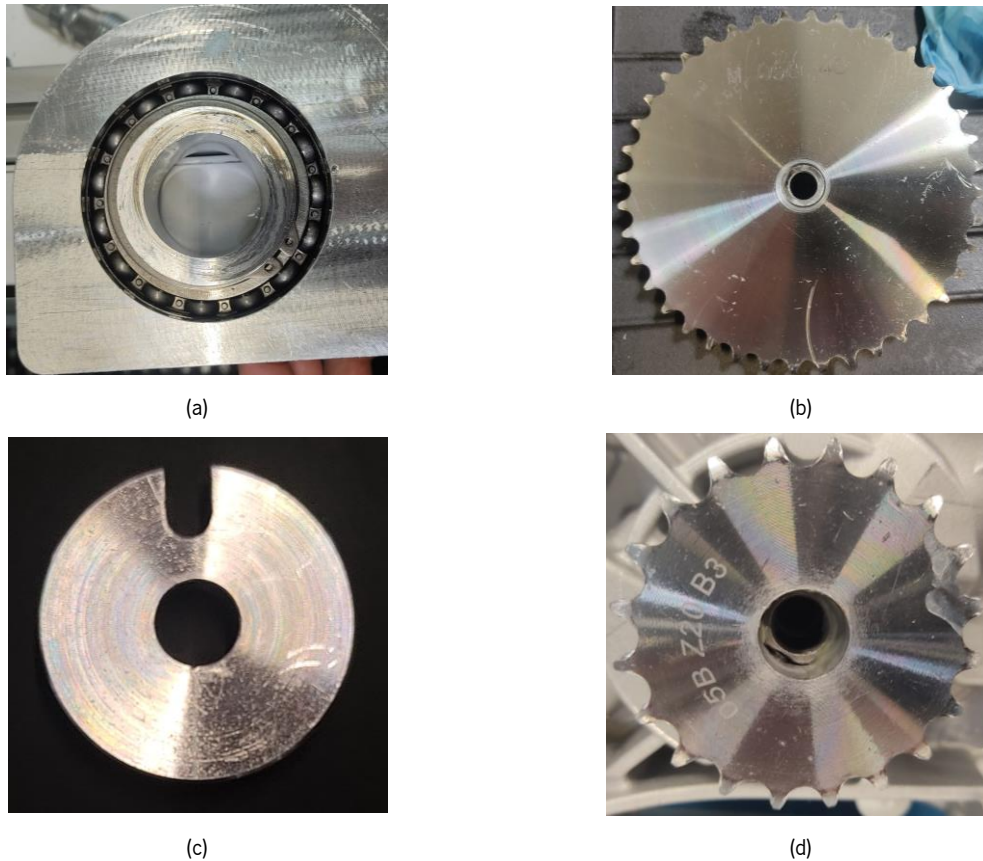


Figure 56: Problems with the pump bench setup. Ground bearing inner ring (a); Drive gear with added insert (b); Spacing washer to align drive gear with driven gear (c); Rebores driven gear with structural adhesive (glue) (d).

Finally, the pump test bench's control interface setup screen is shown in Figure 57.

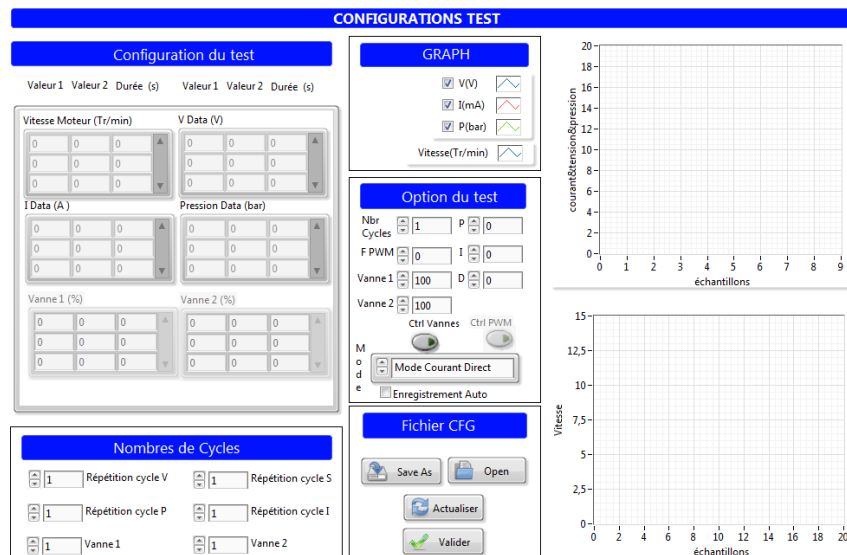


Figure 57: Pump test bench setup interface. Parameters required to run a pump test are: pump rotation speed, SV voltage, desired current, and vannes' opening percentages. Each of these parameters is written in steps with an associated duration, in seconds. Vanne openings can be set as a fixed value or vary throughout the test.

4.4. Functional Tests

All Bontaz solenoid valves are functionally checked with oil. In the prototype phase, test benches are used to carry out the functional performance checks, which aim to verify correct behaviour of the valve according to customer specifications (hydraulic data specified in 4.1.2). However, in order to analyse the tests, it is necessary to take the data retrieved by the test benches and graph it using an appropriate software.

4.4.1. Test Curve Analysis Software – Bontaz Data Processor

Bontaz has developed an in-house data processor in Octave for the purpose of analysing test data, which can be used to read parameters related to any of the required tests.

Within this software, the user can change displayed variables, customise axes (displayed variables, scales, gridlines), as well as track values in the curves. Another important feature is the filtering of high frequency noise, which could be created by electromagnetic interference. Presented in Figure 58 are two different graphic representations of the data collected through a disconnected pressure sensor (0 bar):

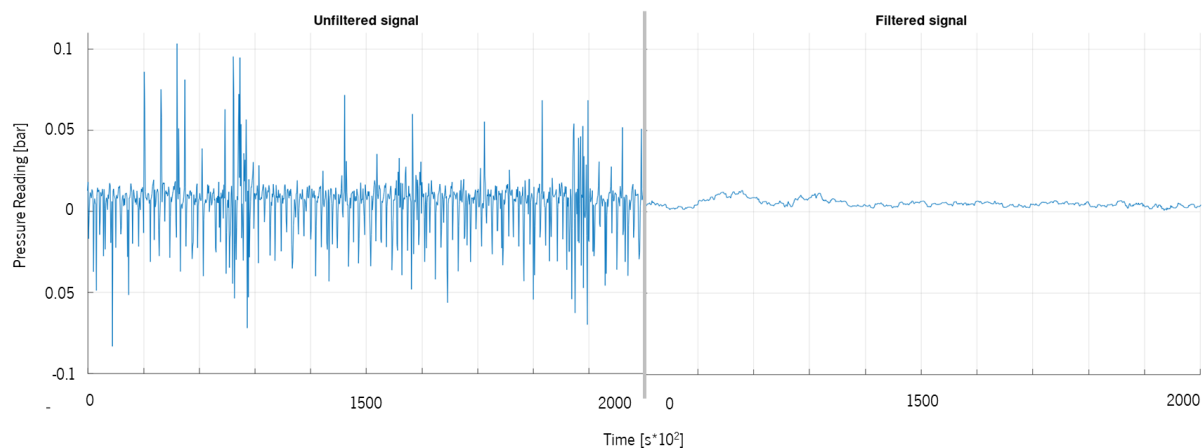


Figure 58: Two different graphics obtained from the same data: On the left - unfiltered signal, showing noise. On the right - filtered signal with minimal noise.

Although sensor noise is of a relatively small scale overall, it can reach peaks of up to 0.1 bar, which could hinder analysis. By filtering the noise, it is reduced to insignificant levels.

Using this software, any number of curves can be compared at the same time. In the following chapters, several graphics created with this data processor are going to be displayed.

4.4.2. SLR Calibration

The first functional test consists of the calibration of the valve in order to provide the correct mechanical regulation pressure. The test procedure is as follows:

- close port A;

- apply pressure sweep at port P - from 0 bar up to 8 bar;
- read corresponding pressure at port P when pressure at port A is 1 bar.

When starting calibration, the stopper has not been compressed yet, so the spring has more space, meaning the pressure required to open the valve is too low. After checking that the pressure at port P is lower than the 6.2 bar requirement when pressure at port A equals 1 bar, the valve is removed from the test bench and placed in the fixture shown in Figure 59.

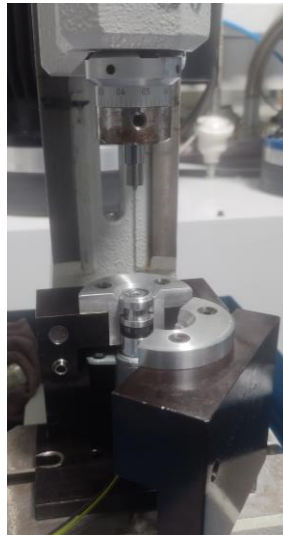


Figure 59: SLR calibration fixture with solenoid valve placed (filter not yet placed on port A).

Here, the operator can push the valve's stopper down 0.02 mm at a time (minimum scale on the press) and retest the valve until the SLR pressure corresponds to the desired value. Further pushing of the stopper results in a progressively higher SLR pressure (Figure 60), until it is within the defined tolerance.

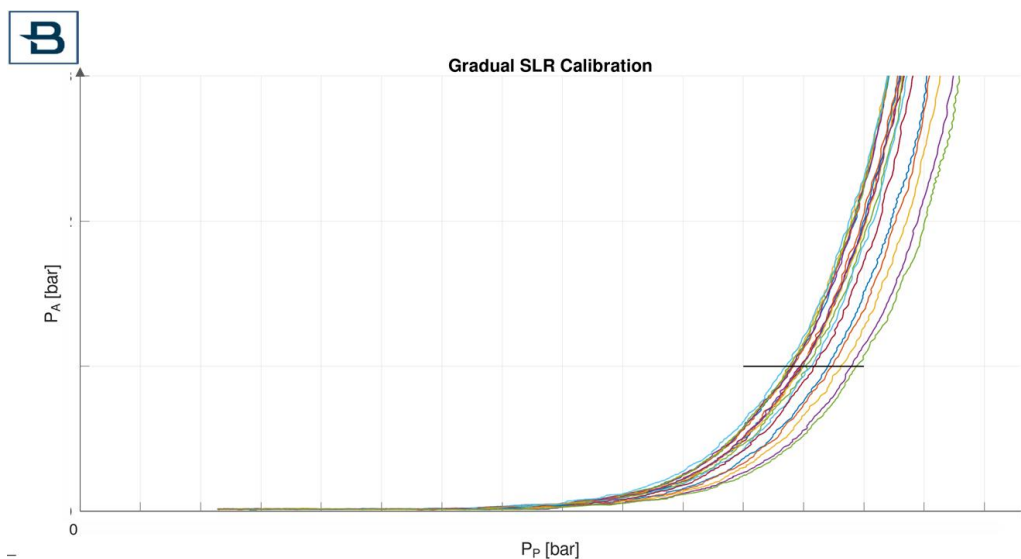


Figure 60: Gradual increase (from left to right) of P port pressure necessary to attain 1 bar at port A, as the stopper is pushed further.

Black line represents 1 bar at port A.

In this case, the SV is considered OK when P port pressure value falls between 5,7 and 6,7 bar for an A port pressure of 1 bar, like in the example shown in Figure 61.

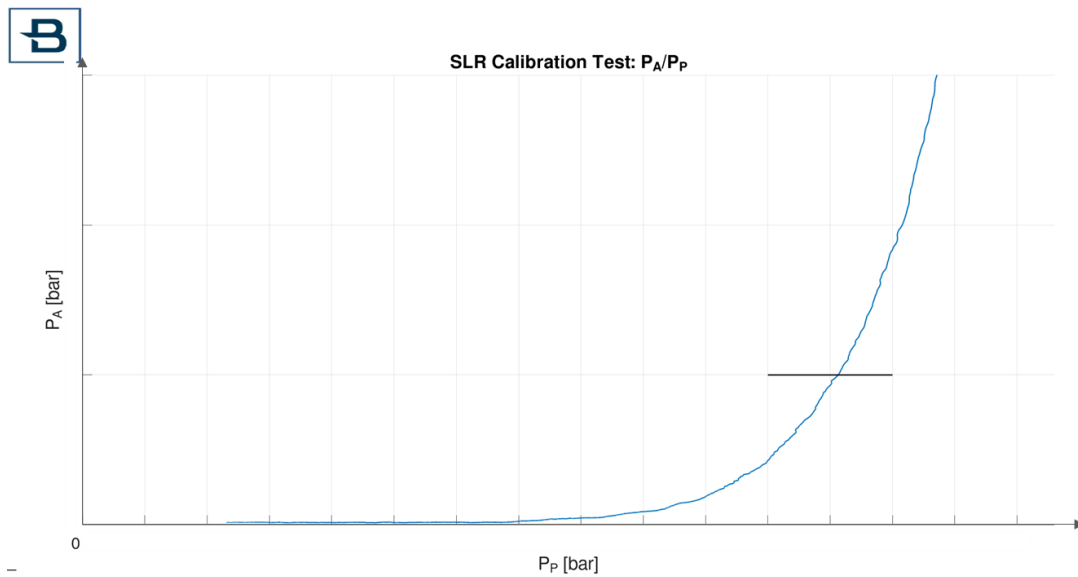


Figure 61: SLR curve of a calibrated SV. SV is OK if the curve rises through the black line. Example shown: 6.26 bar SLR pressure calibration.

4.4.3. P-A Flow

The second functional test checks the filling flow capacity of the valve, seeing how much oil can flow through the valve to fill the pump's regulation chamber, in this case. For this, the operator has to:

- open port A;
- apply a constant supply of 1 bar at port P;
- apply current sweep - from 0 A to 1 A;
- read flow value when current is at 0.8 A.

The SV is considered OK when the flow value is greater than 1 L/min at 0,8 A, so an example of a good flow curve is shown in Figure 62.

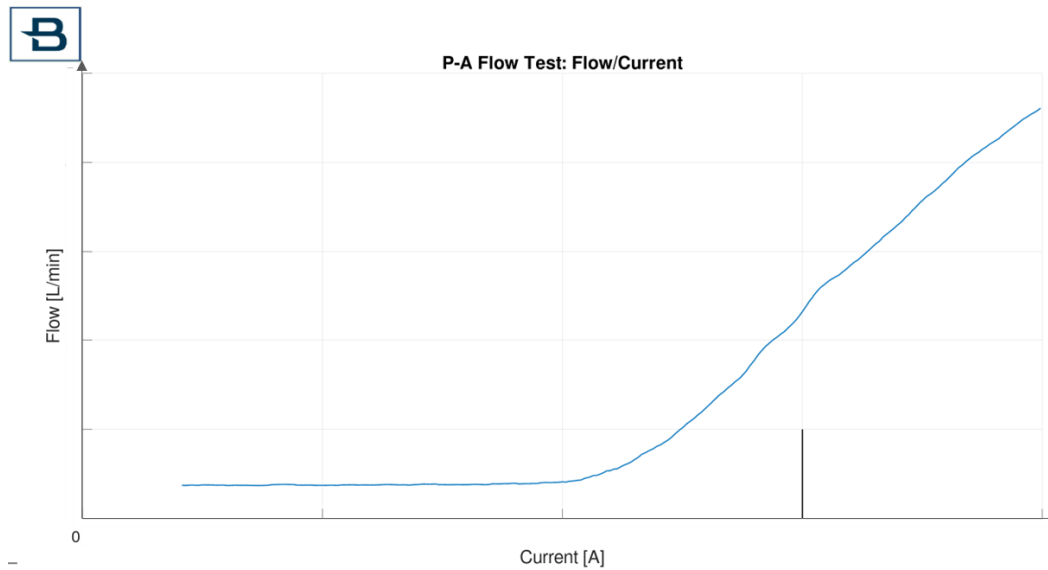


Figure 62: P-A flow test curve. SV is OK if the curve passes above the black line and does not intersect.

4.4.4. A-T Flow

The third functional test evaluates the valve's draining capacity. The procedure involves the following actions:

- apply pressure sweep at port A – from 0 to 1 bar;
- read flow value when pressure at port A equals 0.2 bar.

Flow must be greater than 2 L/min for the SV to be OK, such as the one shown in the graphic (Figure 63).

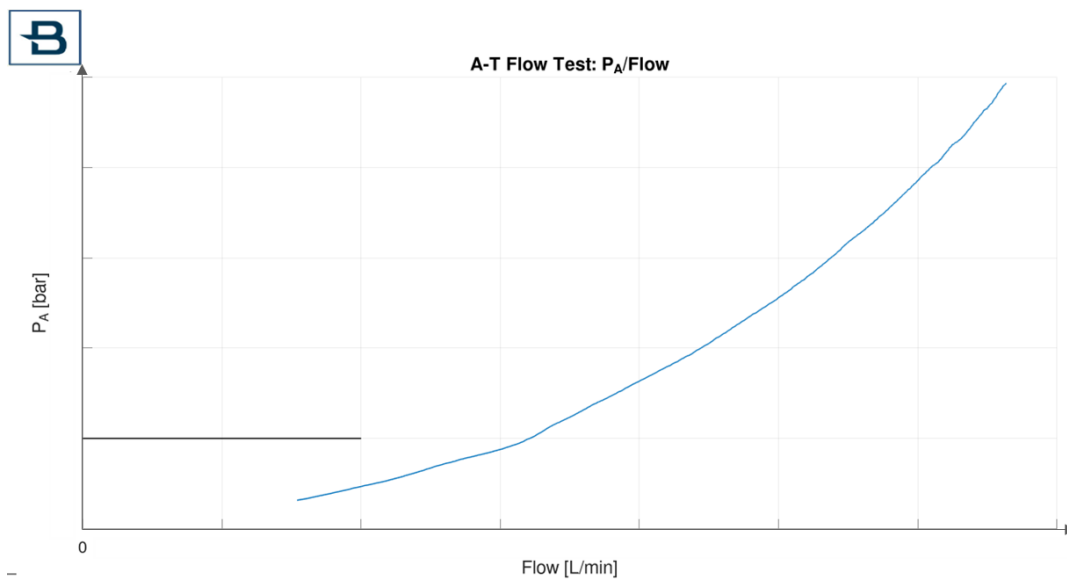


Figure 63: A-T flow test curve. SV is OK if the curve passes below the black line and does not intersect.

4.4.5. Pass-through connector leak test

To ensure that the connector's sealing properties match the requirements, a seal test is carried out, using a specific fixture, as seen in Figure 64. An Ateq F5200 leak tester is used to fill the space between the fixture and the connector seal with air at the specified pressure value and measure the leakage.

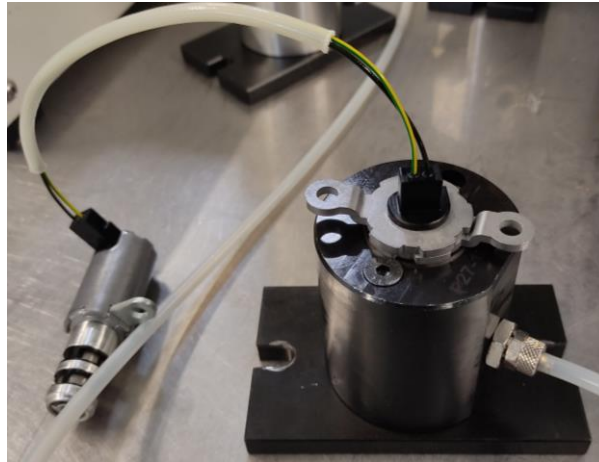


Figure 64: Pass-through connector leakage test: connector inserted in the fixture.

5. ROOT CAUSE ANALYSIS

Bontaz employs the 8D (8 Disciplines) analysis method for problem solving. An 8D analysis encompasses several quality tools, and is designed to find the root cause and ways to fix the problem, both immediately and on a longer time scale, preventing recurrence [67]. Among those quality tools are the 5-Why method, DFMEA, Ishikawa diagrams, evaluation matrix, among others, which enable a team to organise the steps to take in order to understand and tackle a specific problem.

These quality tools were instrumental to organise the thought process and decide which tests were necessary to carry out, along with their respective priorities.

5.1. Problem Description

As described in the introduction of chapter 4.1, the problem has been detected by the final customer, to whom Bontaz is a tier 2 supplier, i. e., supplies the OEM's tier 1 supplier. The problem is a first-time occurrence at Bontaz, and in order to tackle it, the first step is to briefly describe the issue. This was done using the dedicated chart, Table 5.

Table 5: Problem description table.

PROBLEM DESCRIPTION		
What	What happened ?	Oscillation of oil pressure at final customer's (OEM) EoL test. Bontaz customer (Tier 1) replicates issue on pump test stand.
who	Who detected it ? (Name and function)	Final customer (OEM)
When	When has it been detected and manufactured ?	Engine End of Line (EoL) tests.
Why	Why is it a problem ?	Triggers alarm for instability, may mask other potential problems.
How	How has it been detected ?	Test stand alarm triggered.
How many	How many parts ?	30% of parts tested by the final customer (OEM).

Alongside the information shown above, the team who had previously worked on this issue provided a graphic of a pump test bench curve, showing the oscillation and where it occurs.

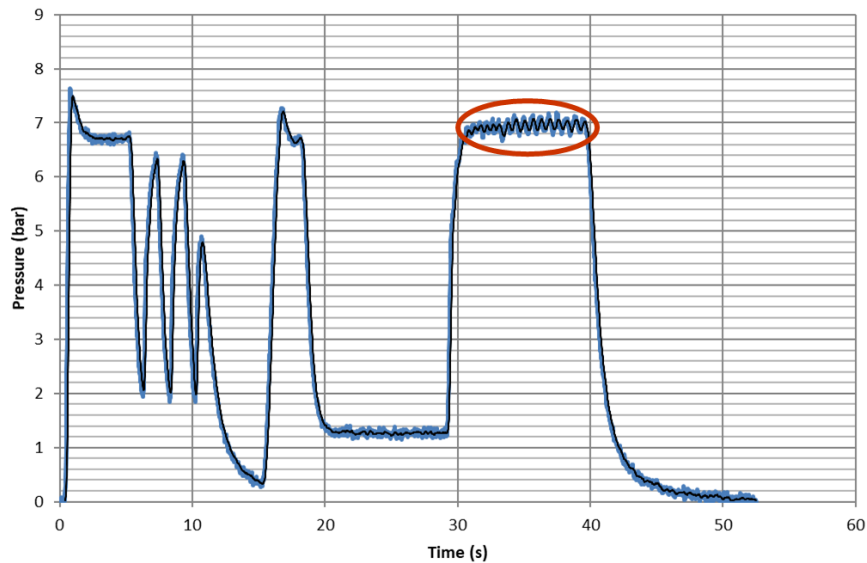


Figure 65: Pump outlet pressure over time. Problematic oscillation zone circled. Graphic provided by the previous problem-solving team, who tested the system on a pump test bench.

From the analysis of this graphic, it is possible to conclude that pump output pressure is not stable over time in the circled portion of the test, and the oscillation increases slightly at about one third of the oscillation part. Maximum oscillation appears to have a magnitude of around 0.2 bar. Using the 5-Why method, a brief description of the problem was elaborated:

Table 6: 5-Why method for problem solving applied to the problem in study.

FAILURE OCCURRENCE ANALYSIS	
Problem:	Customer EoL engine test pressure oscillation
Answer :	Pump outlet pressure not stable
1: Why	Pump outlet pressure not stable
Answer :	Regulation chamber not stable
2: Why	Regulation chamber not stable
Answer :	Pressure balance within pump not stable
3: Why	Pressure balance within pump not stable
Answer :	Regulation chamber inlet pressure oscillating
4: Why	Regulation chamber inlet pressure oscillating
Answer :	Pump outlet pressure not stable
5: Why	Regulation chamber not stable

As shown above, using only the 5-why method, the root cause of the problem remains unclear, since there is a recursive relationship between Why number 4 and 1, meaning the answer to Why 4 implies recalling the first question once again. This happens because, as has been previously explained, the oil used to regulate the pump comes from the pump itself. The instability might, then, be explained either through internal phenomena taking place within the pump, external factors such as the solenoid valve, or a combination of both.

5.2. DFMEA (Design Failure Mode and Effect Analysis)

At the time of development, each product has its own DFMEA, which aims to answer the following questions [68]:

- How might the product fail and how can failure be prevented?
- What are the consequences of a specific type of failure?;
- How is the problem detected?;
- What are the recommended actions to avoid these problems, if any?

For this reason, whenever an issue arises, it is wise to look for the failure mode in this document, as it may have already been considered. The predicted failure types for SLR mode in this product are shown in Annex 3.

None of the presented failure modes predict an oscillation in output pressure of the pump, so the problem has to be more deeply investigated.

5.3. Ishikawa Diagram

The preceding investigation team from BCUS had already resulted in an Ishikawa diagram, which is exposed in Annex 4.

As seen, the diagram is organised into eight different categories. The purpose is to organise the thought-process and help target the source of the problem more efficiently. It is possible that only part of the categories are relevant, so not all of the Ishikawa fields or categories have to be filled out. A brief explanation of the categories present on the Ishikawa diagram is exposed in Table 7:

Table 7: Scope of the possible root causes that can be included in each category of the Ishikawa diagram.

Category	Scope
People	Direct impact of humans on the tests.
Machine	Functioning of the system's mechanisms.
Materials	Raw materials, components used, part dimensions.
Method	Methodology followed during production.
Management	Management decisions that could affect the process.
Measurement	Techniques followed to measure quality standards.
Environment	External factors related to the system surroundings.
Money	Funding of the project.

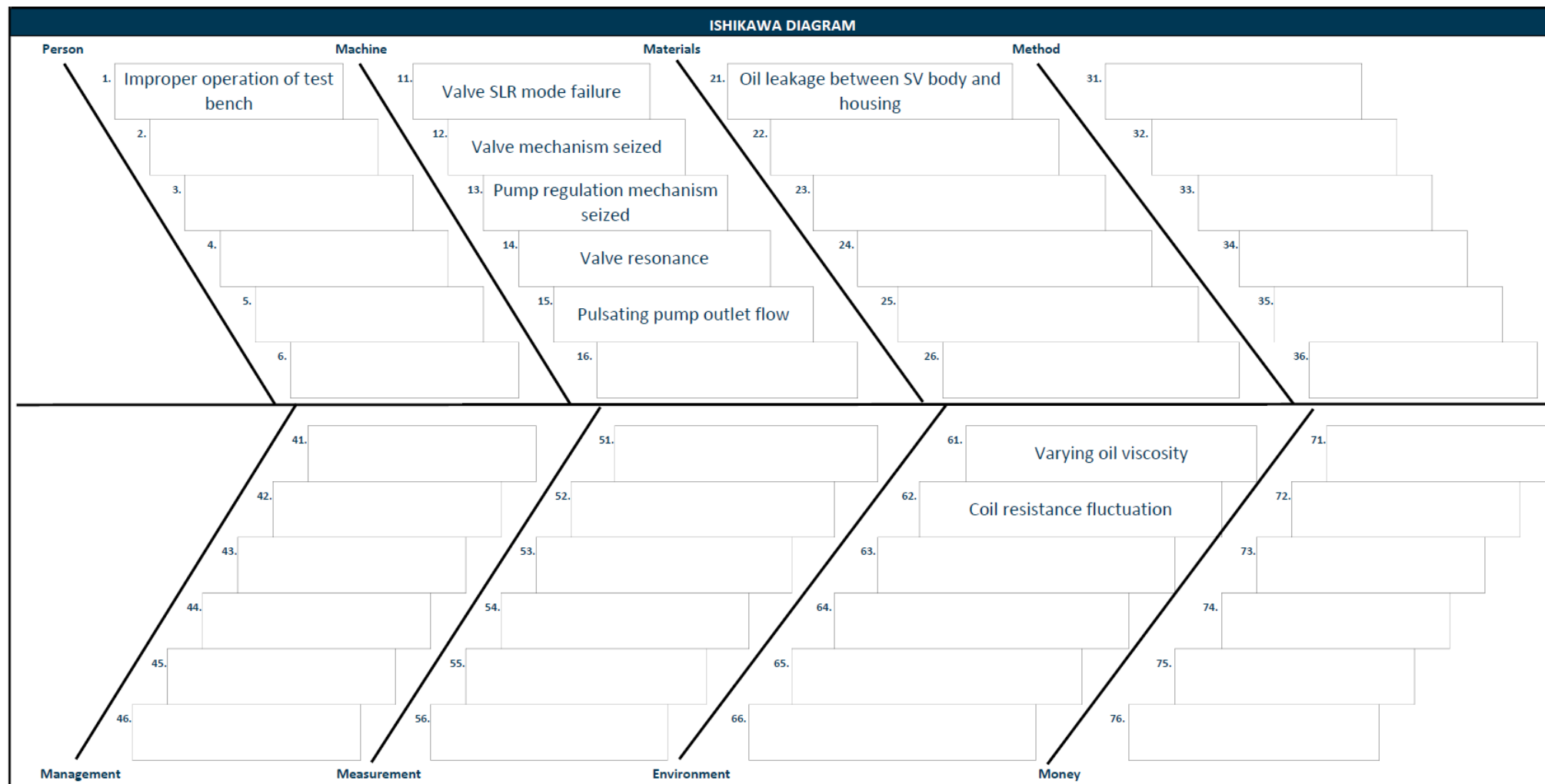
While investigating the root causes numbered on the diagram, they experimented with different changes to SV design. The most important experiments to note include:

- Varying underlap/overlap condition;
- Testing springs with different spring rates;
- Comparing bodies with highest *vs* lowest surface roughness;

Having said this, none of these attempts provided satisfactory results – all of the tested valves still showed occasional oscillation. For this reason, these experiments were not repeated in this study, and focus shifted to other different possibilities.

In order to study relevant parameters, a series of possibilities was proposed and placed into an Ishikawa diagram, shown below in Table 8.

Table 8: Ishikawa diagram with the possible root causes targeted in this study.



A total of nine possible root causes were considered, and the explanation of each is going to be elaborated in chapter 6.

5.4. Evaluation Matrix

A systematic approach to the identification of the root cause implied prioritising within the nine possibilities. With the aid of the product development team at BCTP, the items on the Ishikawa diagram were given different levels of priority with the goal of fulfilling the following objectives in order:

- Guarantee correct operation of testing equipment to ensure accurate results;
- Verify functional parameters of the SV and overall system functionality;
- Deeper studies of other alternatives from least to most time consuming.

This way, the quicker verifications are made in the beginning, paving the way to attaining a better understanding of the system and the testing facilities, which enables more complex studies.

The resulting final order of investigation is shown in Table 9, alongside a brief description of the possible root cause. Nevertheless, each entry will be expanded further in chapter 6.

Table 9: Evaluation Matrix of the possible root causes present on the Ishikawa diagram.

No.	Category	Priority	Parameter	Comments
1	Person	1	Improper operation of test bench	Previous team might be operating the test with incorrect setup or parameters, causing the system to behave abnormally.
11	Machine	2	Valve SLR mode failure	Something might be causing the SV to not regulate in SLR mode, which would result in the opening of the pump's pressure relief safety valve.
12	Machine	3	Valve mechanism seized	Limited freedom of movement of the valve's mechanism may be limiting SLR regulation.
61	Environment	4	Varying oil viscosity	Oil viscosity varies with temperature. If the customer tested the system at a lower temperature, the higher viscosity of the oil might have compromised SLR calibration.
13	Machine	5	Pump regulation mechanism seized	Irregular movement of the pump's regulation mechanism may be preventing regulation near the minimum flow mode.
62	Environment	6	Coil resistance fluctuation	Electrical resistance changes with temperature, according to Ohm's law. Maintaining voltage, a resistance change implies a change in current, which could impact the solenoid's behaviour.
23	Materials	7	Oil leakage between SV body and housing	Excessive leakage of oil between the SV body and pump housing may cause instability.
14	Machine	8	Valve resonance	Valve may go into resonance below the typically applicable frequency limit, causing oscillation on the mechanism.
15	Machine	9	Pulsating pump outlet flow	Vane pumps' output flow might pulsate, causing oscillations in output pressure.

6. TESTS AND RESULTS ANALYSIS

In this chapter, each possible root cause is described in the following structure:

- Explanation of why it is a possible root cause;
- Description of the tests performed to confirm/rule out the root cause in question;
- Analysis of test results.

Test viscosity, unless otherwise indicated, is of 4.5 cSt, as per the solenoid valve requirements stated on the customer drawing.

Sample size used in each study was of 50 (fifty) parts, unless otherwise stated. For ease of reading, the presented data does not show results for all of the parts when they can be considered analogous.

6.1. Improper Operation of Test Bench

Although Bontaz is equipped with pump test benches (Figure 55), these are typically only employed when a customer has a more specific request and ships the development team at Bontaz a pump to test the solenoid valve on.

For this reason, as the bench is of less frequent usage than the prototype test benches (Figure 50), the team at BCUS is not as experienced in its operation, and might have been using incorrect parameters or a test configuration file which may, for example, have caused the pump to behave in an unusual manner. To rule out this possibility, the test's configuration file made by the BCUS team was passed on to BCTP to be verified and remade, matching the slightly distinct test bench program.

The time parameter had to be changed from hundredths of seconds to seconds, to match BCTP's bench software, and the number of steps in the test file was simplified without altering the actual test sequence.

The final test file elaborated at BCTP can be interpreted in Figure 66.

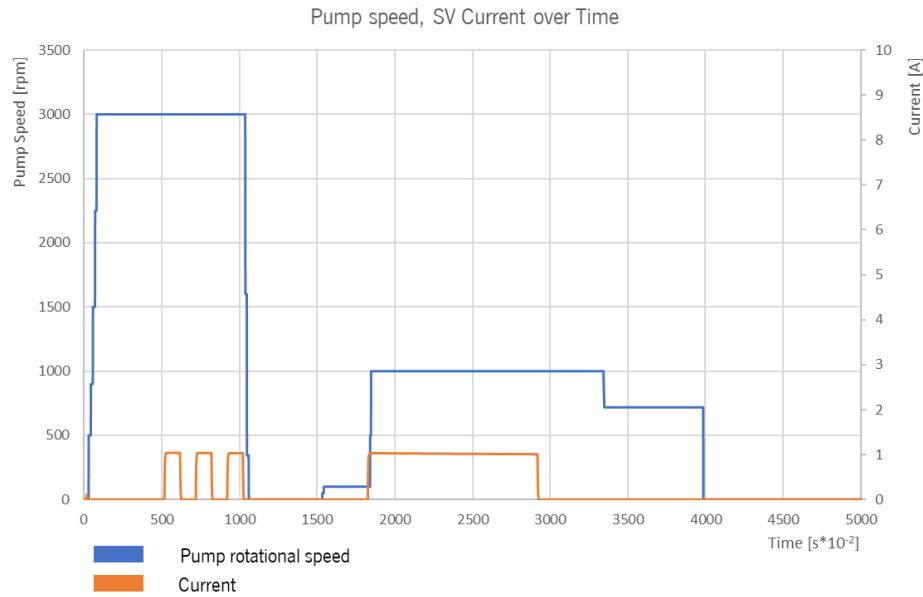


Figure 66: Graphic of pump speed and solenoid valve current over time using the provided test file.

In this test file, the vanne 2 is set to 0% (fully closed), which is unusual, as it is meant to correlate to engine backpressure. Having vanne 2 fully closed represents a completely clogged engine downstream of the main oil gallery. As this is the file with which correlation was established with the customer, this value was respected, nonetheless.

Confirmation runs were tested on the pump bench with the final configuration file, which provided the test curves shown in Figure 67:

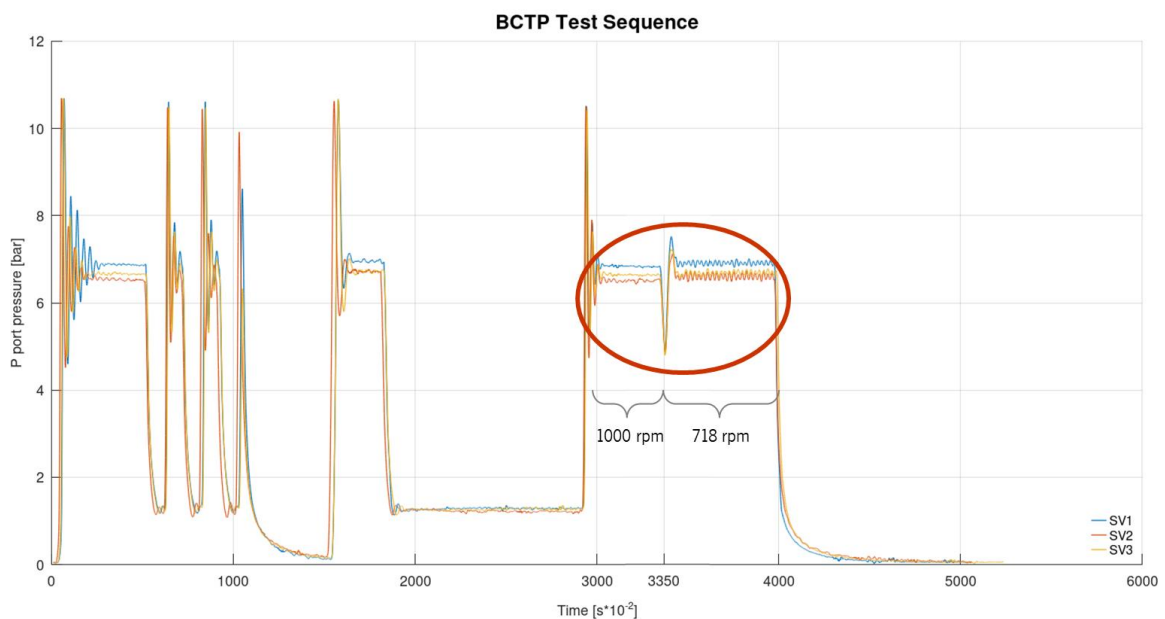


Figure 67: BCTP test curve recreation graph. Previously defined problematic oscillation zone circled, with pump rotation speeds identified.

There are clear differences between the data gathered from the test at BCTP and the graph from BCUS (Figure 65). These are mainly a consequence of the different sensors which are used on the test stands. BCTP data acquisition has a 100 Hz frequency (0.01 s period), whereas the BCUS bench only records data every 0.02 s (50 Hz). This means that the BCPT data can record faster fluctuations in the pressure values, capturing the spikes caused by sudden pump acceleration or solenoid valve closing (pump deregulation). Nonetheless, the maximum oscillation magnitude detected in this test was approximately 0.2 bar, as found in the BCUS test, and the mean pressure in this portion of the test is around 7 bar. Additionally, the increase in pressure oscillation which had been previously observed in Figure 65 can be located after the sudden drop at $(3350 \cdot 10^{-2})$ s and is much more noticeable in Figure 67. The pressure drop and variation in oscillation frequency was found to be caused by a sudden change in pump rotation speed, which is programmed to decrease from 1000 rpm to 718 rpm at this instant.

6.2. Valve SLR Mode Failure

Although every Bontaz solenoid valve is functionally checked with oil at the end of the production line, something might have happened afterward (impacts, mishandling, etc.) which may have caused the SLR regulation to fail. If this is the case, the output pressure of the pump might be limited by the pump's own pressure relief valve, instead of its regulation mechanism, for example.

In order to check the valves' functional requirements (SLR, P-A flow and A-T flow), the three test procedures explained in 4.4.2, 4.4.3, and 4.4.4 were run on every valve which was tested along the present work.

All of the tested valves exhibited good results (within customer requirements), with stable curves analogous to those presented in Figure 61, Figure 62, and Figure 63.

For this reason, SLR mode failure is discarded as the root cause of the oscillation.

6.3. Valve Mechanism Seized

If contact between mobile components, namely the piston and body, involves any burrs or particle contamination, the movement of the piston along its intended length may be partially or totally impaired, causing it to have a choppy movement or to not move at all.

Additionally, from previous experience at Bontaz, it is known that excessively high surface roughness on the piston could result in an increase in friction to a point where its advance upon activation could be unstable. For this reason, a maximum surface roughness (Ra) is specified on the internal drawing of this part (Figure 68).

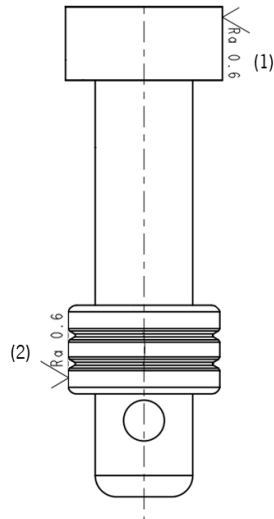


Figure 68: Specified maximum roughness (Ra) values on the piston's outer surfaces.

In order to assess this, a request was made to BCTP's Mechanics team to make pistons with high roughness. In parallel, a few lower roughness samples were tested for comparison. From the piston's drawing (Figure 68), it can be seen that the maximum surface roughness specified is 0.6 mm (Ra). Roughness measurements were obtained using the profilometer in Figure 69.



Figure 69: Mitutoyo Contracer SV-C-3200 profilometer.

Seven valves were then assembled and calibrated using these pistons.

To check if this is the cause of instability, one can graph the test data from the P-A flow test and analyse the evolution of the flow as the current increases. If the piston movement is smooth – as intended – the curve will not present any visible steps and, instead, follow a smooth line. Should there be something hindering piston movement, the curve will show that some values of flow might stay constant as the

current increases. This means that the solenoid is forcing its movement against an obstacle and opening as the it is cleared.

The obtained data graphics can be seen in Figure 70.

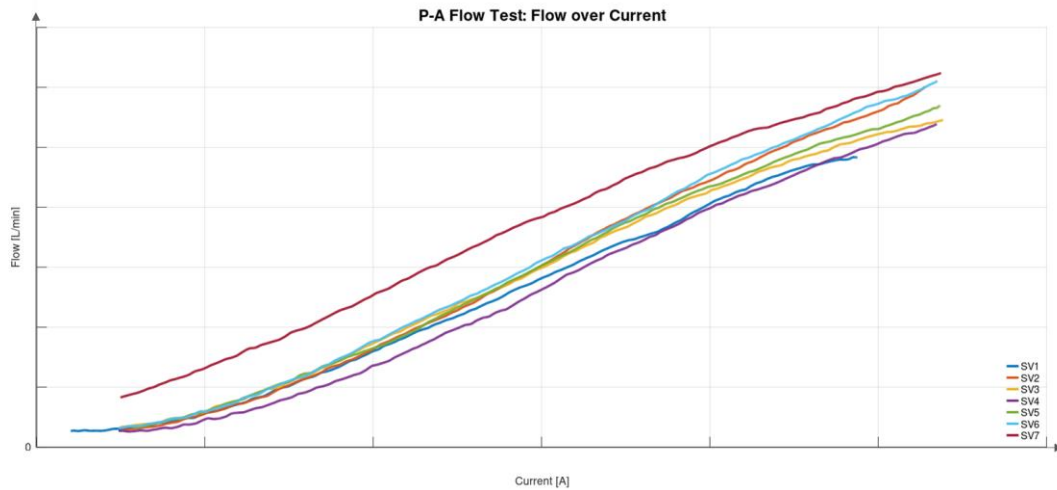


Figure 70: Flow over Current curves from the P-A Flow tests of valves built using the measured pistons.

The smooth evolution of the curve and the absence of any steps shows that piston movement is smooth throughout the whole current range, proving that hindered movement of the piston due to surface roughness is not the cause for the observed instability. As for the other pre-built solenoid valves, the result of this test was analogous, as can be seen in Figure 71.

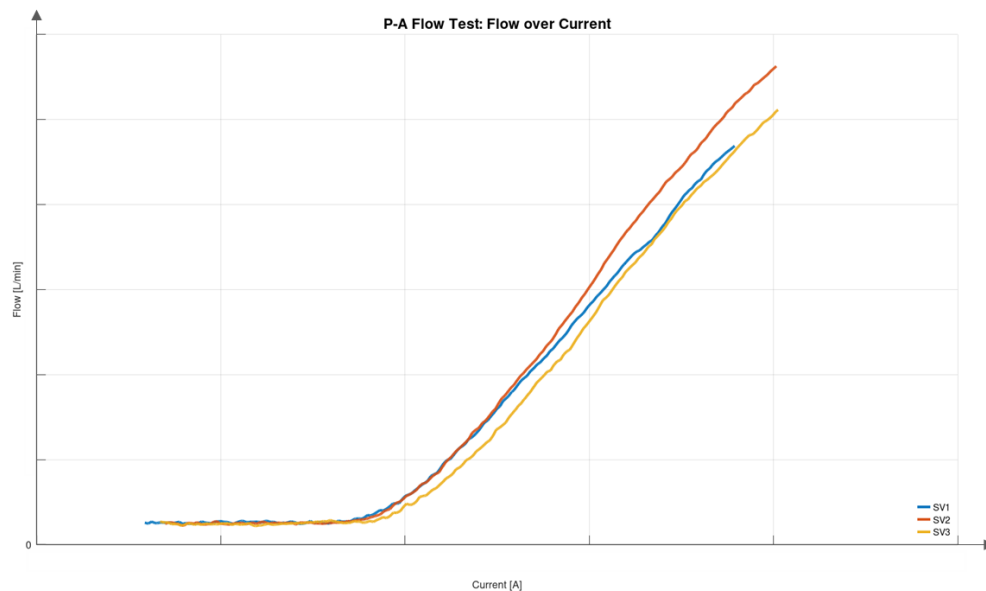


Figure 71: Flow over current curves extracted from the P-A Flow tests of valves 1, 2, and 3.

For this reason, it can be said that the valve mechanism is working as intended, providing smooth movement of the piston throughout its functional range, excluding seizing of the mechanism as a possible root cause.

6.4. Varying Oil Viscosity

The solenoid valve's functional requirements are specified at a kinematic viscosity value of 4.5 cSt, as per the customer drawing. As such, the verifications done in the course of this study were conducted with oil at this value of viscosity, including the pump tests. However, engine oil during the final customer's EoL test might have a varying level of viscosity – which could make it so that the SLR regulation is no longer set at the desired pressure, changing the regulation system's characteristics.

In order to verify this, the SLR calibration was measured once again, as done previously in 6.2, but now using the oil with the maximum viscosity available for testing at BCTP at this time - 0W16 engine oil at ambient temperature (25 °C) - which meant a kinematic viscosity of around 68 cSt, more than 15 times greater than the previous tested value.

The data obtained from these tests showed that the SLR calibration pressure stayed the same despite the great increase in viscosity, which indicates that the valve continues to function in accordance with the customer's requirements in order to regulate the pump properly.

6.5. Pump Regulation Mechanism Seized

The same thought process that applied in 6.3 can be adapted to the pump's regulation mechanism itself. Even if the valve mechanism is moving as intended, an impairment in the regulation of the pump can cause it to not regulate at all, or to show instability at certain stages of regulation.

In order to test this, obviously, the pump has to be put into scenarios where it would need to regulate. This can be achieved, for example, by:

- Without electrical activation, increasing pump rotation speed until SLR regulation occurs;
- Assessing pump output flow at a constant rotation speed and progressively activating the solenoid valve.

6.5.1. Failsafe Mode (Without Electrical Activation)

This situation exemplifies how the solenoid valve and the pump's regulation mechanism would prevent overpressure by reducing output flow per revolution.

The test parameters consisted of a speed ramp from 0 rpm to 3000 rpm and back to 0 rpm over a period of 30 seconds total (15 seconds of increasing speed, 15 seconds of decreasing speed). Valve openings were set at 80 % and 60 % for vanes 1 and 2, respectively, as previously explained in 4.3.2.2.

The flow over time curves can be observed in Figure 72:

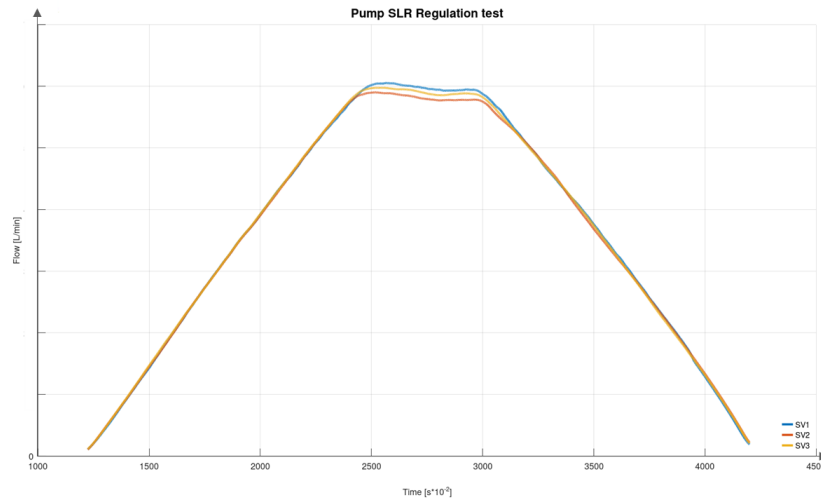


Figure 72: Pump SLR Regulation test curves.

As shown, the pump regulates similarly across the tested solenoid valves, with flow stabilising at around 60 L/min for the set restriction. This means that, with the failsafe mode, the pump is successfully self-regulating, impeding an excessive output of oil flow.

In parallel, pressure measurement at the main oil gallery appeared to be quite stable, matching what can be seen in Figure 72. Pressure during this test can be seen in Figure 73:

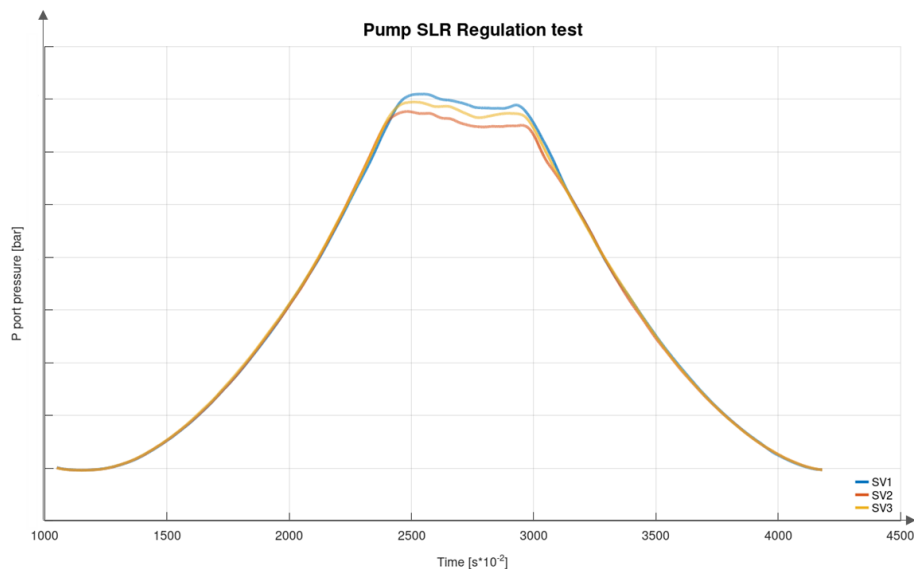


Figure 73: Pressure at the entrance of the solenoid valves during the pump SLR test.

Pressure achieved during maximum flow mode is around 7 bar, matching what was achieved in Figure 65 and Figure 67.

6.5.2. Electrical Activation

This test serves to exemplify what should be the operation of the pump within an engine, when the ECU commands activation and deactivation of the solenoid valve in response to varying oil flow requirements, while the pump is rotating at a constant speed.

In this test sequence, the pump was accelerated to 3000 rpm and kept at this speed for 20 seconds. During this period, a current ramp – from 0 A (off) to 1 A (on) and back to 0 V – was supplied to the solenoid, providing a cycle of progressive activation (for 10 seconds) and the following progressive deactivation (also for 10 seconds).

As such, the pump should respond by regulating accordingly. The test result can be found in Figure 74.

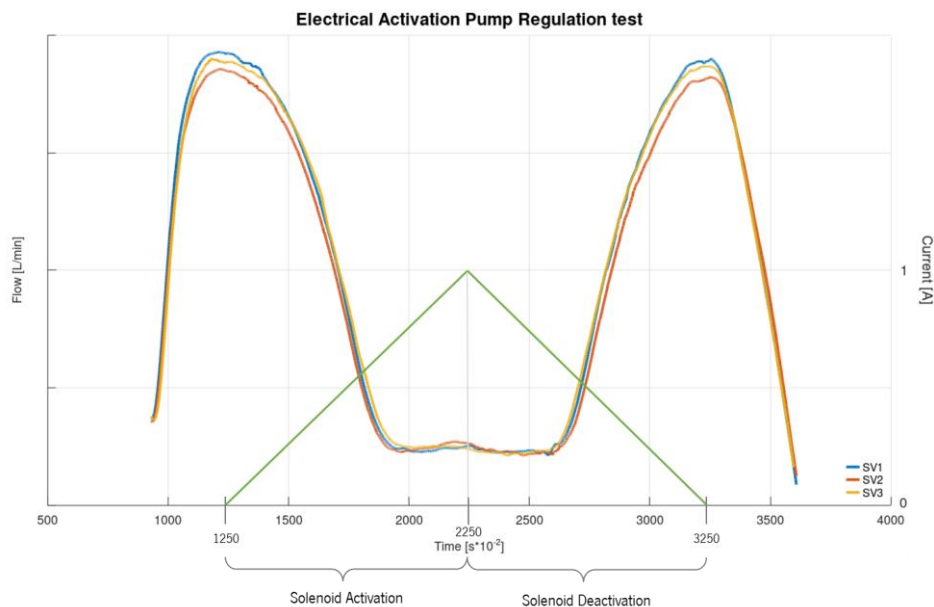


Figure 74: Electrical solenoid activation pump regulation test curves. Current supply to the solenoid valves shown in green.

The output flow of the pump can be seen decreasing as the solenoid valve is progressively activated, meaning that the regulation mechanism is moving accordingly. After a certain point of activation, the flow reaches its minimum value, where the pump stabilises, showing that the regulation mechanism cannot rotate any further, despite the further opening of the solenoid valve.

When the current starts to decrease back to 0 A, as expected, the opposite happens, with the pump's flow increasing as the regulation mechanism recedes. The graphic is approximately symmetrical, showing a smooth, predictable behaviour.

Alongside the SLR regulation test, this result shows that the pump regulation mechanism is able to rotate and stabilise, which indicates that this is not the cause of the instability found.

6.6. Coil Resistance Fluctuation due to Temperature

As seen in Figure 74, the regulation of the pump's output flow depends on the current flowing through the solenoid. In 3.4, it was explained that the resistance of a conductor is constant at a given temperature, which means that the current flowing through a circuit is directly proportional to the applied voltage. However, for a varying temperature, the resistance value also changes. Due to Joule heating of the solenoid, if kept active, its temperature increases with time, increasing the resistance. Therefore, maintaining the solenoid valve at 12 V for an extended period would result in different current values in the beginning and at the end of that period – higher current in the beginning than at the end.

As shown in equation 1, the strength of the magnetic field provided by the solenoid changes based on the current flowing through it. The intention with this study was to find whether different temperatures and, consequently, different currents, could impact the system's behaviour.

Nonetheless, as it was found that the portion of the test where the problematic oscillation takes place is during failsafe mode (solenoid off), this possibility was discarded as a possible root cause, since the valve is kept at 0 A, making any temperature changes indifferent from an electrical point of view.

6.7. Oil Leakage Between SV Body and Housing

The tolerances defined for the diameter of the body of the solenoid valve are well established throughout Bontaz practice in order to keep leakage within the acceptable range and guarantee the correct operation of the valve. However, leaks between the housing and the solenoid valve's body are inevitable, given the respective tolerance, which leave certain clearance around the diameter.

If excessively leakage between the body of the solenoid valve and the housing occurs, the supply of oil to the regulation chamber might become insufficient to achieve the established 1 bar regulation pressure, causing the SLR calibration to happen at a higher P port pressure than intended. This occurrence could be putting the pump into an unstable regime.

The possible leakages between the solenoid valve's body and housing are exemplified in Figure 75.

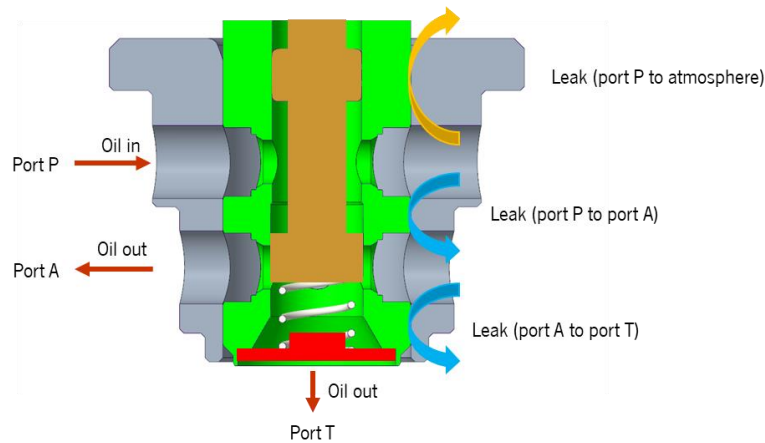


Figure 75: Section view of the solenoid valve inside its housing. Orange arrows represent in- or outflow of oil via the valve's ports, whereas blue and yellow arrows represent the possible leaks.

Every valve is susceptible to minor leaks such as those represented in blue, however, some Bontaz SVs which must not leak to the exterior at all, like the one shown in Figure 36, have an o-ring preventing the path illustrated in yellow. This possibility was studied to rule out the possibility that excessive leakage may be causing instability in the system.

For this, three bodies were turned on a manual lathe to make grooves for the o-rings, whose dimension (Figure 76) was based on other tested and proven solenoid valve designs. The reworked bodies are shown in Figure 77.

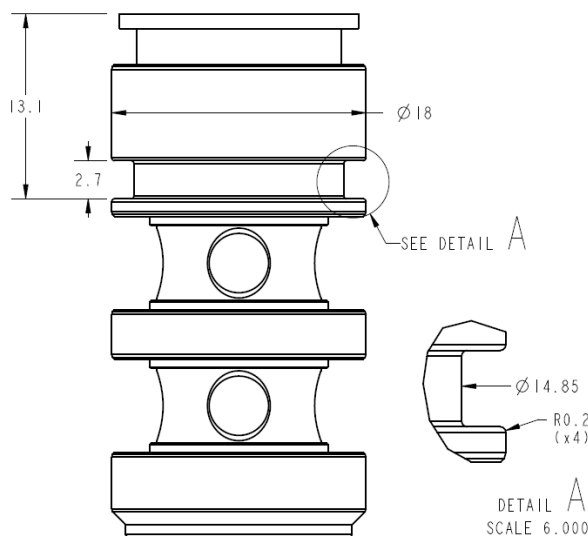


Figure 76: Drawing of the groove for the o-ring on the valve's body.



Figure 77: From left to right: normal body; reworked body with groove; reworked body with groove and o-ring.

Each body was assembled into a solenoid valve, which was then calibrated and tested on the pump, providing the following results:

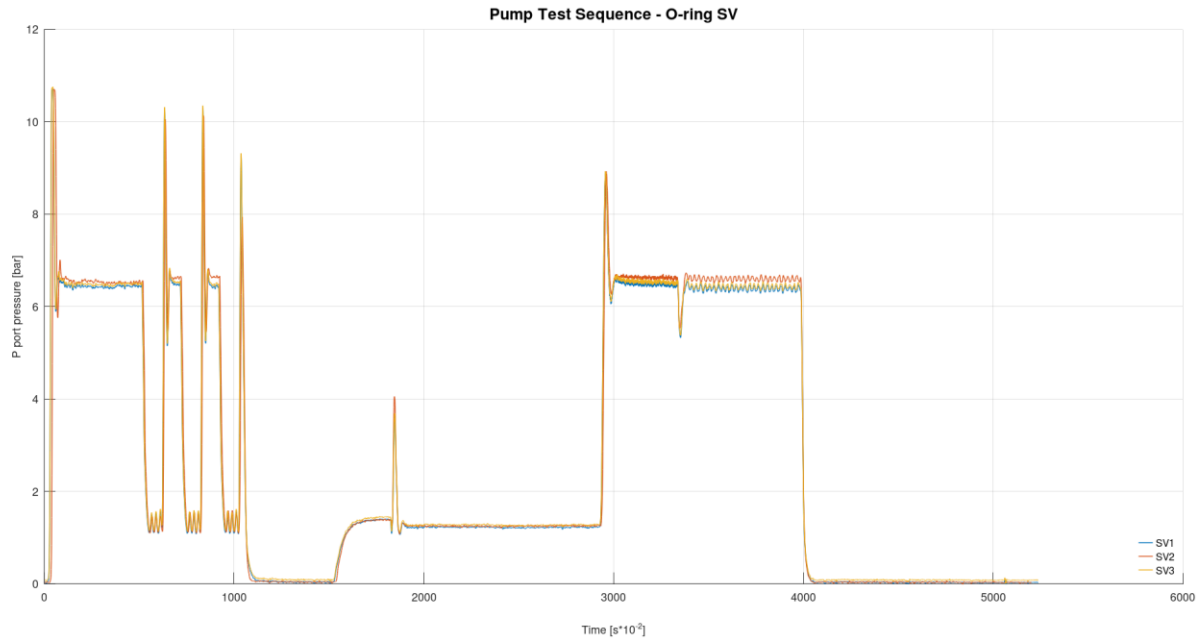


Figure 78: Pump test curve obtained from the three valves with o-rings.

Once again, the oscillation found matches that of the BCUS graph in Figure 65 and Figure 67, with a magnitude of around 0.2 bar. For this reason, the excessive leakage was discarded as a plausible root cause.

6.8. Valve Resonance

Given that the oscillation maintains a visually perceptible constant frequency which changes with the pump's rotation speed, it is possible that the SV could be exhibiting resonant behaviour in this specific part of the test.

The valves in study are tested for resonance below 300 Hz during their development, as per customer requirement. This value is defined as being higher than typical oscillation frequencies which could be caused by the engine.

Bontaz is equipped with a shaker, which is a vibrating platform on which the parts can be placed on a validation fixture, where a monitoring accelerometer is placed.

The fixture used for this test is shown in Figure 79. Because the oscillation problem happens with the valve placed horizontally in the pump housing, vibration tests were run with the valves placed in this same position (see Figure 80).



Figure 79: Fixture used for vibration test.

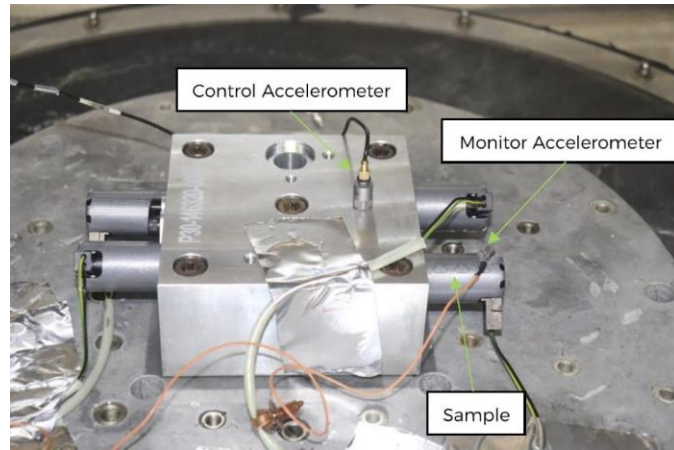


Figure 80: Fixture with solenoid valves placed on the shaker. Sample - SV being tested; Control and Monitor accelerometers – sensors for fixture and SV acceleration measurement, respectively.

The test procedure involves placing a control accelerometer in the fixture, and another, monitoring accelerometer on each valve at a time. The control accelerometer reads the fixed acceleration at the fixture, caused by the shaker. The monitor accelerometer reads the acceleration of the solenoid valve. When a difference between accelerometer readings is found, the valve is vibrating in relation to the fixture. When this vibration rises over a set limit, the valve is considered to be resonating (see Figure 81).

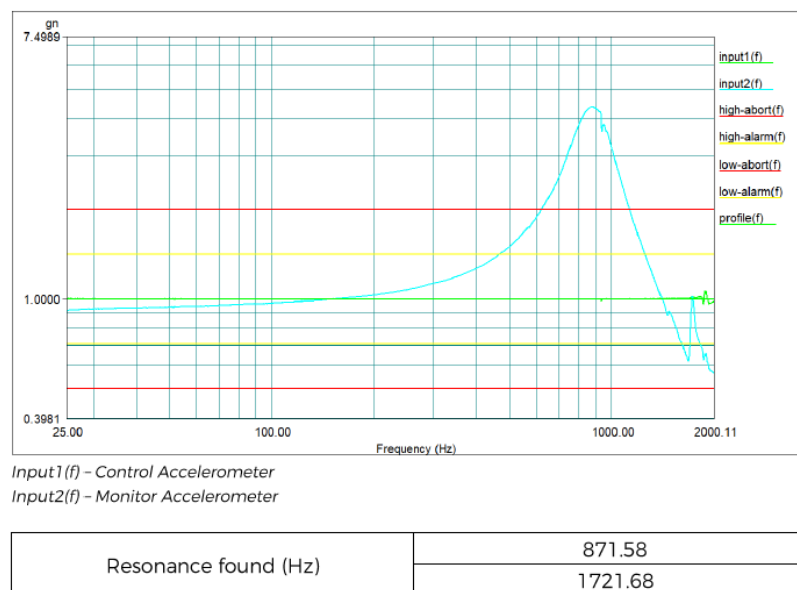


Figure 81: Vibration test results from one of the four tested solenoid valves. Green line represents fixture acceleration, blue line shows solenoid valve acceleration. Results were analogous on the remaining three SV.

The indicated values are correspondent to the peak relative acceleration between the fixture and the valve, at a frequency of 871.58 Hz and 1721.68 Hz. Both of these values are far above the established 300 Hz for all tested valves, so these do not exhibit resonance below the set limit, meaning that this factor should not be the cause for the oil pressure oscillation.

6.9. Pulsating Pump Output Flow

From the research done in 2.2.2, it is known that, even though positive displacement rotary pumps cause less pulsation than their reciprocating counterparts, their output flow still pulses due to the manner in which the pump operates.

6.9.1. Reasoning for Possible Root Cause

In a viscous flow along a flow path, there is an intrinsic relation between flowrate and pressure drop. The following explanation provides an example of this relationship.

According to Bernoulli's Principle (equation 5), the energy of the fluid flowing between two points in a pipe with fixed characteristics (as represented in Figure 82) will decrease by a certain amount, denominated pressure drop, or ΔP , due to friction losses.

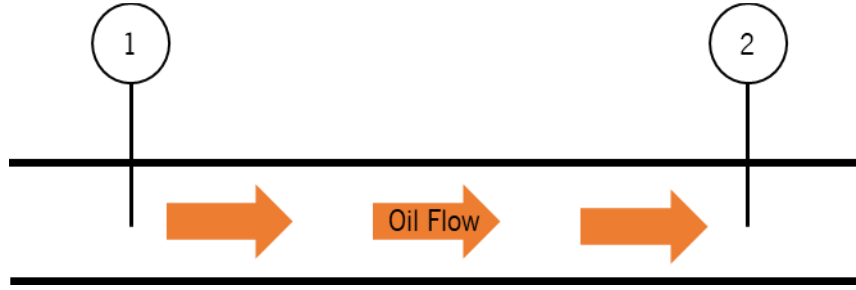


Figure 82: Schematic representation of oil flow through a straight pipe with constant cross-sectional area and internal surface roughness. Points 1 and 2 serve as references for equation 5. Point 1 is located upstream of Point 2.

$$p_1 + \frac{1}{2}\rho v_1^2 + \rho g h_1 = p_2 + \frac{1}{2}\rho v_2^2 + \rho g h_2 + \Delta p_{1-2} \quad 5$$

where:

- P_x is the pressure at point x ;
- ρ is the fluid's density;
- v is the fluid's velocity;
- g is the Earth's gravity;
- h is the height of the fluid.

For a pipe like this, ΔP between points 1 and 2 is provided by the Darcy-Weisbach formula, stated in equation 6:

$$\Delta P_{1-2} = f_D * \frac{L}{D_p} * \frac{v^2}{2} * \rho \quad 6$$

The Darcy friction factor, f_D , is dependent on both fluid and pipe geometry, so it is constant if these are unchanged. D_p and g are maintained for the same pipe as well [69].

It is, then, possible to conclude that for the same pipe and fluid, Δp_{1-2} varies only with fluid velocity, which is a function of the flow rate, Q , and cross-sectional area of the pipe, A , the latter being constant.

$$v = \frac{Q}{A_{sec}} \quad 7$$

Considering $p_2 = 0$ bar (atmospheric pressure), then $p_1 = \Delta p_{1-2}$, meaning that pressure upstream varies in function of the flow rate.

From these notions, it is concluded that pulsation in the pump's output flow rate will theoretically result in a pulsation of the outlet pressure value, therefore, this could be the cause of the pressure oscillation. This in turn implies that pressure must also pulsate with the flow pulsation at the outlet of the pump, which feeds the solenoid valve at port P after passing through vanne 1. An outlet pressure oscillation could, therefore, trigger a cyclic instability, where the outlet pressure oscillation causes the input pressure at the solenoid valve to oscillate as well, thus compromising SLR mode stability.

6.9.2. Investigation through Simulation Model

To study this hypothesis, a simulation model of the pump had to be developed using the Amesim software. The goal of this test was to obtain a functioning theoretical system which would help evaluate the oscillation at the outlet of the pump at different speeds and stator eccentricities, and not necessarily to perfectly represent the object of study, as such an investigation would require very extensive know-how on the pump and its simulation.

6.9.2.1. Test bench Hydraulic Characterisation

As a tool to aid in setting up the simulation model and test bench if necessary, a more detailed version of the hydraulic diagram represented in Figure 54, now including the solenoid valve, was elaborated using the Fluidsim Hydraulic software (Figure 83):

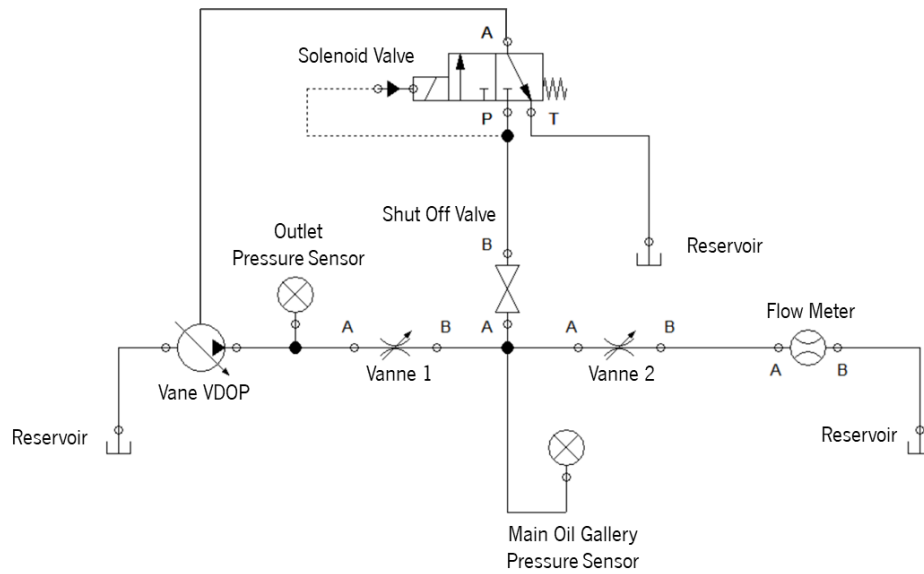


Figure 83: Hydraulic diagram of the pump bench and solenoid valve.

From the analysis of this diagram, it is possible to conclude that, in normal operation, the flow meter shown does not receive the entirety of the oil pumped, because a portion of it goes to the solenoid valve and fills the regulation chamber, eventually making its way back to the reservoir. Moreover, when vanne 2 is fully closed, as said in 6.1, no oil reaches the flow meter at all. Instead, all of the oil goes to the solenoid valve, so it is not possible to measure pump output with this setup.

6.9.2.2. Dimensional Characterisation of the Pump

The software includes a virtual vane pump model which needs to be parameterised, by providing some important dimensions, which could be retrieved from the CAD model of the pump.

- Number of vanes (8);
- Vane depth (height of the pump chambers);
- Rotor outside diameter;
- Stator inside diameter;
- Maximum angle of rotation of the stator.

6.9.2.3. Hydraulic Characterisation of the Pump

In order to establish a correlation between the simulation model and the real pump, a baseline reference for pump behaviour across a range of speeds is necessary. To guarantee that the stator maintains its position across the different tests, regulation was disabled completely, by closing the shut off valve shown

in Figure 83. The pump was then tested at several different speeds with the standard restriction only in vanne 1 (80 %), providing the values shown in Table 10:

Table 10: Unregulated pump flow per revolution values obtained in bench testing. Cells highlighted in yellow correspond to the pump speeds where oscillation has been found in the EoL test.

ω_{pump} [rpm]	Q [L/min]	Outlet pressure [bar]	Q/ω_{pump} [L/rev]
200	5.5	0,00	0.0275
400	11.1	0.16	0.02775
600	16.4	0.25	0.0273
718	19.8	0.34	0.0276
1000	27.3	0.6	0.0273
1500	41.2	1.23	0.0275
2000	55,0	2.1	0.0275
2500	67.7	3.15	0.0271
3000	81.0	4.2	0.0270
Average flow per revolution:			0.0274

As expected, without regulation the flow per revolution remains approximately constant even as pressure increases at the outlet.

On the other hand, for measuring output flow during the oscillation test, an additional flow meter was placed between the shut off valve and the SV, such as illustrated in the update hydraulic diagram (Figure 84).

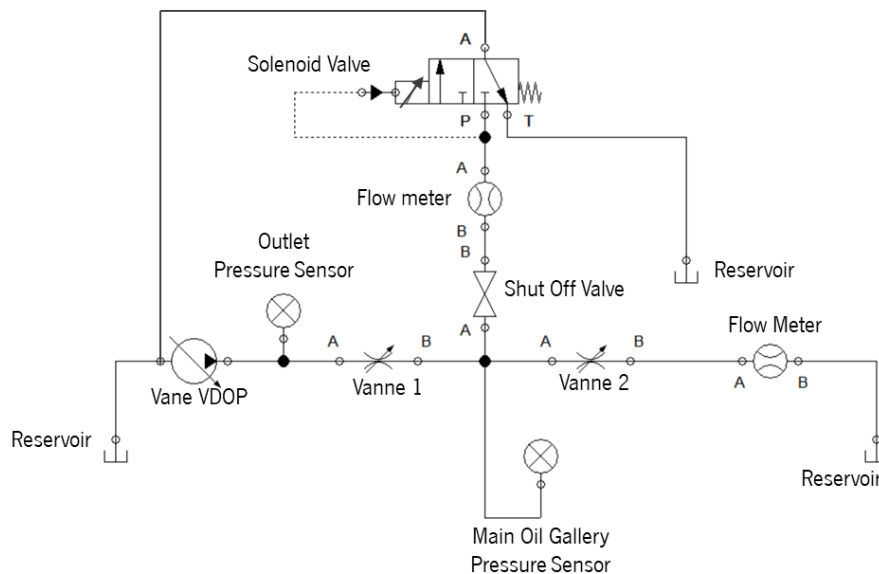


Figure 84: Updated hydraulic diagram of the pump bench and solenoid valve with additional flow meter installed.

The original oscillation test configuration was followed, with vanne 2 fully closed, meaning that all of the output flow from the pump goes to the solenoid valve. An example of obtained in this test are shown in Figure 85, with the points where flow was measured noted.

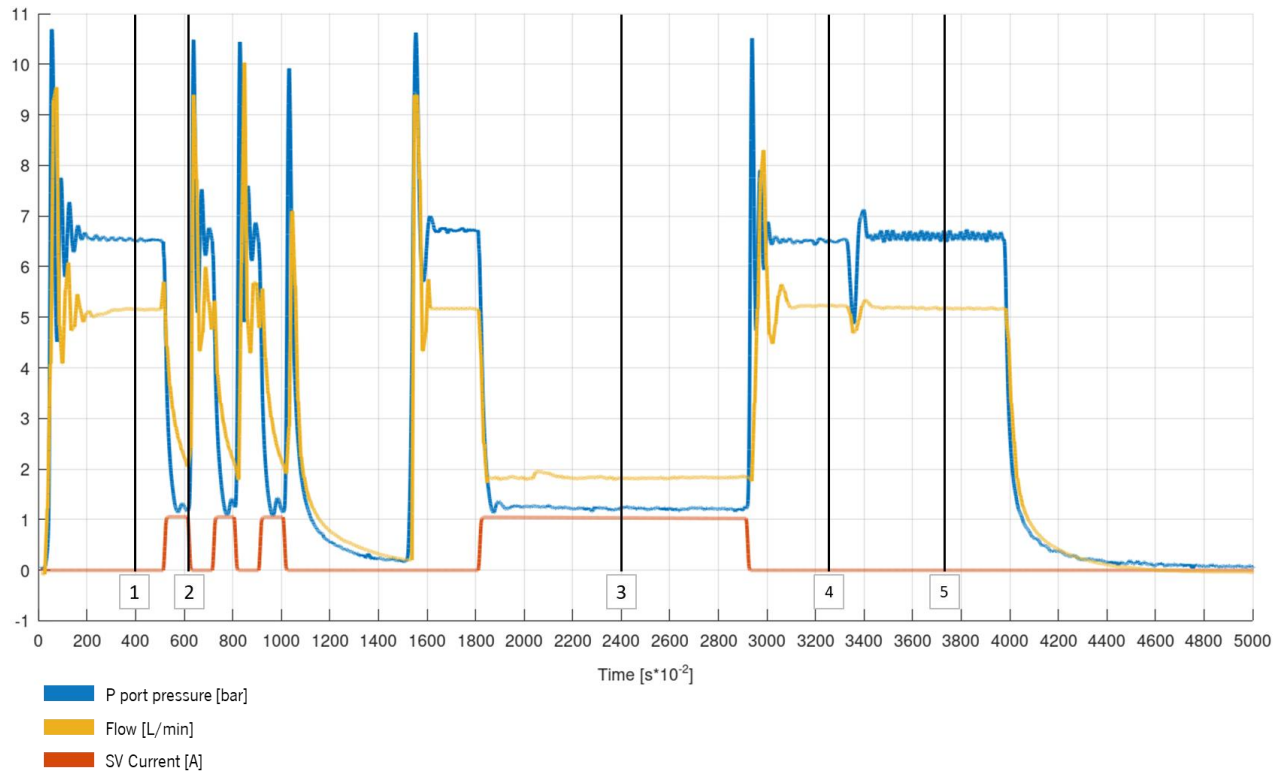


Figure 85: Example of BCTP sequence pump test curve where flow to the solenoid valve has been measured. Blue line represents pressure, orange line represents current, and yellow line represents flow. Numbers show the points where parameters were monitored.

Table 11 contains the flow rates measured during the oscillation portion of the test:

Table 11: Data gathered in each point specified in Figure 85.

Point	SV on/off	Test instant [$s \cdot 10^{-2}$]	ω_{pump} [rpm]	Pressure at port P [bar]	Q_{sv} [L/min]	Q/ω_{pump} [L/rev]
1	Off	400	3000	6.48	5.2	0.0017
2	On	624	3000	1.60	2.2	0.0007
3	On	2400	1000	1.23	1.8	0.0018
4	Off	3252	1000	6.61	5.2	0.0052
5	Off	3750	718	6.64	5.2	0.0072

This information shows that the pump is providing the same total output flow at 1000 rpm and at 718 rpm despite the speed difference, implying that the stator is in a distinct regulation angle. This is confirmed by checking the disparity in the values of flow per revolution for points 4 and 5.

6.9.2.4. Simulation Model Obtained

Having provided the data for characterisation of the pump, the simulation model was made using Amesim's vane pump template.

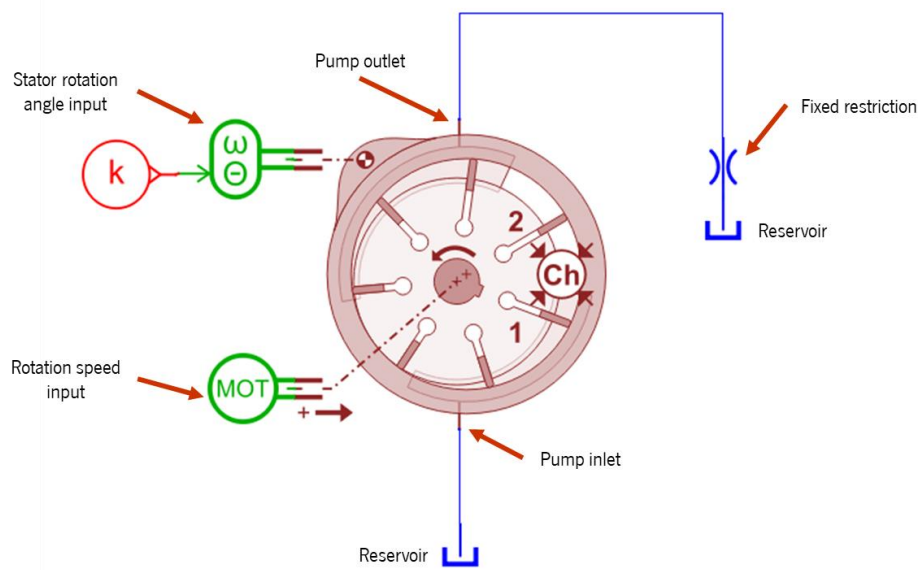


Figure 86: Virtual vane pump model hydraulic diagram made with Amesim. Inlet and outlet purely representative.

As previously explained, this model allows for a good understanding of the pump but is not 100 % accurate at representing the system. The angle of the stator is perfectly fixed and not regulated by the solenoid valve, and there is only one restriction to the output flow instead of the two vanes in the pump bench. This restriction was defined to match the flow/pressure relationship found in Table 11.

Besides this hydraulic scheme representation, Amesim provides a scheme of the pump to allow a better visualisation of the chambers' volumes throughout their rotation and the eccentricity between rotor and stator (Figure 87).

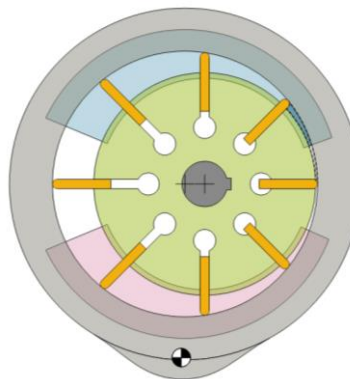


Figure 87: Amesim pump model in fully unregulated position. Inlet represented in blue, outlet shown in red. Shaft rotates counterclockwise.

Having defined the pump model, behaviour during the part of the test which exhibits the greatest magnitude of oscillation (point 5 in Figure 85) was mimicked in order to verify whether the flow output from the pump shows this - it was necessary to find the position of the stator during this phase, portrayed in Figure 88.

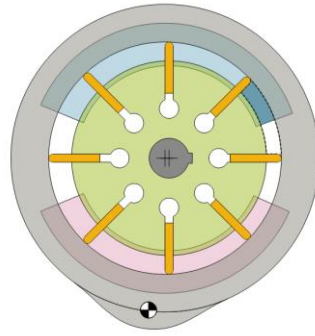


Figure 88: Amesim pump model with stator regulated to 4.8° rotation. Inlet represented in blue, outlet represented in red. Shaft rotates counterclockwise.

This angle of 4.8° was defined through an iterative process, applying angles between 0° and 8.4° (maximum angular regulation), until average output flow approximately matched that of point 5 (5.2 L/min). The output of the pump, in this position, provided by the simulation software is presented in Figure 89.

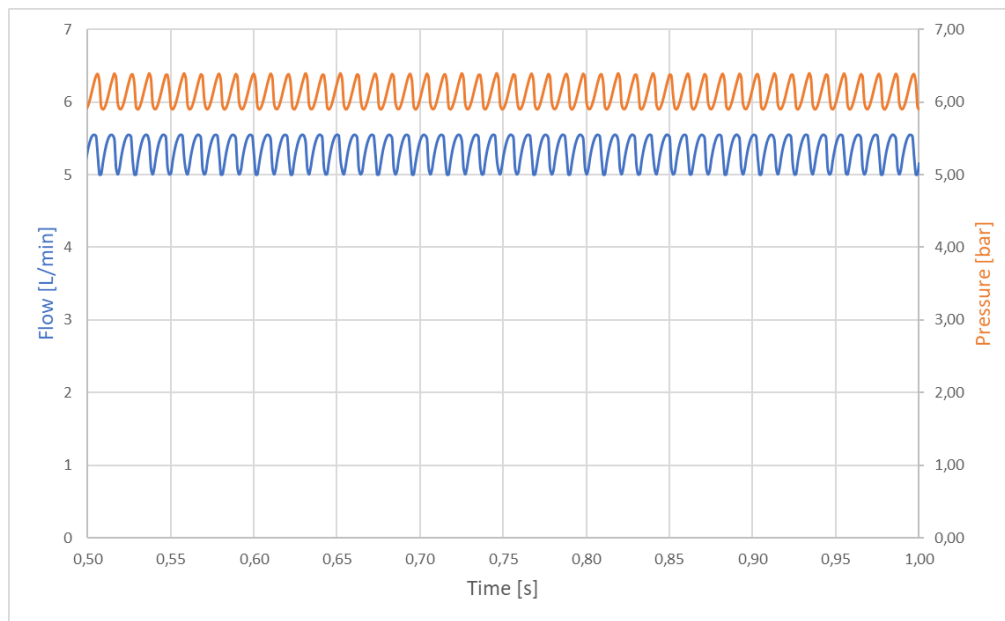


Figure 89: Regulated (4.8°) pump output flow and pressure at 718 rpm provided by the Amesim model.

It is very clear from the analysis of this graphic that the software is capturing the theoretical output oscillation characteristic of this kind of pump.

As a way to comprehend how pulsation might vary according to the stator's position, another simulation was run. However, this time, the stator was left in its unregulated position (Figure 87), and rotation speed was adjusted until the average output of the pump was approximately that of Figure 89 – 177 rpm. The output of the pump at this speed (with the same restriction applied) is shown in Figure 90.

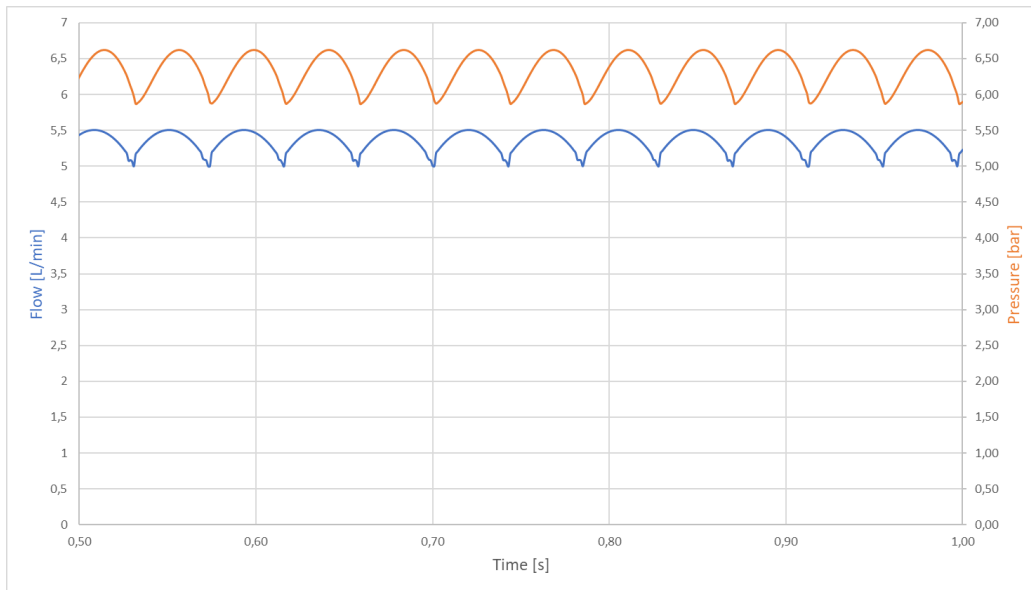


Figure 90: Unregulated pump output flow and pressure at 177 rpm provided by the Amesim model.

As the pulsations in these graphics is periodic, their respective frequencies can be estimated by counting the number of peaks during one second. A comparison of the values from Figure 89 and Figure 90 is shown in Table 12.

Table 12: Comparison between Figure 89 Figure 90 speeds, frequencies, and pressure values.

Speed [rpm]	Estimated frequency [Hz]	Pressure [bar]		
		Maximum	Minimum	Magnitude
177	23	6.62	5.87	0.75
718	95	6.39	5.90	0.50

Since the pump has eight vanes and, therefore, eight chambers in between them, for each rotation of the rotor there are eight discharges of oil. To verify the frequencies counted in these graphics, the number of discharges per second can be calculated using the pump speed in rpm, Z , per equation 8:

$$f = \frac{8Z}{60} \text{ Hz} \quad 8$$

This equation provides a discharge frequency of 23.6 Hz for 177 rpm, and 95.7 Hz for 718 rpm, confirming the estimates in Table 12.

When it comes to pressure, minimum values are approximately equal, however, the maximum and magnitude figures are greater for 177 rpm (unregulated pump) than for 718 rpm (regulated). That disparity is explained by what is happening inside each chamber in the pump.

The Amesim model gathers data regarding the evolution of chamber volume, as well as the inlet and outlet areas covered by the chamber in question during the pump's rotation. Both the scenario in Figure

87 and Figure 88 can be analysed. For the case of Figure 87, evolution of chamber characteristics is shown in Figure 91.

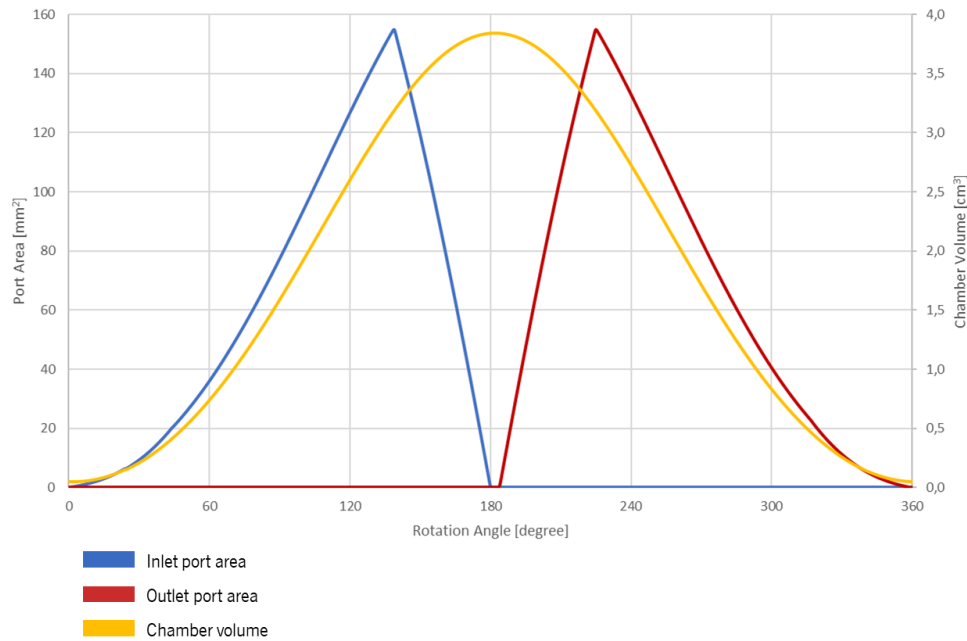


Figure 91: Evolution of chamber characteristics in function of the shaft's rotation angle in unregulated mode.

For the situation presented in Figure 88, chamber characteristics are described in Figure 92.

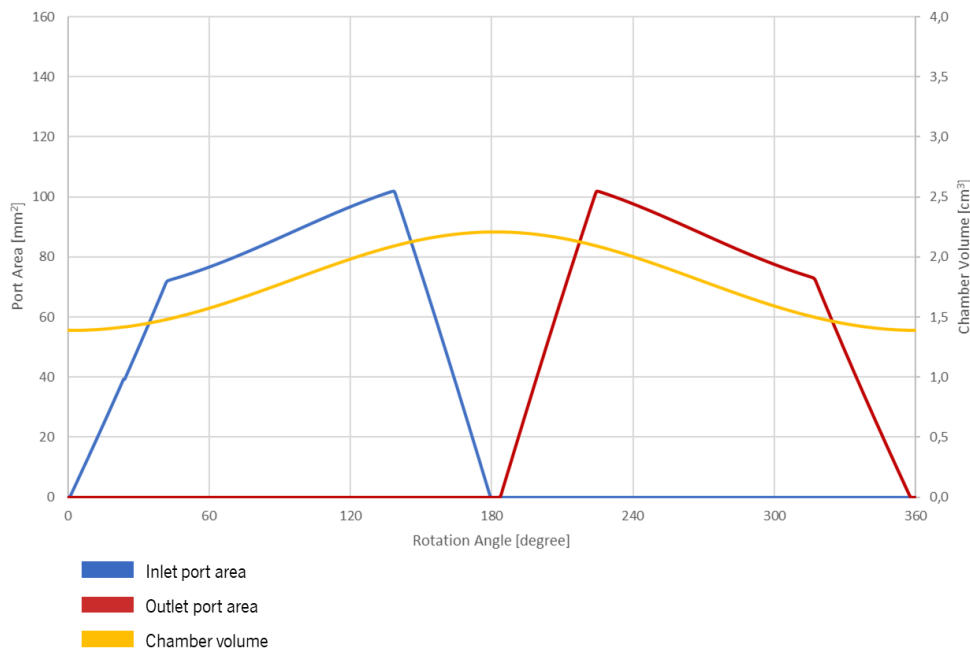


Figure 92: Volume of chamber 1 and inlet/outlet area covered in function of the rotor's angle of rotation in unregulated mode.

Some values from these graphics can be found in Table 13.

Table 13: Amesim data regarding chamber characteristics from Figure 91 and Figure 92.

Stator position	Maximum port area [mm ²]	Maximum Chamber Volume [cm ³]
Unregulated	154.95	3.84
Regulated (4.8°)	101.78	2.21

Given the overall greater area available for fluid exchange between the chambers and the ports and the greater maximum volume for each chamber, it is clear from this analysis that the unregulated pump has the capacity to output more flow than the regulated one, causing the output pressure to rise higher each time a discharge takes place, which in turn raises the magnitude of pressure oscillation.

This relationship between stator regulation and magnitude of oscillation is exemplary of what happens between the 1000 rpm and 718 rpm portions of the oscillation zone in Figure 65 and Figure 67, which show the initially obtained values, as well as in the curves shown in Figure 78 and Figure 85.

From the data in Table 11, it has been concluded that the pump is regulated further when spinning at 1000 rpm than at 718 rpm. Therefore, it is to be expected that it should show less magnitude of oscillation, which is exactly what happens in the aforementioned curves.

For these reasons, the cause of oscillation in this specific test sequence is considered to be a consequence of the natural behaviour of a positive displacement vane pump, as confirmed by the Amesim model. One further aspect to take into consideration is the fact that the pressure oscillations shown in Figure 89 and Figure 90 might have an impact in the solenoid valve's failsafe mode regulation, causing it to open and close in response to the pump's output pressure. This effect could potentially make the situation worse, by changing the pump's regulation position, furthering output flow oscillation. However, this situation was not possible to evaluate during the present work.

7. CONCLUSION

The main objective of the present dissertation was to assess the operation of a solenoid valve manufactured by the company BONTAZ – BCTP Equipamento Automóvel, used to manage an automotive variable displacement oil pump (VDOP). This system, had a disconformity, consisting of oscillation in the engine oil pump's output pressure. This work aimed at assessing the origin of this disconformity and if possible, propose some corrective measures.

A broad state of art for oil pumps and solenoid valves was done, expanding internal knowledge of these systems at Bontaz. The test bench at BCTP was set up, and testing allowed analysis of the problem seen at BCUS. Quality tools have been used to track down the root cause, and a simulation model of the pump has been created, which will allow a deeper study of the products in future development. It has not been possible to verify and replicate exactly the conditions experienced in the customer's facilities because it was not possible to establish a direct contact with the customer.

On the other hand, from the test results analysis, defects concerning bench operation, valve malfunctions, or pump inoperability have been ruled out as the root cause for the observed oscillation, as well as other functional aspects of the solenoid valve.

Subsequent studies of the system allowed for a deeper comprehension of the pump through the created simulation model. The root cause of oscillation has been identified as the natural output pulsation of the variable displacement vane oil pump, which is characteristic of positive displacement pumps.

A deeper understanding of the consequences of this oscillation and ways to avoid it in the engine will require communication with the customer. Despite that, there are possible solutions that could be assessed, such as changing the OEM test bench alarm's sensibility in order to allow for the normal pump output pulsations or equipping the circuit with an appropriately sized expansion vessel. Nevertheless, it is understood that these solutions must be validated on the customer's side and cannot be employed solely through Bontaz.

Finally, this investigation has fostered Bontaz's Product Development and Simulation teams' knowledge on solenoid valve VDOP systems, providing a stronger understanding of the functionality of developed valves and of the pumping systems they integrate.

8. FUTURE WORKS

A continuity of this investigation could be carried out in the future, provided it is possible to establish contact with the customer and get a correlation between Bontaz's results and their EoL testing, namely when it comes to explaining having the test bench's vanne 2 fully closed – a parameter established between the customer and BCUS - and how this might relate to a real engine scenario.

Additionally, reading the pump's regulation chamber pressure would probably provide more insight into the fact that the pressure oscillations could affect the solenoid valve's failsafe mode, but this would also require receiving an additional pump from the customer and their permission.

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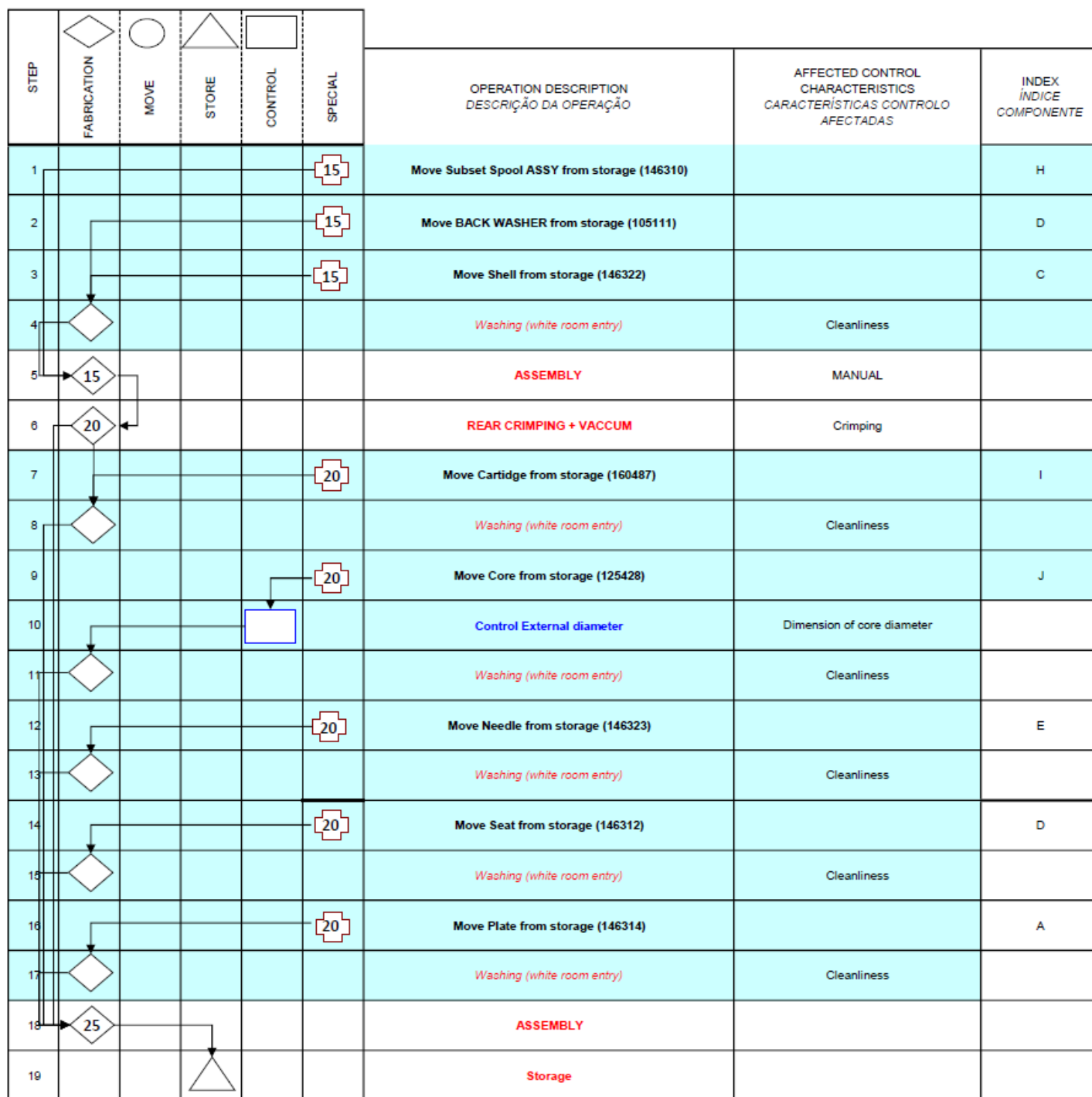
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ANNEXES

Annex 1 –146313 Electrical Subassembly Production Register Flowchart



Annex 2 – 146324 Mechanical Subassembly Production Register Flowchart

STEP	FABRICATION	MOVE	STORE	CONTROL	SPECIAL	OPERATION DESCRIPTION DESCRIÇÃO DA OPERAÇÃO	AFFECTED CONTROL CHARACTERISTICS CARACTERÍSTICAS CONTROLO AFECTADAS	INDEX ÍNDICE COMPONENTE
1				15		Move Body from storage (146321)		F
2						Holes Deburring	Burr	
4						Internal diameter control	Internal diameter of body	
3						Washing (white room entry)	Cleanliness	
5				15		Move Tiroir from storage (146329)		D
6						External diameter control	External diameter of Tiroir	
7						Washing (white room entry)	Cleanliness	
8				15		Move Spring from storage (240104)		E
9				15		Move Plug from storage (101213)		C
10						Washing (white room entry)	Cleanliness	
11	15					ASSEMBLY		
12	20					Crimping plug inside body		
13						Storage with cover		

Annex 3 – DFMEA Predicted SLR Failure Modes

Item / Function	Requirement / Specification	Potential Failure Mode	Potential Effect(s) of Failure	Potential Cause(s) / Mechanism of failure	Occur
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Wrong spring rate	3
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Port A obturated by filter overmoulded plastic	1
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Port A obturated by clogged filter	6
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Valve inlet / outlet holes not facing pump ports	2
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Piston stuck because of unsymmetrical pressure distribution	2
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Piston stuck because of closed fluid chambers	3
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Excessive spring load setting	3
FAIL SAFE MODE Switch oil pressure from port P to port A	Mechanical pressure regulation 6,5±0,5bar @ Pa=1 Bar in customer drawing	No pressure increase in port A, or unsufficient pressure	No reduction of oil pump displacement or unsufficient reduction of oil pump displacement	Slider diameters do not allow pressure to activate the valve	3

Annex 4 – Ishikawa Diagram Previously Created by BCUS Team

