**A study of the influence of lubricant viscosity on the operation and efficiency of a Continuously Variable Valve Lift hydraulic valvetrain system**

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Confidential Report

# SUMMARY

With the ever-increasing demands of reducing carbon emissions as mandated by Euro 5 and 6 emission legislation, vehicle manufacturers are exploring new techniques and technologies. The valvetrain has the most influencing parameters in reducing wasted energy in propelling an internal combustion engine. Schaeffler, a large Tier 1 automotive components supplier, have designed and manufactured an electro-hydraulic system called UniAir, that can fully control of intake valves in an internal combustion engine. Jaguar Land Rover (JLR), a medium-sized luxury vehicle manufacturer, are implementing the UniAir to their AJ200 engine family in order to comply with the emission standards while benefitting from the performance enhancing effects.

This thesis details the design, development and validation of a motored valvetrain test rig that enables the characterisation of the performance of the UniAir system as utilised by JLR. The rig forms part of a test methodology also developed during this work, that can determine the influence of the viscosity of different lubricants to understand the performance, frictional operation and efficiency of a continuously variable valvetrain (CVVT). A testing programme was carried out, measuring the valve lift profiles and torque required to operate the continuously variable valve lift (CVVL) system, while varying oil formulation and temperatures from 0°C to 120°C, and environmental temperatures varying from 0°C to 120°C using climate chambers. The results were then analysed and used to inform recommendations and future research direction for JLR.

# ACKNOWLEDGMENTS

I wish to express my sincere appreciation to my supervisor, Dr Tom Slatter, who has had the patience of guiding, motivating and encouraging me throughout the project. Without his consistent input, the goal of this project would have not been realized.

The physical and technical contribution of Oliver Cooper and Chris Todd have absolutely made the possibility of this project. I am indebted to the above and beyond approach applied which enabled the completion of the project.

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# CONTENTS

# NOMENCLATURE

| | | |
|---|---|---|
| FL | Full Lift | |
| NL | No lift | |
| EIVC | Early Intake Valve Closing | |
| LIVO | Late Intake Valve Opening | |
| VVL | Variable Valve Lift | mm |
| CVVL | Continuously Variable Valve Lift | mm |
| CVVT | Continuously Variable Valve Train | |
| CA | Crank Angle | ° |
| ECU | Engine Control Unit | |
| TDC | Top Dead Centre | |
| BDC | Bottom Dead Centre | |
| EHL | Electrohydrodynamic lubrication | |
| IC | Internal Combustion | |
| SI | Spark Ignition | |
| CI | Compression Ignition | |

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# 1. INTRODUCTION

The transport of goods and people is almost entirely (99.9%) powered by internal combustion engines (ICE/ICEs) (1). The world has approximately 1.6 billion passenger and commercial vehicles, with this number likely to increase in emerging economies such as China and India (1). Currently the transport of goods and people account for about 20% of the total primary energy consumed and around 23% of $CO_2$ emissions. In Great Britain alone, 327.1 billion miles were driven on the roads in 2017 by ICE vehicles with an increase of 1.3% from 2016 and 16.9% from 1997 (2).

The Kyoto Protocol (1998) is an initiative where the members of the United Nations are attempting to achieve a quantified emission limitation and promotion of sustainable alternatives to fossil fuels (3). In addition to this, are the European regulations of Euro 5 and Euro 6 (4). The regulation (EC) No 715/2007 explains that these are used to regulate common technical requirements of ICE motor vehicles to replace or control parts to reduce pollution and emissions (4). The ever-demanding regulations set by the European Parliament and council have financial incentives for new vehicles that comply with the regulations and penalties for infringement by manufacturers (4). These financial incentives have vehicle manufacturers striving to reduce carbon emissions and as a result, manufacturers have started to investigate areas to improve.

From the 1st September 2017 the Worldwide Harmonized Light-Duty Vehicle Test Procedure (WLTP) was introduced to replace the previous New European Driving Cycle (NEDC). This test is significantly more realistic in terms of actual driving behaviour with the goal to help compare the fuel consumption and pollutant emissions of vehicles from different manufacturers (5). To date exhaust emissions measurements have been done on dynamometers. Furthermore, the Real Driving Emissions (RDE) road test will now be used to determine pollutant emissions and inform the WLTP cycle design. This is achieved by describing a vehicle's emissions under real on-road conditions using a Portable Emissions Measurement System (PEMS) to measure nitrogen oxide and carbon monoxide emissions.

There is currently much interest in creating an alternative by using battery electric vehicles (1), but the existing transport system is built around ICEs. To change the infrastructure to entirely electric "will have extreme economic, social, environmental and political impacts"(1). In addition to this, for electrical vehicles to reduce emissions below ICEs, electricity generation to charge the vehicles needs to be decarbonized (1). As a result, manufacturers are investigating other alternatives to improving ICEs. Figure 1 shows an overwhelming amount of energy wasted in the use of gasoline to propel a vehicle. The most influencing parameters in reducing wasted energy is the valvetrain. Its 'sphere of influence' demonstrates the potential thermodynamic and friction improvements that can be made with the valvetrain.

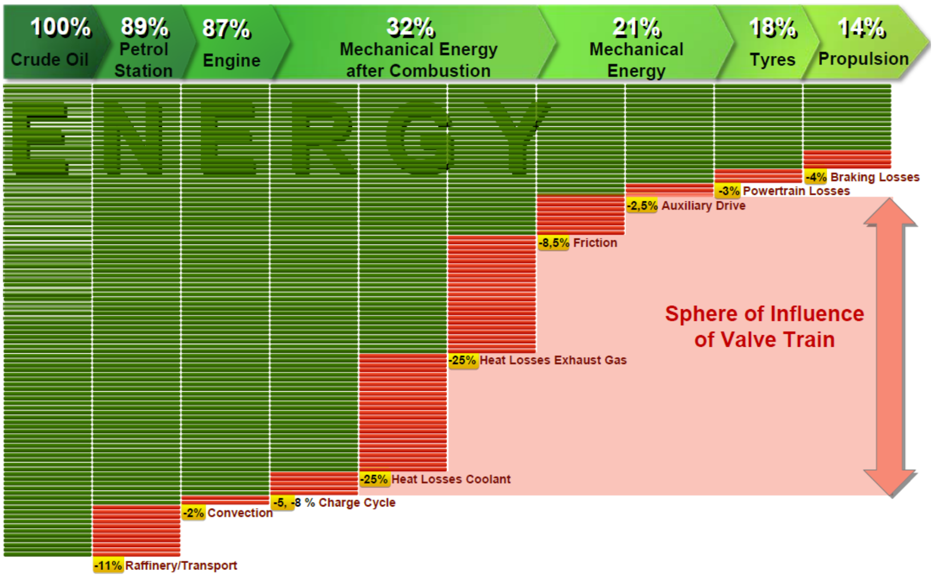


**FIGURE 1 - EFFICIENCY CHAIN IN A GASOLINE ENGINE *(6)***

The operational variables that require optimising in order to improve the valvetrain are phasing, lift and timing (7). Depending on the requirements of the engine, for optimal efficiency all three of these variables need to change which can be achieved by various different designs of variable valvetrains (VVT) (7).

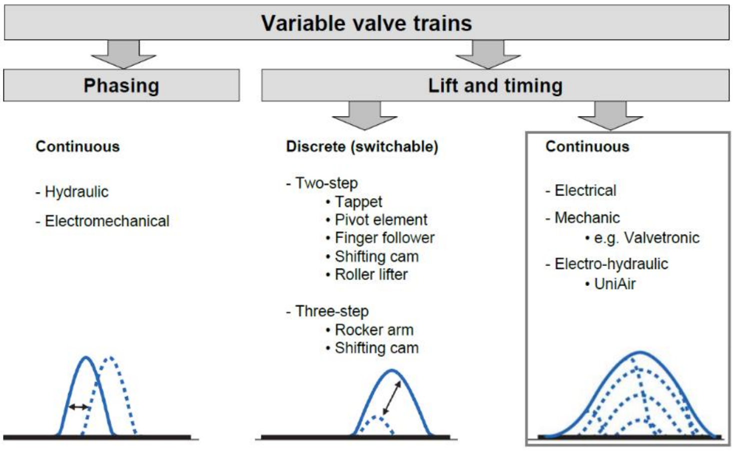


**FIGURE 2 - VARIABLE VALVETRAIN CONTROL *(8)***

Figure 3 shows the current VVTs are mechanical, electromagnetic and electro-hydraulic. The mechanical and electromagnetic VVTs are often heavier and more complex as they more interacting parts, which causes potentially higher levels of friction than their electro-hydraulic counterparts.

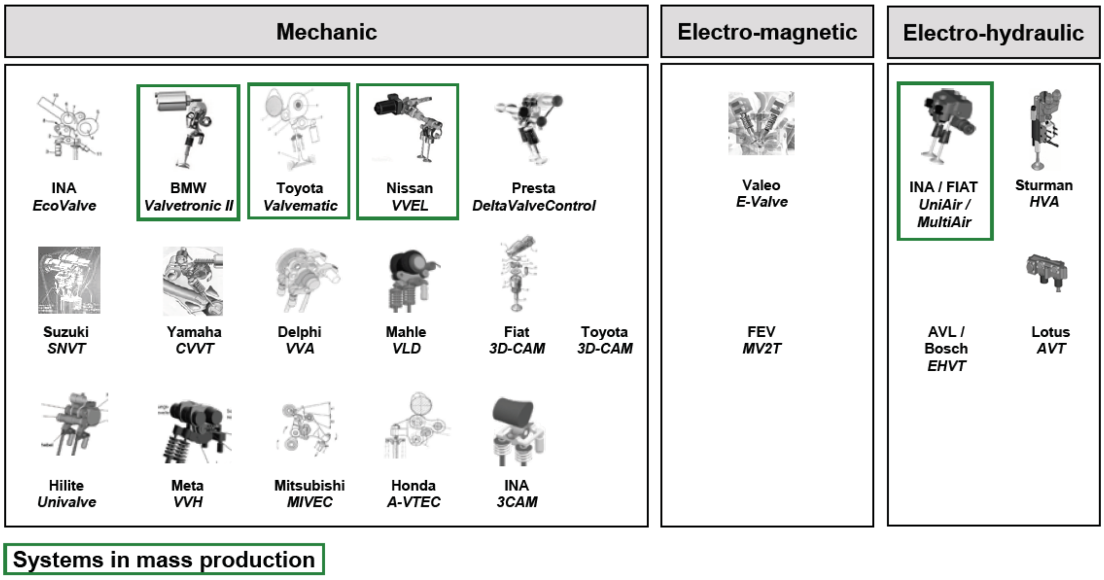


**FIGURE 3 - VARIABLE VALVETRAINS BY VARIOUS MANUFACTURERS *(8)***

## 1.1 Scope of research

Schaeffler and Jaguar Land Rover are working together to develop vehicles with a continuously variable valvetrain integrated with both gasoline and diesel combustion engines. Schaeffler have designed and manufactured an electro-hydraulic system that can fully vary the control of intake valves in an internal combustion engine, marked as “UniAir” (7).

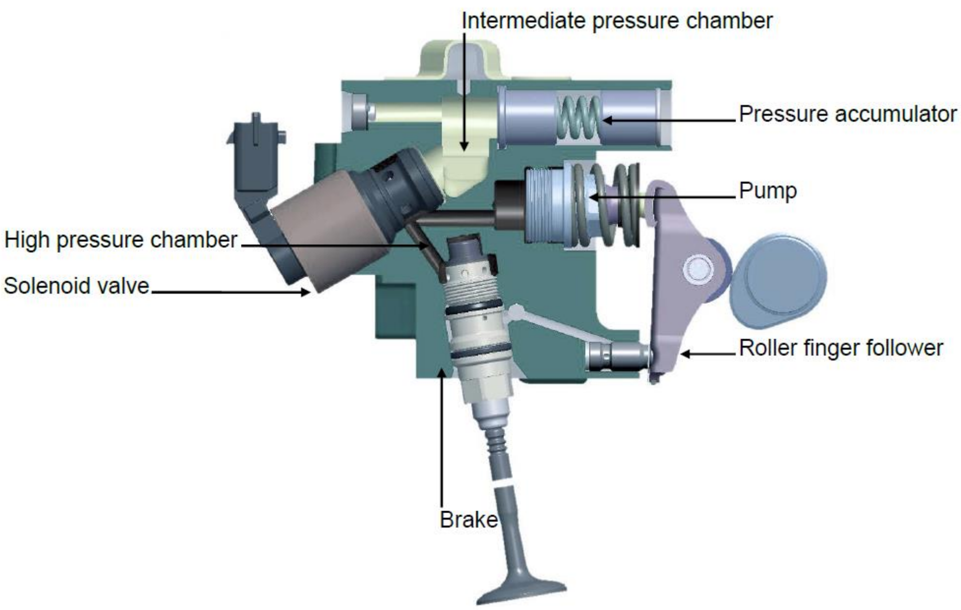


**FIGURE 4 - UNIAIR CROSS SECTION *(8)***

The UniAir system is hydraulically actuated by the engine oil which is circulated around the whole engine. UniAir reduces fuel consumption and increases torque while complying with requirements of emission standards Euro 5 and 6. This allows the engine to always be operated at optimal efficiency. JLR are implementing the UniAir to their AJ200 engines in order to comply with the emission standards while benefitting from the performance enhancing effects (8).

During its development, to understand the performance and efficiency of the UniAir, Schaeffler took measurements of friction force in the valvetrain. The engine oil temperatures ranged from 35°C to 120°C in a room temperature environment. However due to new RDE requirements, JLR engines have a specification for more extreme temperatures, which have a greater effect on the engine oil viscosity. Therefore, further understanding is required to observe the influence of lubricants on the UniAir system, in particular lower viscosity lubricants (through industry trend of lower viscosity grades and potential high levels of fuel contamination in the oil), and to see how the currently used oil specifications can be optimised for engines running CVVL.

This thesis details the design, development and validation of a valvetrain test rig that will determine the frictional operation and efficiency of the UniAir system. This will be achieved by measuring valve lift profiles and power required to operate the system. To study the influence of viscosity it will measure these parameters while varying oil formulation and oil temperatures from 0°C to 120°C and environmental temperatures varying from 0°C to 120°C using climate chambers.

## 1.2 Aim

The aim of this work is to develop an experimental methodology that enables the study on the influence of the viscosity using different lubricants, to understand the performance of frictional operation, and efficiency of a continuously variable valve lift (CVVL) hydraulic valvetrain system UniAir, as used in JLR powertrain applications.

## 1.3 Objectives

In order to achieve this aim, the following objectives were defined:

1. To design, set-up and instrument a valvetrain test rig to evaluate the influence of the viscosity of a lubricant, by monitoring the frictional operation and efficiency of the hydraulic valvetrain. (This also enables the study of all lubricant properties in general, i.e. contamination, aging, formulation, and aeration).
2. To validate the valvetrain test rig by comparing results with previous experimental and simulated tests.
3. To understand the influence of oil viscosity on the UniAir operation (valve lift deviation and leak rates) and efficiency (relative power losses).
4. To generate recommended parameters for optimal performance of the UniAir over the various factors studied, in particular temperature and to recommend future direction for JLR oil specifications.

# 2. Overview of Internal Combustion Engines

To give context to the project and to discuss further the advantages of CVVL, a basic review of conventional internal combustion (IC) engines and technology is presented here.

## 2.1 Operation, Use, Emissions and Economy

The reciprocating internal combustion engine is the most used common form of engine. There are two main types of IC engine; spark ignition (SI), where fuel (usually gasoline in passenger and commercial vehicles) is ignited by a spark, and compression ignition (CI), where the rise of temperature and pressure causes the fuel (usually diesel in passenger and commercial vehicles) to spontaneously ignite upon injection into the combustion chamber. During each crankshaft revolution there are two strokes of the piston, and both types of engine can be designed for the thermodynamic cycle to operate over either four or two strokes of the piston. (9)

Figure 5 illustrates a SI four-stroke engine that has four main stages in the operating cycle:

1. The induction stroke: Inlet valve opens as the piston travels down creating a low-pressure volume which draws in either the air and fuel mixture (indirect injection designs) or the air (direct injection designs).
2. The compression stroke: Both inlet and exhaust valves are closed, and the piston travels up the cylinder. At a defined point during this stroke fuel is injected (direct injection designs). Once the piston is close top dead centre (TDC) ignition occurs either due to the compression or a spark.
3. The power stroke: The combustion causes a rise in pressure and temperature which forces the piston down towards bottom dead centre (BDC). At the end of the power stroke the exhaust valve opens.
4. The exhaust stroke: The exhaust valve remains open as the piston travels back up the cylinder expelling the combusted gases.

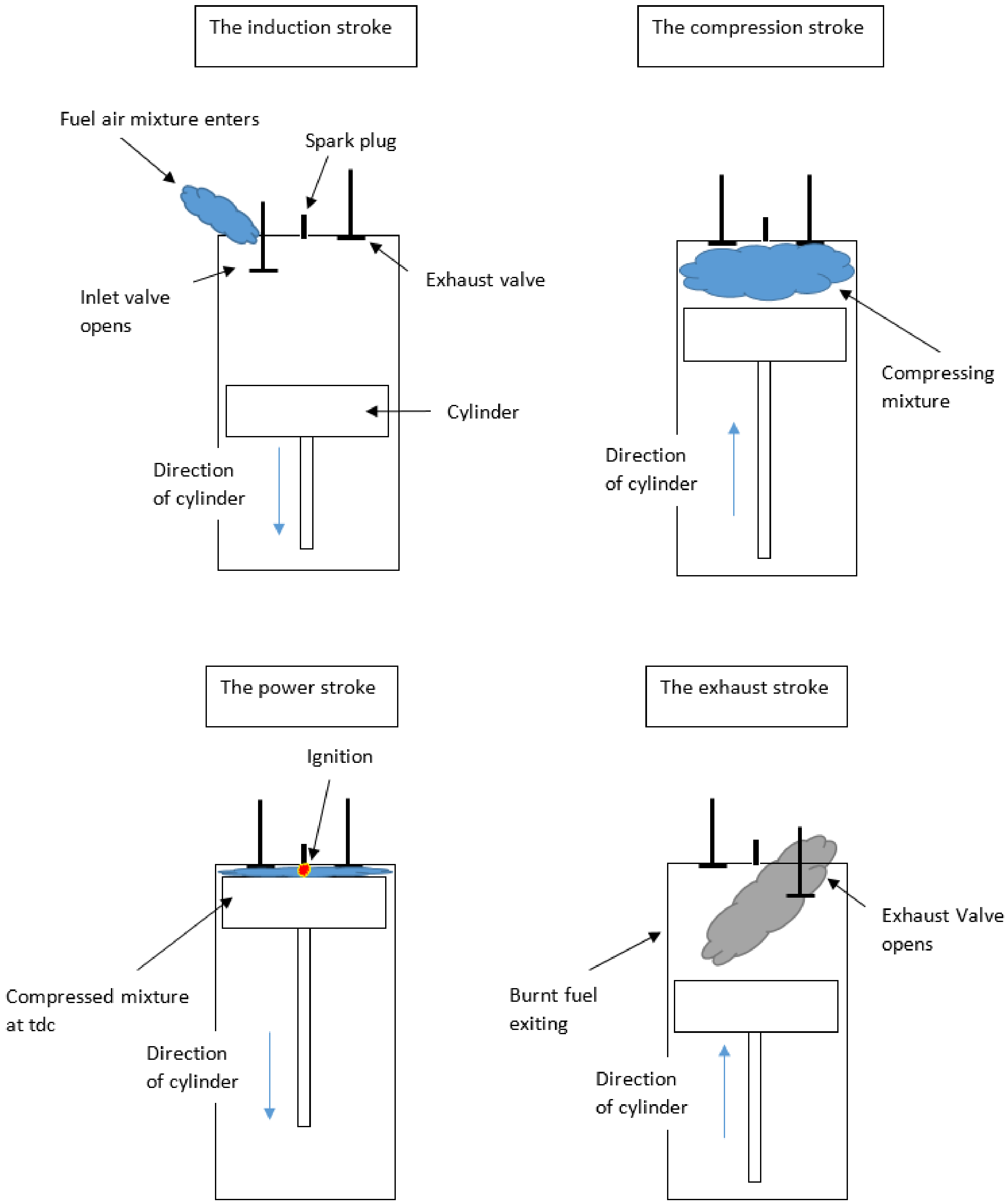


**FIGURE 5 - FOUR-STROKE OPERATING CYCLE**

The Otto cycle is an ideal cycle for SI engines and Figure 6 shows what happens (ideally) to the pressure and volume within such an engine.

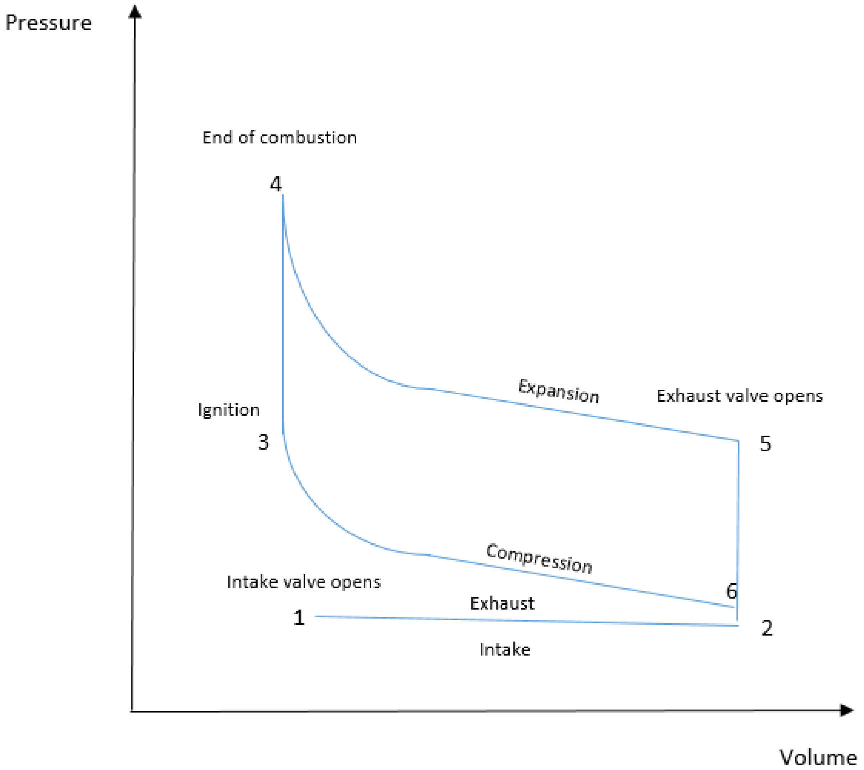


**FIGURE 6 - P-V DIAGRAM OF OTTO CYCLE FOR SI ENGINES.**

Figure 7 shows a realistic example of the Otto cycle during a 'part load' operation. Part load is where the engine is not at full speed, and the throttle is partially pressed down, such as a vehicle cruising at highway speeds (typically 70mph). The throttle governs the mass flow of air drawn into the cylinder. Therefore, when the power requirement is reduced, the throttle reduces the amount of available air.

In a traditional valvetrain design, when the part load operation occurs, the piston moves from TDC towards BDC as normal. This then causes the piston to attempt drag more air into the cylinder (to fill its fixed volume) than is available in the intake manifold downstream of the throttle. As a result, the pressure in the intake manifold can fall below atmospheric pressure, which requires the piston to work against the manifold depression. This is called 'pumping work'. This is illustrated in Figure 7 as the yellow area, known as the pumping loop. If this pumping loop can be reduced, less pumping work is done in the gas exchange process and the cycle has a greater thermodynamic efficiency.

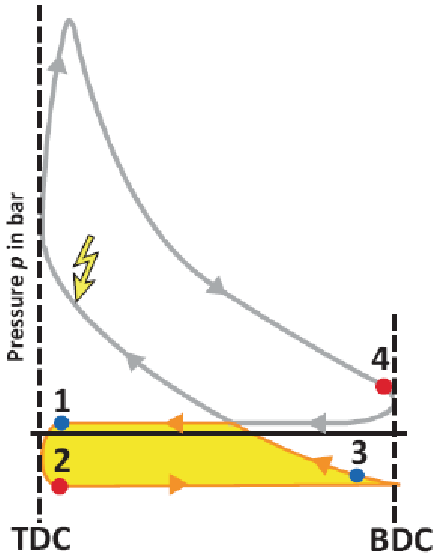


FIGURE 7 - REALISTIC EXAMPLE OF A 'PART-LOAD' OTTO CYCLE.

Figure 8 shows the full load Otto cycle, which is when the throttle position is fully open. This allows the maximum amount of air into the intake manifold, which makes the manifold pressure close to atmospheric. As a result, there is more air available to transfer into the cylinder, which reduces the pumping work done by the engine, which means fewer losses.

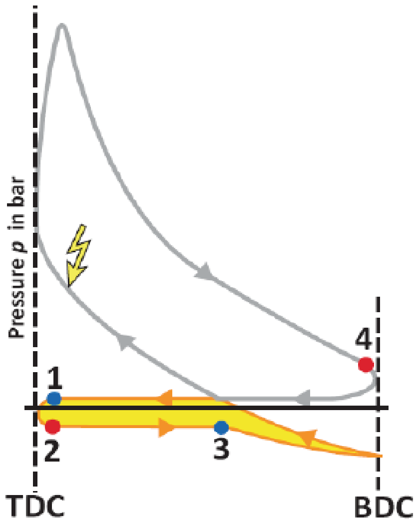


FIGURE 8 - FULL LOAD OTTO CYCLE.

## 2.2 Valve actuation, lift and duration

IC engines require a cyclical supply of fresh air and fuel, while exhausting the burnt gases from the engine during combustion stage. In a four-stroke engine, this intake of fresh air and removal of exhaust is called charge exchange. During the charge exchange, the valve allows gases to leave and enter the cylinder as shown in Figure 5. Figure 9 shows as the camshaft rotates the cam pushes the valve and compresses the spring to its maximum lift by reaching the peak on the cam. This then allows the gases to either flow in or out of the combustion chamber.

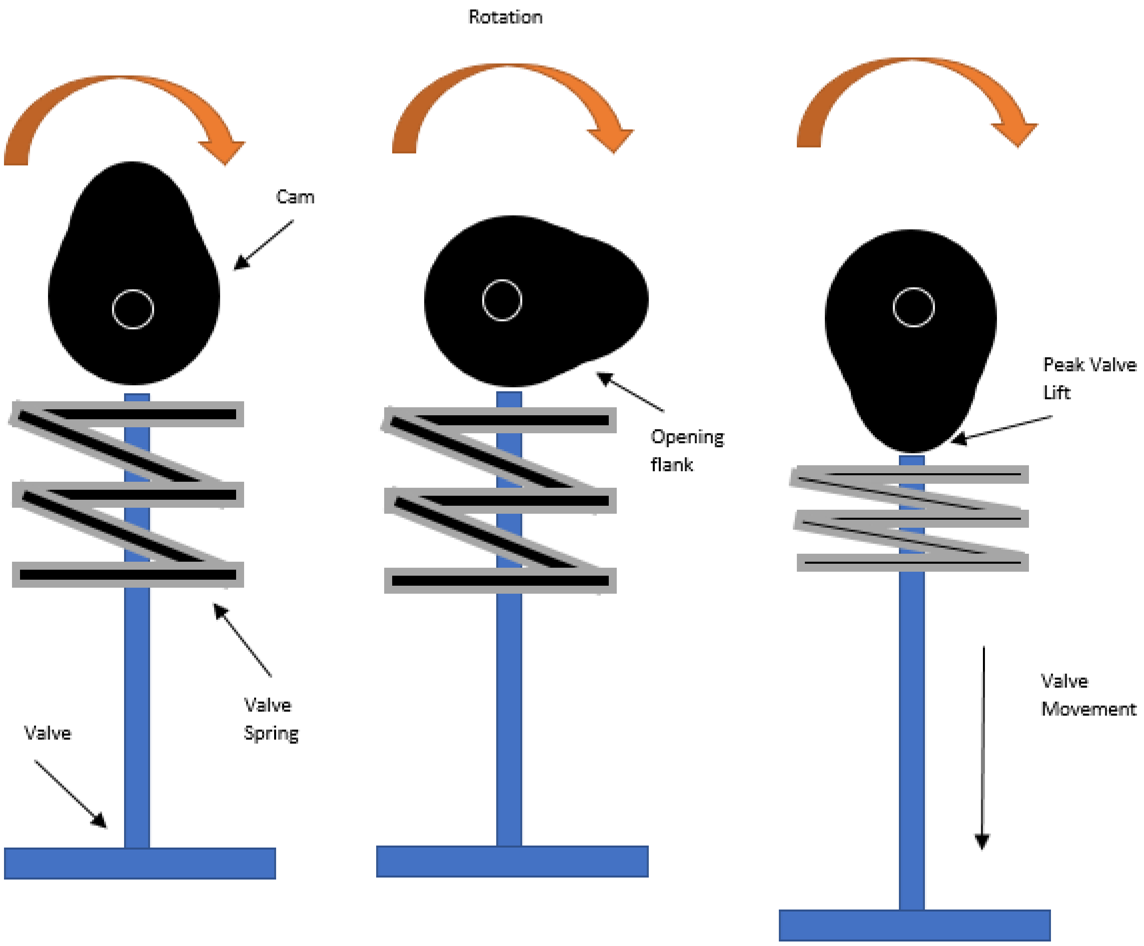


**FIGURE 9 – CROSS SECTION OF TRADITIONAL VALVE LIFT ACTUATED BY CAMSHAFT.**

The cam is shaped so that the valve remains stationary for most of the camshaft rotation (at least 180°). However, as the cam reaches its ‘opening flank’ the valve begins to be pushed open, otherwise known as valve lift. As a result, this fixed valve lift will happen at the same time at the same phase every cycle as illustrated in Figure 10.

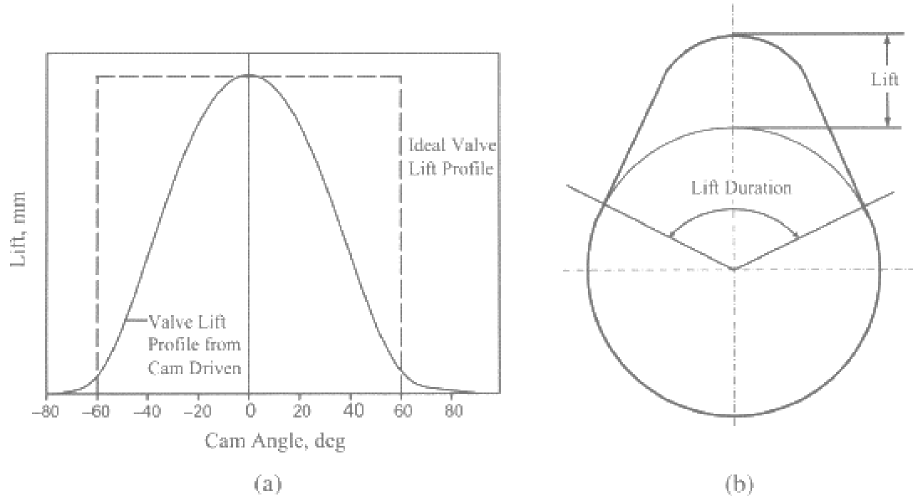


**FIGURE 10 - VALVE LIFT AND DURATION. (A) IDEAL VALVE LIFT (B) TYPICAL CAM PROFILE *(10)*.**

## 2.3 VALVE TIMING AND OVERLAP

The valve timing is the valve opening and closing position relative to the piston position. Figure 11 shows a theoretical valve timing diagram for a typical, conventionally designed valvetrain. The exhaust valve is only open during exhaust stroke and the inlet valve only opens during intake stroke. Both valves are closed during compression and power stroke. (10)

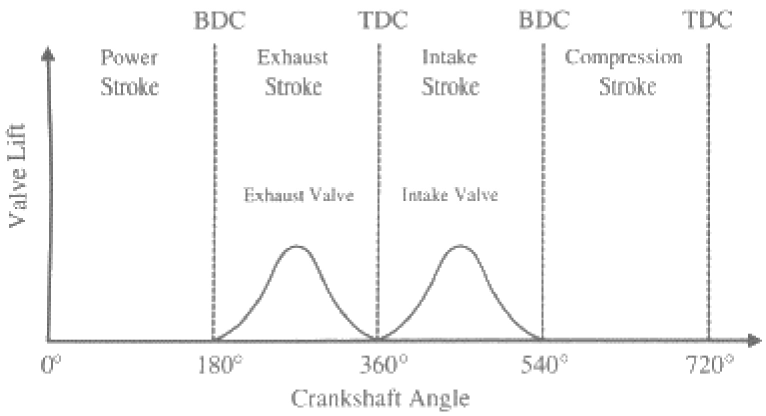


**FIGURE 11 - THEORETICAL VALVE TIMING, LIFT AND DURATION *(10)*.**

The valve overlap is when the exhaust valve opens into the intake stroke and the intake valve opens before the piston reaches TDC on the exhaust stroke. Valve overlap uses the momentum of the exhaust gas leaving the cylinder to induce a fresh charge of fuel from the inlet into the chamber when piston is near TDC. The valve timing and overlap affect volumetric efficiency and therefore engine performance at both high and low speeds (10).

With large valve overlap, the part load and idling operation suffers due to the reduced induction manifold pressure which causes back flow of the exhaust reburning burnt fuel (10).When the engine speed is low, the lift of the valves is the same as at higher engine speeds. This allows excessive amounts of fuel and air into the combustion chamber, which causes higher emissions and fuel consumption. Full load economy is also poor because the exiting exhaust fumes drag some unburnt fuel straight out the combustion chamber. With the requirements of emission standards Euro 5 and 6, there is a need for an alternative method of controlling the valves in an internal combustion engine to improve the efficiency in these situations (10).

## 2.4 Variable valve lift (VVL)

In conventional engines with fixed valve timing, the timing of the intake valve is a compromise between the two contradictory requirements for low and high engine speeds. The magnitudes of the peak lift of the valves are selected for gas flow at maximum engine speeds. As a result, at low engine speeds, the inlet valve has the same maximum lift causing incoming gases to produce less turbulence and may lead to lower torque than having a smaller valve opening (10). For optimal efficiency during the combustion cycle, the lift, phase and timing of the valve lift need to continuously change (7). Therefore, by varying the valve lift with the engine speed, the torque may be enhanced over the entire operating range of the engine. Additionally, reduced valve lift at lower engine speeds reduces the frictional losses of the valvetrain and the energy consumed by the valvetrain decreases with reduction the valve lift, this can improve fuel efficiency at low speeds.

Reduced intake valve lift at low engine speeds has another potential benefit, it increases the intake air velocity (10). This leads to a faster burn rate and improved idle stability. Therefore, VVL can allow reduced valve lift on ‘part-load’ cycles, which regulates the mass flow of air. This reduces pumping work done by the engine, which creates fewer losses from the engine, consuming less fuel and becoming more efficient. This can be seen in Figure 12, which shows a P-V diagram of commercially available continuously variable valve lift systems against a baseline optimised or conventional valvetrain.

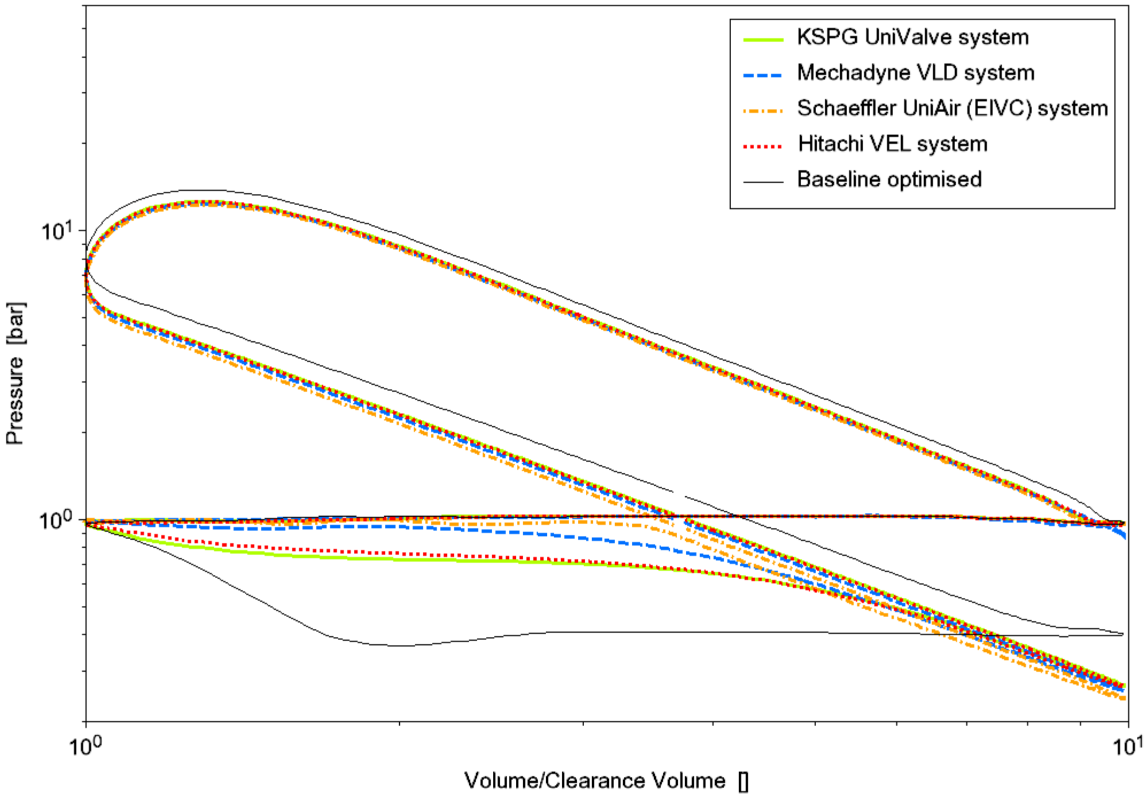


**FIGURE 12 - P-V DIAGRAM COMPARING CVVLS TO CONVENTIONAL VALVETRAIN (CONFIDENTIAL JLR PRESENTATION).**

### 2.4.1 Continuously variable valve lift (CVVL) by UniAir

Continuously variable valve lift is one of the key technologies for implementing low-CO2 emission strategies. Compared to the other CVVL systems seen in Figure 3, the Uni-Air is the only mass-produced system that is not fully mechanical. Figure 4 shows the CVVL system UniAir developed by Schaeffler. On engines with conventional valve timing systems, the camshaft acts on reciprocating hydraulic pumps (one per cylinder) via a roller-type finger follower, illustrated in Figure 13.

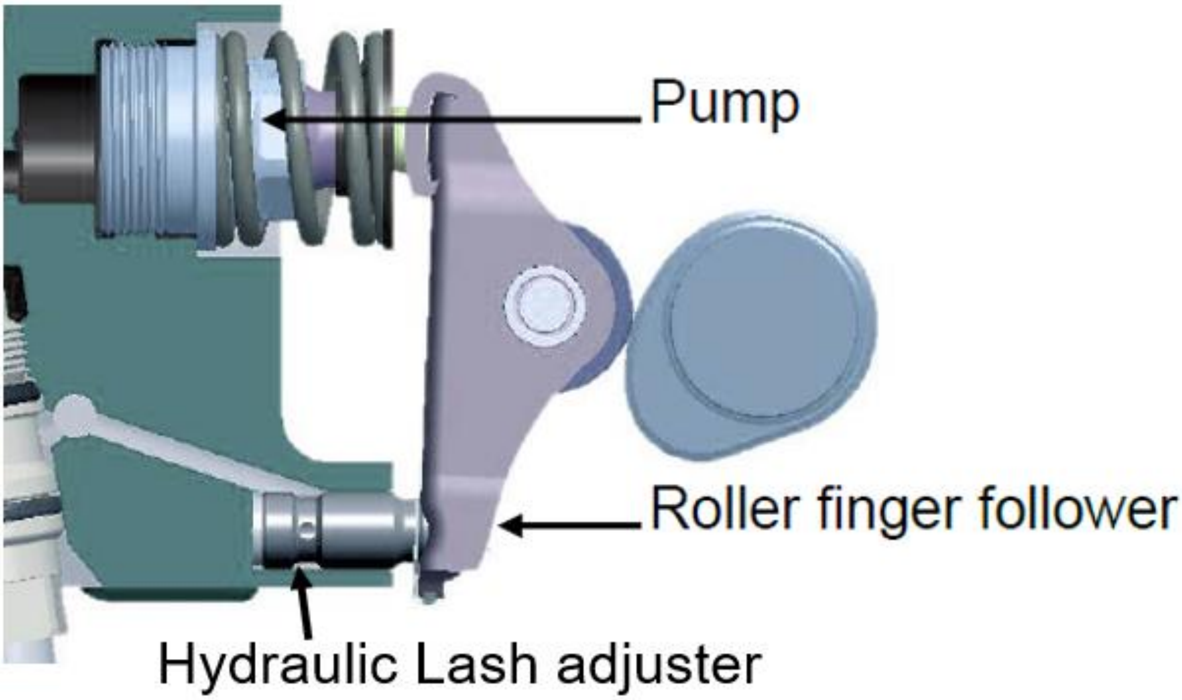


**FIGURE 13 - UNIAIR CROSS SECTION HIGHLIGHTING THE HYDRAULIC PUMP, ROLLER FINGER FOLLOWER AND HYDRAULIC LASH ADJUSTMENT *(8)*.**

The roller finger follower then interfaces with two hydraulic sections of the UniAir, the pump and the hydraulic lash adjuster (HLA). With mechanical solutions when the valve is closed, the valve must have a precisely defined clearance which is referred to as the 'valve lash' or 'valve play'(11). The HLA is an automatic controlled hydraulic valve lash adjustment which eliminates the need to set lash (12).

From the pump, with the solenoid open, engine oil is fed directly to the high-pressure chamber where the solenoid switching valve (SSV) can then control the pressure of fluid flowing to the valve by opening and closing. If the SSV is closed, the oil acts like a hydraulic pushrod forming a 'rigid' link between the cam and valve.


**FIGURE 14 - CROSS SECTION OF THE UNIAIR HIGHLIGHTING THE PUMP, HIGH PRESSURE CHAMBER AND SOLENOID SWITCHING VALVE *(8)*.**

When the SSV is open, engine oil flows through into the intermediate pressure chamber and accumulator, which disconnects the cam and valve. The pressure accumulator serves as a supply to the system volume with the exact amount of oil required, this ensures that any oil that was forced out of the high-pressure chamber in a previous cycle remains available within the system.

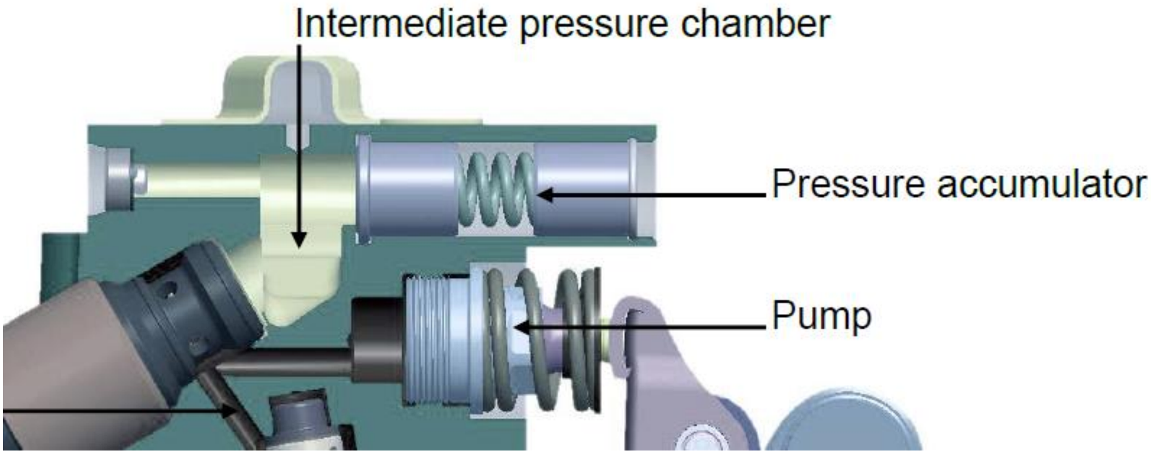


FIGURE 15 - CROSS SECTION OF THE UNIAIR HIGHLIGHTING THE INTERMEDIATE PRESSURE CHAMBER, PRESSURE ACCUMULATOR, SSV AND PUMP *(8)*.

In addition to the CVVL, the UniAir has a brake unit. This is to prevent excessive valve hammering (impact wear) on the valve seat. It does this by hydraulically slowing the valves speed on closure, in much the same way as closing ramps on mechanical cam systems. This protects the valve and valve seating material which enables longer life. Extremely high levels of accuracy are required by the UniAir system to ensure constantly identical valve lifts of the same valve and for each actuator over the entire cylinder head. It is therefore crucial that all the hydraulically driven components ranging from pump to brake conform to the (very high) specified tolerances (7). Under normal driving conditions maximum intake valve lift is rarely required (7). Therefore, maximum lift during the entire cycle, i.e. early opening and late closing of the intake valves along with full lift height, is only used when the driver requires full engine output which only occurs at high engine speeds and high engine torque levels.

#### 2.4.1.1 Lift Modes

The UniAir has lots of interacting parts and surfaces with the main operation and efficiency dependent on the oil. Therefore, particular attention will be focused on the oil to understand the influence of the lubricants viscosity on the UniAir operation (valve lift deviation and leak rates) and efficiency (relative power losses, friction).

##### 2.4.1.1.1 Full Lift

The valve lift profile is fully defined by the cam profile considering mechanic and hydraulic ratios. When in Full Lift all the camshaft motion is transferred to the valves because the solenoids are energized and closed (Figure 17). It is considered as the maximal possible engine valve lift which is influenced by: engine speed, oil temperature and leakages. The valves are individually controlled by the control software. They are individually monitored using the current during each switching process for each cylinder and then re-adjusted depending on the operating condition using data maps in the engine control system (7,8). The challenge in this case is detectability of the current curve over the entire required temperature range and the oil viscosity associated with it. All switching valve components must be perfectly matched to each other to ensure this function. To generate Full Lift, Figure 16 illustrates the valve lift profile (black line) and solenoid status (red line) and current (orange line). This further illustrates that when the solenoid is closed, valve lift is generated.

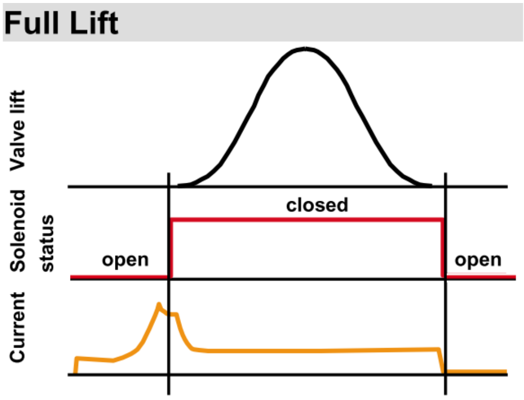


FIGURE 16 - EXAMPLE OF FULL LIFT VALVE LIFT ILLUSTRATING SOLOENOID AND CURRENT STATUS

Figure 17 shows the cross section of the UniAir while demonstrating the oil path to generate Full Lift. The oil is illustrated as the orange body, while the direction of oil flow is illustrated with red arrows.

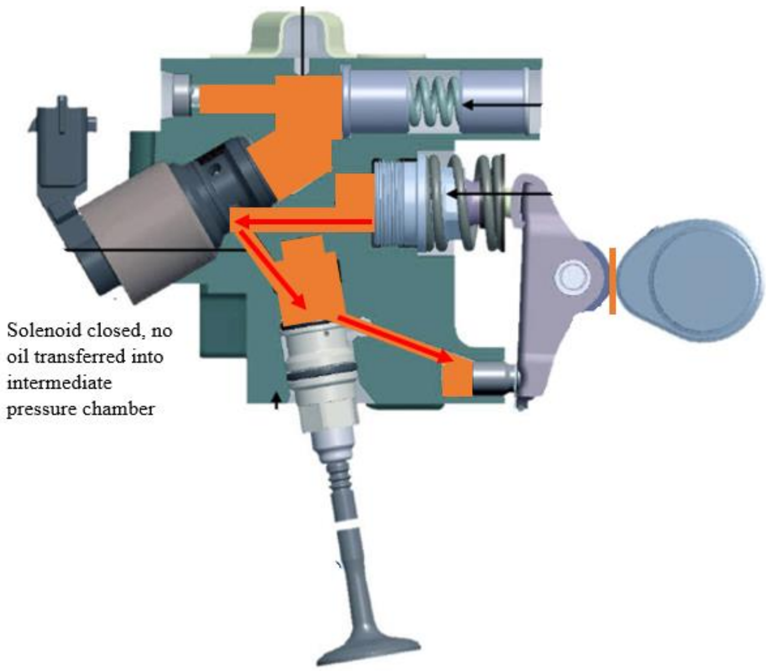


FIGURE 17 - OIL PATH IN UNIAIR GENERATING FULL LIFT

### 2.4.1.1.2 No Lift

Alternatively, for No Lift there is no intake valve lift during this mode as the solenoids are not energized and constantly left open. Figure 18 demonstrates the No Lift valve lift (green line), Full lift reference (grey line), solenoid status (red line) and current (orange line).

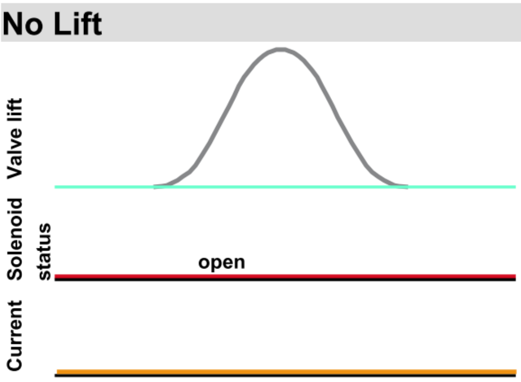


FIGURE 18 - EXAMPLE OF NO LIFT VALVE LIFT ILLUSTRATING SOLOENOID AND CURRENT STATUS

Figure 19 demonstrates the oil path in the UniAir when generating No Lift valve lift. This is illustrated with the orange body as the oil, and red arrows showing the direction of oil flow. It can be seen the solenoid is open, which allows the oil to transfer into the intermediate pressure chamber, which results in No Lift.

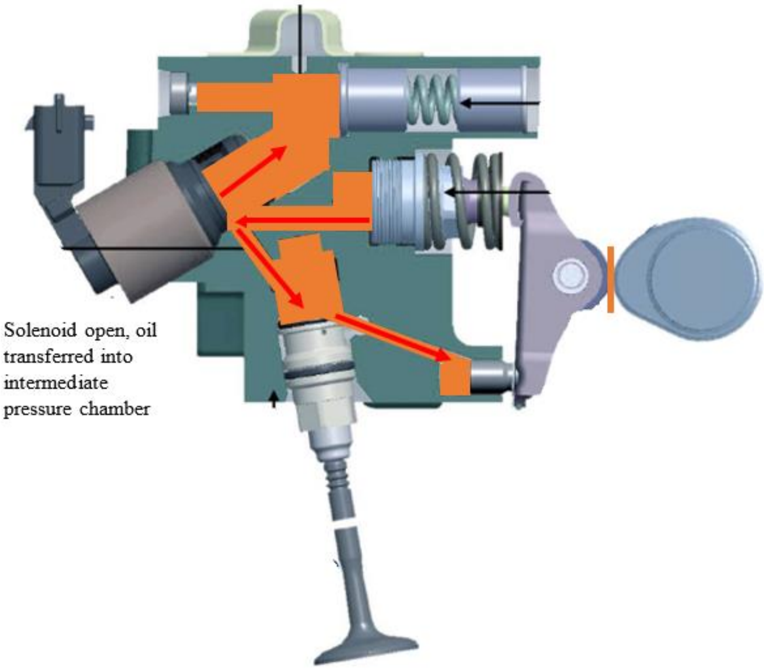


FIGURE 19 - OIL PATH IN UNIAIR GENERATING NO LIFT

#### 2.4.1.1.3 Early Intake Valve Closing (EIVC)

To generate EIVC, the engine control unit (ECU) specifies the intake valves closing point in relation to the crank shaft. Figure 20 illustrates for EIVC, the valve opening point is fixed as a result of the cam timing. At this point, the solenoid is closed and generates valve lift as experienced in Full Lift. With a given closing angle, the solenoid opens mid valve lift allowing the pressurized oil to transfer into the intermediate chamber as seen in No Lift. Figure 20 demonstrates a Full Lift reference (grey line), various different EIVC lifts dependent on closing angle (green solid and dotted lines), the solenoid status (red line) and current (orange line).

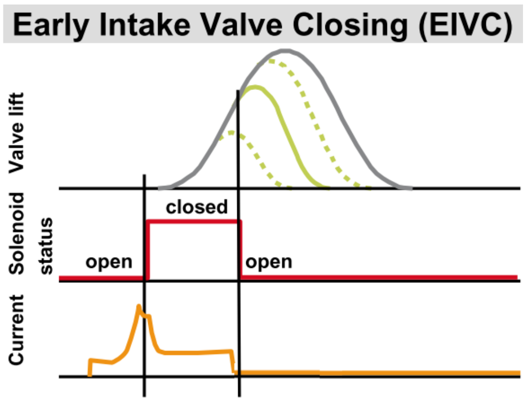


FIGURE 20 - EXAMPLE OF EIVC VALVE LIFT ILLUSTRATING SOLENOID AND CURRENT STATUS

#### 2.4.1.1.4 Late Intake Valve Opening (LIVO)

For LIVO, the ECU specifies the intake valve opening point. The valve closing point is fixed as a result of the cam profile. Figure 21 illustrates how the ECU supplies the required current to energise the solenoid which closes the solenoid to generate LIVO.

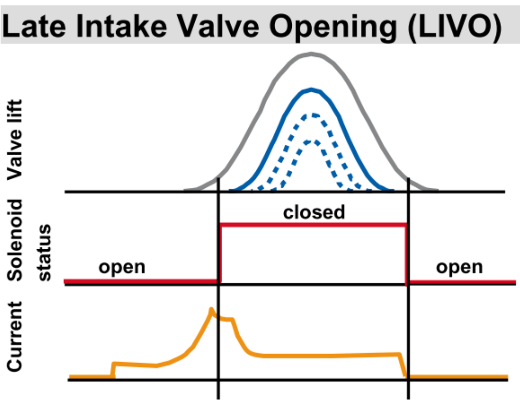


FIGURE 21 - EXAMPLE OF LIVO VALVE LIFT ILLUSTRATING SOLENOID STATUS AND CURRENT

# 3. VALVETRAIN TRIBOLOGY

Tribology is the study of friction, wear and lubrication. A substantial part of the total energy consumption in an IC engine is expended in overcoming wear and frictional losses. Approximately 18% of all frictional losses come from the valvetrain, and more efforts are directed toward reducing valvetrain friction to achieve fuel efficiency and emission reduction (10). Many factors affect the influence of friction loss such as operating conditions and lubrication (10).

## 3.1 FRICTION

Friction is defined as the resistance to movement between two bodies in contact. This can be calculated by:

$$F = \mu N \tag{1}$$

Where:

$$F = Friction\ force$$
$$\mu = \ Coefficent\ of\ friction$$
$$N = Normal\ force$$

It is noted that the force required to begin sliding is often greater than the force required to sustain sliding, therefore the static friction is often higher than the kinetic friction (10). However, it is important to state that these laws can differ in actual practice. This is because to get a true value of the coefficient of friction relies on an understanding of the nature of surface interaction, the real area of contact, atomic interaction, surface roughness and surface energy (10).

### 3.1.1 SURFACE INTERACTIONS

When two solids are placed in contact, some regions of the surface will be close together and others will be further apart due to non-conformed surfaces in contact (10). These regions in contact are known as junctions and the sum of all the junctions constitute the real area of contact $A_r$ (10). The lower the real area of contact is, the higher the contact stresses are.

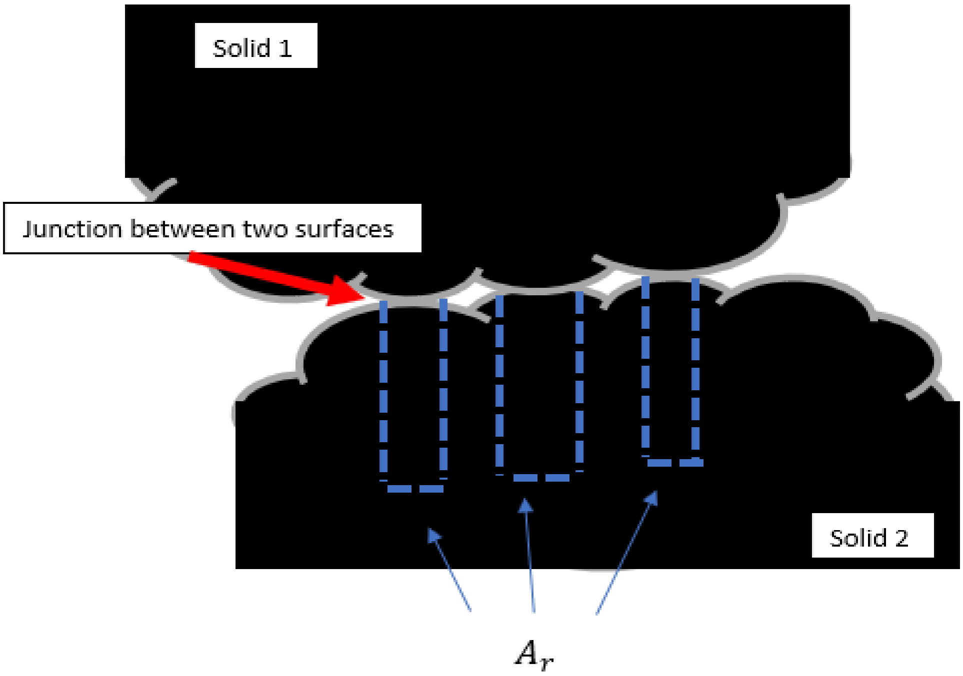


**FIGURE 22 - TWO SURFACES IN CONTACT HIGHLIGHTING THE REAL CONTACT AREA ($A_r$) BY JUNCTIONS**

The largest compressive stress that such a region of material can carry without further plastic yielding is known as its penetration hardness $H$ (10). When the two surfaces are brought into contact and a normal load applied, plastic deformation will occur until the total real area of contact reaches a value given by (10):

$$A_r = \frac{L}{H} \tag{2}$$

Where:

$$L = Applied\ load$$

### 3.1.1.1 Rolling Friction

Rolling friction is the resistance to motion when an object is rolled over a surface as seen in Figure 13, with the interface between camshaft and roller finger follower. Typically, the friction force of a rolling contact is lower for smooth surfaces than rough, however in practice pure rolling rarely exists. Rolling occurs at a very small number of points rather than over the entire surface. The rolling contact is a combination of rolling and sliding or slip. This slip accounts for a major part of the total resistance to rolling as it must overcome the sliding resistance at the interface known as a rolling friction force. The surface temperature rises above the bulk temperature due to the rolling friction or ‘rubbing’. The friction temperature $T_r$ is a function of load, friction, speed and thermal conductivity of the contacting materials:

$$T_r = f\left(\frac{\mu p V}{k}\right) \tag{3}$$

Where:

$$T_f = asperity\ flash\ temperature$$

$$\mu = coefficient\ of\ friction$$
$$p = Hertzian\ pressure$$
$$k = thermal\ conductivity$$
$$V = sliding\ velocity$$

The flash temperature can significantly affect the tribological characteristics, including friction, wear and lubricant degradation at surfaces (10).

## 3.2 WEAR

Wear is the progressive damage, involving material loss which occurs on the surface of a component as a result of its motion relative to the adjacent working parts. Within wear there are many different wear mechanisms that can occur which can be seen in Table 1.

| | | |
|---|---|---|
| Abrasion-corrosion | Fretting fatigue | Run-in |
| Abrasive wear | Fretting wear | Scoring |
| Adhesive wear | Galling | Scratching |
| Burnishing | Goughing abrasion | Scuffing |
| Chemical wear | High-stress abrasion | Severe wear |
| Corrosive wear | Low-stress abrasion | Sliding wear |
| Deformation wear | Mechanical Wear | Slurry abrasion |
| Dry sliding wear | Mild wear | Spalling |
| Erosion | Oxidative wear | Thermal Wear |
| Fatigue wear | Pitting | Three-body abrasion |
| Fracture | Plowing | Transfer abrasion |
| Fretting | Polishing wear | Two-body abrasion |

**TABLE 1 - WEAR MECHANISMS**

Due to the many wear mechanisms that could occur in a valvetrain, it is out of the scope of this work to perform wear studies but it is important as a consideration on its potential effect on the operational performance and efficiency of the UniAir and, of course, if the system is function correctly it should cause minimal wear.

## 3.3 Lubrication

The UniAir system is dependent on the engine oil, it is therefore crucial to understand the important parameters of the oil when analysing the efficiency and function of the system. Lubrication plays a critical role within an internal combustion engine (10):

- Provides thin layers between surfaces
- Reduces friction and wear
- Cool rubbing surfaces
- Protects from corrosion
- Carries debris particles away

The main purpose of lubrication is to provide a film between moving surfaces to reduce friction. The durability and strength of this film is directly related to the viscosity of the lubricant, the speed and loads experienced by the moving surfaces (10).

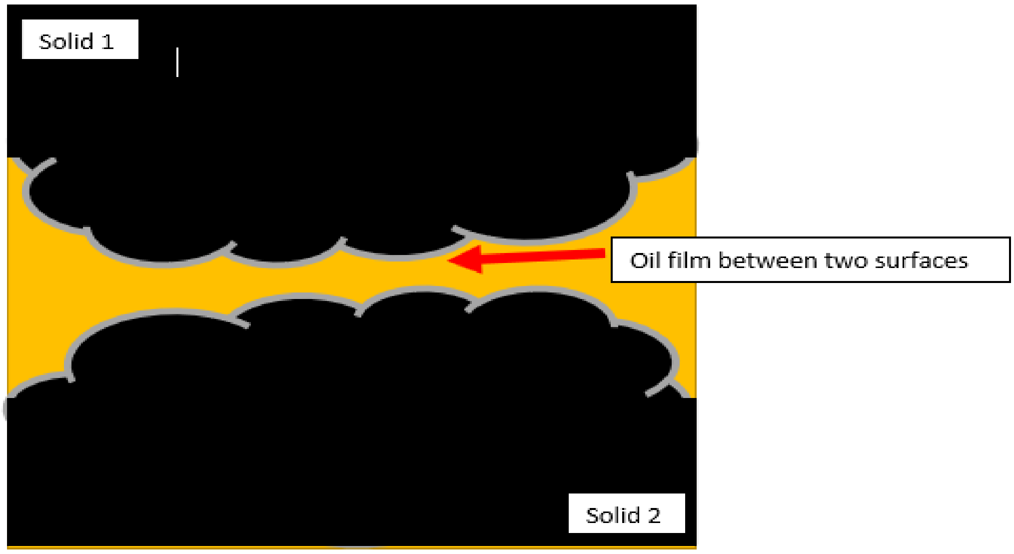


**FIGURE 23 - HOW OIL FILM THICKNESS PROVIDES SEPARATION BETWEEN TWO SURFACES**

The Stribeck curve shows the relationship between coefficient of friction ($\mu$), oil film thickness with lubricant viscosity ($\tau$), moving part speed ($v$) and load/pressure ($p$) (13).

$$Sommerfeld\ number = \tau \frac{v}{p} \tag{4}$$

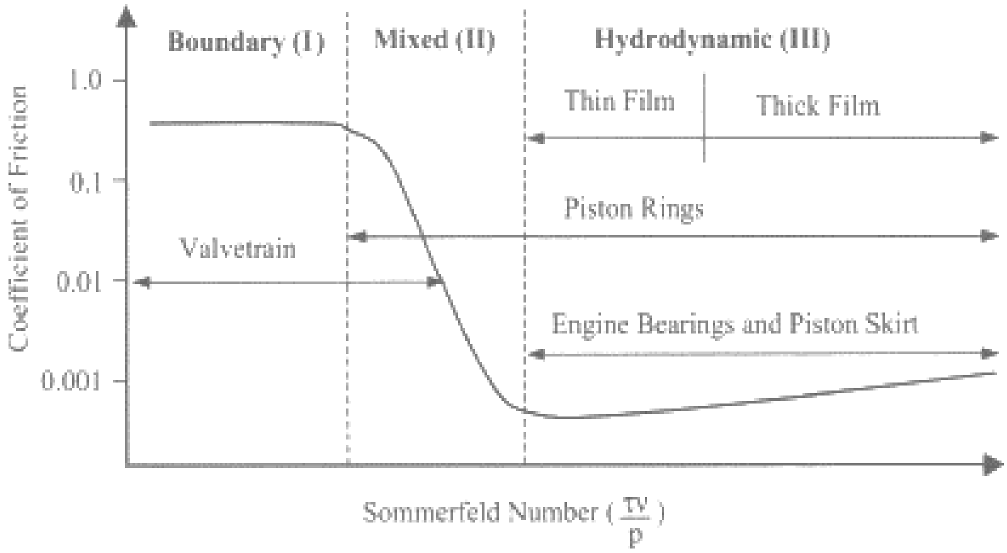


**FIGURE 24 - STRIBECK DIAGRAM, ILLUSTRATING VARIOUS LUBRICATION REGIONS *(10)(13)***

The Stribeck diagram describes four different lubricant regimes:

- Boundary
- Mixed
- Hydrodynamic
- Elasto-hydrodynamic (a special case of hydrodynamic)

This shows that a low lubricant viscosity, low speed and high load will create high levels of friction due to the lack of lubricant film which generates separation between two surfaces. Once the speed and lubricant viscosity starts to increase the friction levels drop quite dramatically due to thin lubricant films starting to develop. This creates the intended separation between the two surfaces and thus reducing friction. However, when the situation is high lubricant viscosity and high speed and low load the friction levels start to rise again. This is because the lubricant film starts to thicken, which causes vicious drag causing excessive energy being consumed to shear the lubricant (10).

The most ideal situation for low friction is the thin film of the hydrodynamic film, which is critically dependant on lubricant viscosity (10). Another type of hydrodynamic lubrication is elasto-hydrodynamic lubrication, which typically occurs at high contact pressures and experiences a large viscosity increase (10). This results in a rigid lubricant film that causes elastic deformation of the surfaces in the lubricating zone (10). This film thickness is known as elasto-hydrodynamic lubrication (EHL) and can be calculated by (14):

$$h = 5 \times 10^{-6} \times (nuR_r)^{0.5} \tag{5}$$

Where:

$$h = EHL\ film\ thickness\ (mm)$$

$$n = Lubricant\ viscosity\ at\ working\ temperature\ (Poise)$$

$$u = Entrainment\ velocity\ \left(\frac{mm}{s}\right)$$

$$R_r = Relative\ radius\ of\ curature\ (mm)$$

### 3.3.1 VISCOSITY

Viscosity is one of the most important properties of a lubricant (10). Viscosity comes from the resistance to sliding in a fluid, or resistance to deformation by shear stress (15).

Viscosity is defined in two ways:

- Dynamic Viscosity – the absolute resistance to shear

$$\tau\ (Shear\ stress) = \ \mu\ (Dynamic\ viscosity)\frac{du}{dy} \tag{6}$$

Where:

$$u = Velocity$$

$$y = Distance$$

- Kinematic Viscosity – combination of dynamic viscosity and density

$$V\ (Kinematic\ viscosity) = \frac{\mu}{\rho(Density)} \tag{7}$$

Empirically, it is found that the double natural logarithm of kinematic viscosity is proportional to the natural logarithm of temperature (15). Engine lubricant viscosities are strongly dependant on temperature and pressure (10).

To quantify the variation of viscosity with temperature, a viscosity index (VI) is used. This is done by comparing the kinematic viscosity of an oil at two different temperatures. When the index value is high, this means a relatively small change in viscosity over a temperature range, whilst a low index indicates a large change (16). If an oil has the incorrect viscosity index, this could indicate that when the engine is cold the viscosity could be unnecessarily high, creating poor circulation of the lubricant and excessive friction (16). Therefore, as the UniAir is entirely dependent on oil for its operation, and importantly the engine oil, the lubricant temperature could play a vital role in its operational performance and efficiency.

### 3.3.2 Compressibility

The degree of compressibility of a substance is characterized by the bulk modulus of elasticity, K, which is defined as (15):

$$K = -\frac{\partial p}{\frac{\partial V}{V}} \tag{8}$$

Where:

$$\partial p = difference\ in\ pressure$$

$$\partial V = small\ increase\ in\ original\ volume$$

$$V = Original\ volume$$

K may also be expressed as:

$$K = \rho\left(\frac{\partial p}{\partial \rho}\right) \tag{9}$$

Where:

$$\rho = density$$

$$\partial \rho = difference\ in\ density$$

However, the bulk modulus of most liquids is very high and even with large pressure changes the change in density is very small (15). Knowing the density will remain relatively constant with pressure change can help with identifying which variables are a priority when taking measurements and help with identifying why other variables change, such as viscosity.

### 3.3.3 OIL

Mineral oils acting alone as a lubricant are not sufficient in modern engines. Synthesized chemicals called additives are blended into the base stocks of mineral oils to help them perform more efficiently in their environments (10). These lubricant base stocks are either mineral or synthetic origin.

Oils are obtained from three main sources (10):

- Animals – used for certain purposes but decompose too easily for modern internal combustion engines
- Vegetable – Again decompose too easily for modern internal combustion engines.
- Mineral – These are refined from crude petroleum and are far more stable than other types. They form the base of the majority of modern lubricants. However, they still require additives to ensure improved lubricating properties

However, synthetic oils consist of chemical compounds not present in crude oil. These can bring a range of properties to the oil:

- Esters – high temperature stability
- Polyglycols – Water soluble
- Silicones – Non-combustible, fire resistant lubrication
- Fluoro compounds – Low volatility, space applications (very expensive)
- Synthetic hydrocarbons – higher viscosity index than mineral oils

Due to the expensive development of fully synthetic oils, semi-synthetic oils are developed to reduce the price. A typical blend of a semi-synthetic oil is a mineral oil with no more than 30% synthetic oil.

#### 3.3.3.1 Oil additives

Small amounts of some or all of the following in Table 2 are added to give the oil improved lubricating properties (10).

| | |
|---|---|
| 'Oxidation inhibitors' | Excessive oxidation of the oil (chemical reaction) can lead to sludging, deposits, viscosity increase. These additives chemically inhibit oxidation. |
| Corrosion inhibitors | Act by adsorbing on the metal surface to form an impervious film. |
| Oiliness agents | Long chain surfactants with a polar end groups. The polar group is attracted to the metal surface forming a thin film. |
| Anti-wear additives | Sulphur or phosphorous containing compounds. Form a thin chemically bonded layer. Withstand high temperatures. Zinc Dialkyl Dithiophophate (ZDDP) adheres to metallic surfaces holding the oil in place. |
| Extreme-pressure additives | Organic compounds with a reactive non-metal (S, CI, or I). React with hot metal surfaces (e.g. when two asperities collide heat is generated) forming a lubricating film. |
| Viscosity index Improvers | Causes the lubricant to have a higher VI (i.e. the viscosity falls less rapidly with temperature). Long chain polymers that 'unravel' at higher temperatures and restrict oil mobility, thereby resulting in a higher viscosity than would be achieved otherwise. |
| Pour point depressants | Added to mineral oils to breakdown wax crystals which form at low temperatures. I.e. they can be poured at low temperatures.' |

**TABLE 2 - TABLE OF OIL ADDITIVES AND THEIR CHARACTERISTICS *(10)***

## 3.4 Engine lubrication

Most of the materials in typical internal combustion engines cannot last long under the stresses, speed and temperatures experienced inside the engine without lubrication (10). Therefore, supplying these components with lubrication are critical to the longevity of the component. Engine parts are lubricated by one of three systems: a full pressure system, a splash system or a combination pressure-splash system (16).

This pressure-splash combination system found in the AJ200 Jaguar Land Rover engine pumps oil to the cylinder head to lubricate the UniAir CVVL system and camshaft. The oil is then forced through drilled and cast passages called oil galleries to the camshaft journal bearings and to pressurize the UniAir. The UniAir has designed-in oil leakage paths which helps replenish the UniAir with a new supply of oil and creates a fine oil mist inside the cylinder head. This is extremely useful as it helps lubricate all components within the valvetrain, such as the interface between the camshaft and roller finger follower.

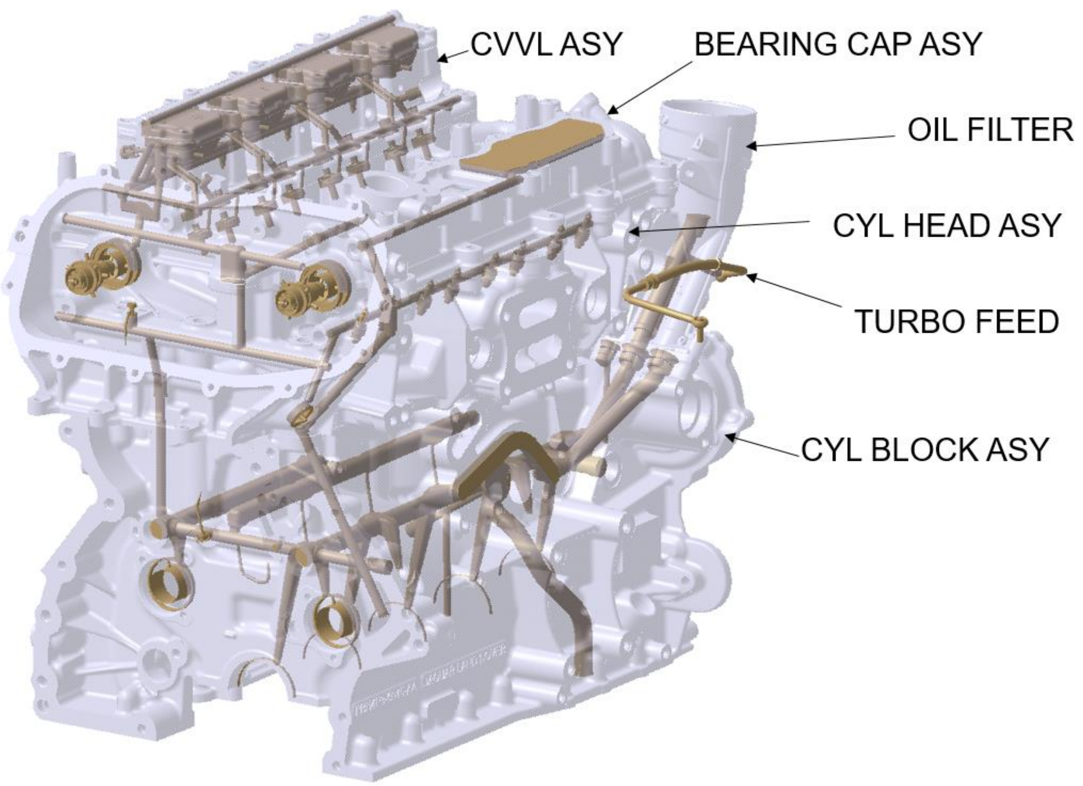


**FIGURE 25 - AJ200 PETROL ENGINE OIL PATH**

## 3.5 Tolerances and Statistical analysis

There are many different factors to consider when analysing a valve train and in conjunction with section 1.3 that there are multiple measurements taken from this test rig. Therefore, it is important to understand what will classify as a valid measurement and how to compare across different parameters.

### 3.5.1 Uncertainty of Measurement

To gain confidence in the robustness of the rig, the National Physical Laboratory suggest finding the uncertainty of measurement. The uncertainty of measurement is the 'margin of doubt that exists about the result of any measurement' (17). To quantify an uncertainty, two values are needed, the width of margin and confidence level. To do this, statistical calculations are used which require at least three measurements of any measured part.

#### 3.5.1.1 Calculating width of margin

A standard uncertainty is a margin around a value as plus or minus the standard uncertainty value. With a set of at least three measurements, the mean and standard deviation $s$ can be calculated. From this the standard uncertainty, $u$, is calculated from:

$$u = \frac{s}{\sqrt{n}} \tag{10}$$

Where $n$, is the number of measurements (17).

#### 3.5.1.2 Calculating the confidence level

Multiplying the standard uncertainty by a coverage factor $k$ gives a result called the expanded uncertainty $U$.

$$U = ku \tag{11}$$

Using different coverage factors gives different levels of confidence:

$$k = 1\ for\ a\ confidence\ level\ of\ 68\%$$

$$k = 2\ for\ a\ confidence\ level\ of\ 95\%$$

$$k = 2.58\ for\ a\ confidence\ level\ of\ 99\%$$

$$k = 3\ for\ a\ confidence\ level\ of\ 99.7\%$$

By using standard uncertainty and coverage factor, this provides a confidence level in the result that is it robust and repeatable. However other variables other than uncertainties need to be understood for robustness, such as tolerances.

### 3.5.2 Tolerances

Tolerances are not uncertainties. They are accepted limits which are chosen for a process or product (17). This rig has various inputs and outputs, so to calculate the level of robustness each tolerance for every variable needs to be understood.

#### 3.5.2.1 - JLR System functionality for Lift

The tolerance of the UniAir system is explained by JLR and supplied in 14.4, as the use of two different parameters to understand the measured value:

- Variation – The maximum difference of the mean peak valve lift of the two valves of one cylinder among all cylinders
- Repeatability – The maximum difference of the mean peak valve lift of one cylinder to an identical separate test, cycle-by-cycle repeatability.

This will help quantify the validity of the results generated in this thesis.

### 3.5.3 Uncertainty of Measurement and Tolerances

There may be a situation where a result falls within a specified upper/lower limit tolerance, however the standard uncertainty overlaps this limit (17). This is illustrated in Figure 26:

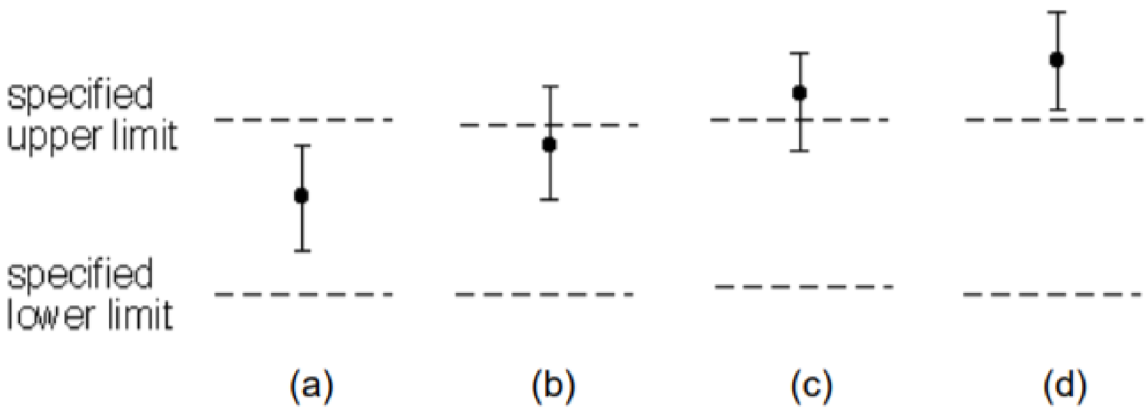


**FIGURE 26 - FOUR SITUATIONS OF A MEASUREMENT RESULT, ITS STANDARD UNCERTAINTY VALUE AND TOLERANCE LIMITS *(17)***

- In case (a), both the results and uncertainty fall inside the specified tolerance limits. This is classed as a 'compliance'.
- In case (d), both the result and uncertainty fall outside the specified tolerance limits. This is classed as 'non-compliance'.
- In case (b) and (c), both are neither completely inside nor outside the limits. Therefore, no firm conclusion about compliance can be made.

### 3.5.4 Statistical difference

Once a set of data is measured is important to be able to distinguish a difference between another set of data. To do this a t-test is used which is used to determine if there is a significant difference between the mean of two groups (18). To use a t-test requires three key data values:

1. The difference between the mean values from each data set (the mean difference)
2. The standard deviation of each group
3. Number of data values from each group

The outcome of a t-test is a p-value. To understand what this p-value is, the level of confidence must be set. To set the confidence the term alpha is used to represent that value. A standard confidence level of 95% (or 0.95) is required, the alpha would be the remaining percentage of 100 as 0.05. Therefore, if the p-value is less than the alpha, the result would be there is a 95% chance that the result is significantly different. The t-test defines whether the difference between the groups represents a true difference in the study or if it is a possible a meaningless random difference (18).

# 4. VALVETRAIN TESTING

There are many valvetrain test apparatus and methodologies presented in the literature and it is likely that many more are also considered to be proprietary information within automotive companies. To gain a better understanding of how valvetrain test rig is to be designed, other studies will be analysed to take into consideration their findings. These can then be compared and adapted to JLR requirements to validate the rig. First, a review of the test platform used internally by Schaeffler during the development of UniAir is presented before a discussion of other test rigs found in the literature.

## 4.1 SCHAEFFLER VALVETRAIN TEST RIG

Schaeffler have previously designed and tested a valvetrain test rig as shown in Figure 27. The rig design was created to test and compare the friction torque on a Jaguar AJ200P 2l cylinder head with a UniAir. The UniAir that was tested on Schaeffler's test rig was pre-production, therefore it is expected that the friction values on this rig will differ to the production UniAir tested in this project.

Schaeffler's primary objectives was to measure the friction torques of the valvetrain at various oil temperatures. The friction measurements were captured by using a motored cylinder head (Figure 27), which used a torque transducer to monitor the torque required to overcome the frictional forces in the UniAir system. The friction measurements will capture the friction in the following areas:

- CVVL system friction
  - Hydraulic dissipation/losses
  - Pump drive friction
- Valve train friction
- Camshaft friction
  - Radial bearings
  - Axial thrust face with sealing rings

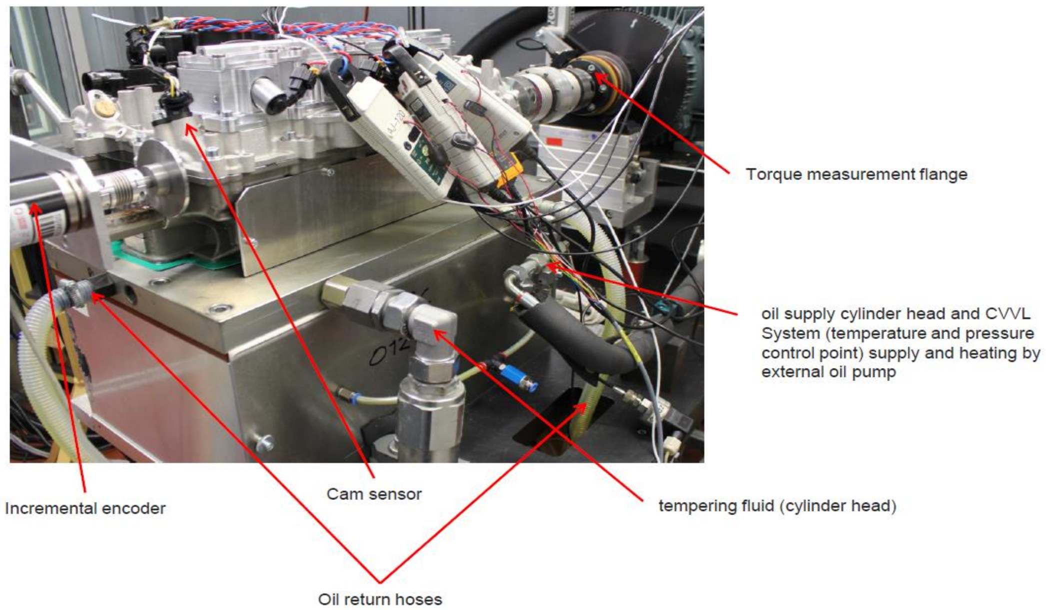


**FIGURE 27 - JLR AJ200P CVVL FRICTION MEASUREMENT RIG**

To get realistic friction data the speed related pressure map shown in Table 3 was used and the boundary conditions were as stated in Table 4.

| Speed (erpm) | 680 | 1000 | 2000 | 3000 | 4000 | 5000 | 6000 | 6500 |
|---|---|---|---|---|---|---|---|---|
| Oil pressure (bar) | 1.06 | 1.31 | 1.31 | 2.03 | 2.54 | 3.06 | 3.06 | 3.56 |

**TABLE 3 - SCHAEFFLER'S SPEED RELATED PRESSURE MAP**

| | |
|---|---|
| Crankshaft speed (erpm) | 0 – 5000 rpm |
| Oil and Cylinder head temperature | 35, 90, 120°C ± 1°C |
| Oil pressure to cylinder head entrance | Speed related pressure map in Table 3 |
| Cooling water temperature | Not measured, just used for regulation of cylinder head temperature |
| Engine oil: | Castrol SAE 0W-20 **A12013PBABOT936**, fresh oil |

**TABLE 4 - SCHAEFFLER BOUNDARY CONDITIONS FOR FRICTION TEST**

Figure 28 shows how Schaeffler created a test procedure to achieve constant friction levels by fixing the motor speed for 2 minutes for each camshaft interval. The first minute of each 2 minute motor speed is used to allow the friction levels to stabilize. The second minute is then used to capture the torque value over a 1 minute duration before moving onto the next speed increment. The motor speed follows the same speed increments found in Table 4 and is repeated 3 times to generate an average torque value for each speed.

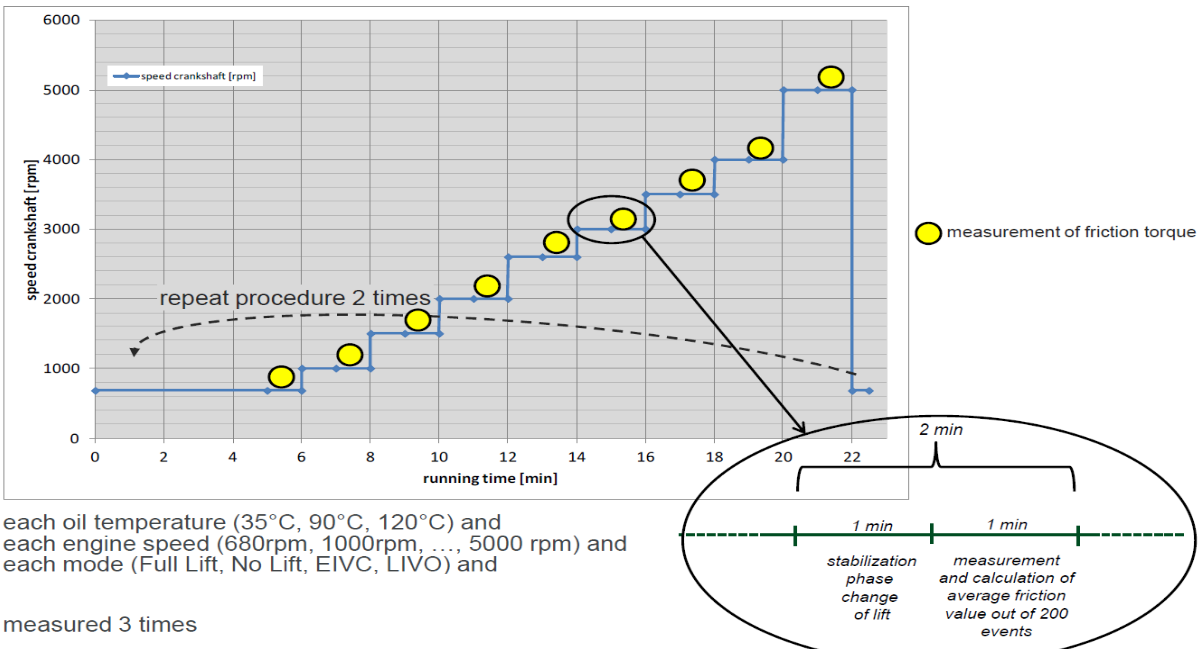


**FIGURE 28 - TEST PROCEDURE FOR MEASURING TORQUE AT VARIOUS CONDITIONS**

It was found that during ‘speed ramp’ up at 90°C using full lift of the UniAir that there were regular spikes of torque (friction) as engine speed increases. This can be seen in Figure 29.

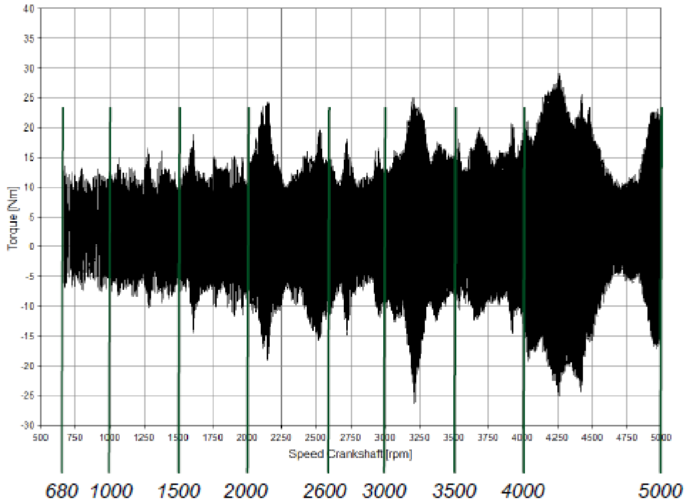


FIGURE 29 - SPEED RAMP AT 90°C, USING FULL LIFT OBSERVING TORQUE

Schaeffler gave no reason for the torque spikes, however, is more than likely due to the coupling of the shafts and changing speed. Figure 27 illustrates between the camshaft and motor is a bellow coupling. These couplings are subject to backlash when changing speeds as it encounters an increase in torque as stated in section 6.4.5.

Figure 30 displays that there was a running in cycle required before the friction torque measurements settled down to ±0.1Nm or less. It was concluded that for this test, capturing results should start after 90 hours running in time once the torque values were “constant” at ±0.1Nm. Checks were initially made at 167 hours and 204 hours and found that the friction levels were still constant.

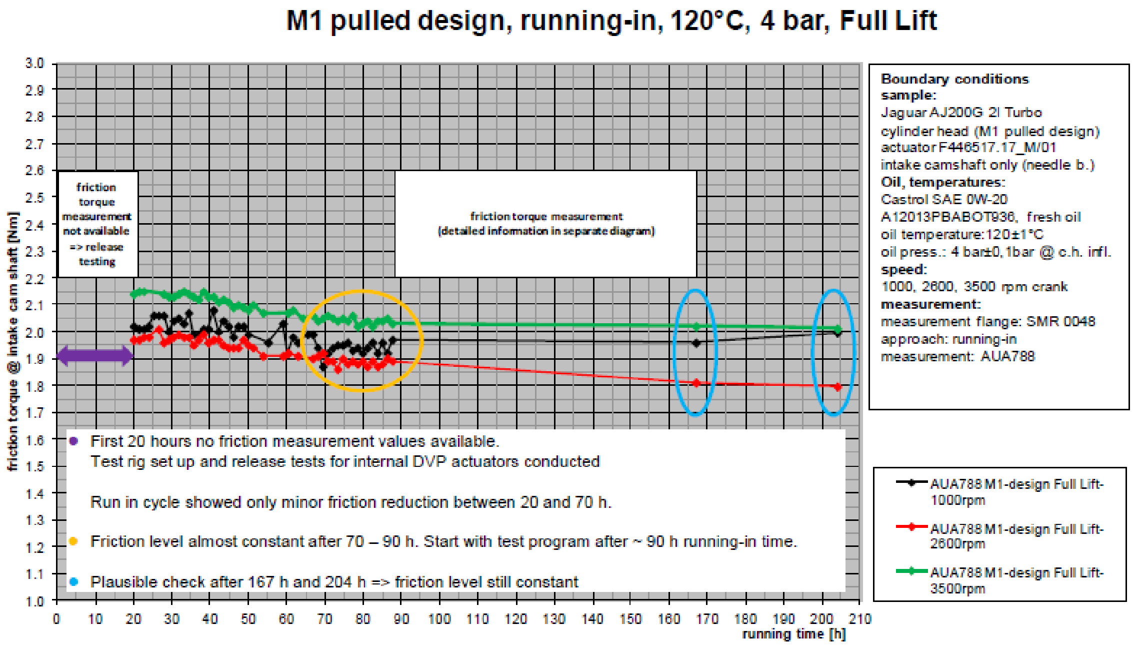


FIGURE 30 - RUNNING IN TIME OF THE AJ200 ENGINE AT 120°C, 4 BAR, FULL LIFT

### 4.1.1 Test results

To understand the characteristics and performance of the UniAir, the results will be analysed by looking at the torque levels for each different lift mode under various different boundary conditions.

#### 4.1.1.1 Full Lift and No Lift

Figure 31 displays the test results for torque levels using Full Lift and No Lift modes while varying camshaft speeds, oil temperature and using the pressure map found in Table 3. It was found that the torque levels for Full Lift changed when varying camshaft speed. This is assumed to be due to the relationship highlighted in Figure 24 with the Stribeck curve, as the lubrication regime changes due to change in speed. This is further illustrated by the torque levels when the oil temperature is at 35°C. At low speeds the torque levels at 35°C are lowest in the Full Lift mode. However, when increasing the speed, the torque levels at 35°C oil temperature are the highest. This is most likely due to the extra energy required to shear the higher viscosity oil at this temperature. Alternatively, when the oil temperature is at 120°C at low speeds, the torque levels are highest. This is assumed to be because of an incomplete fluid film formation as the oil viscosity is not thick enough to fully separate the mechanical interfaces.

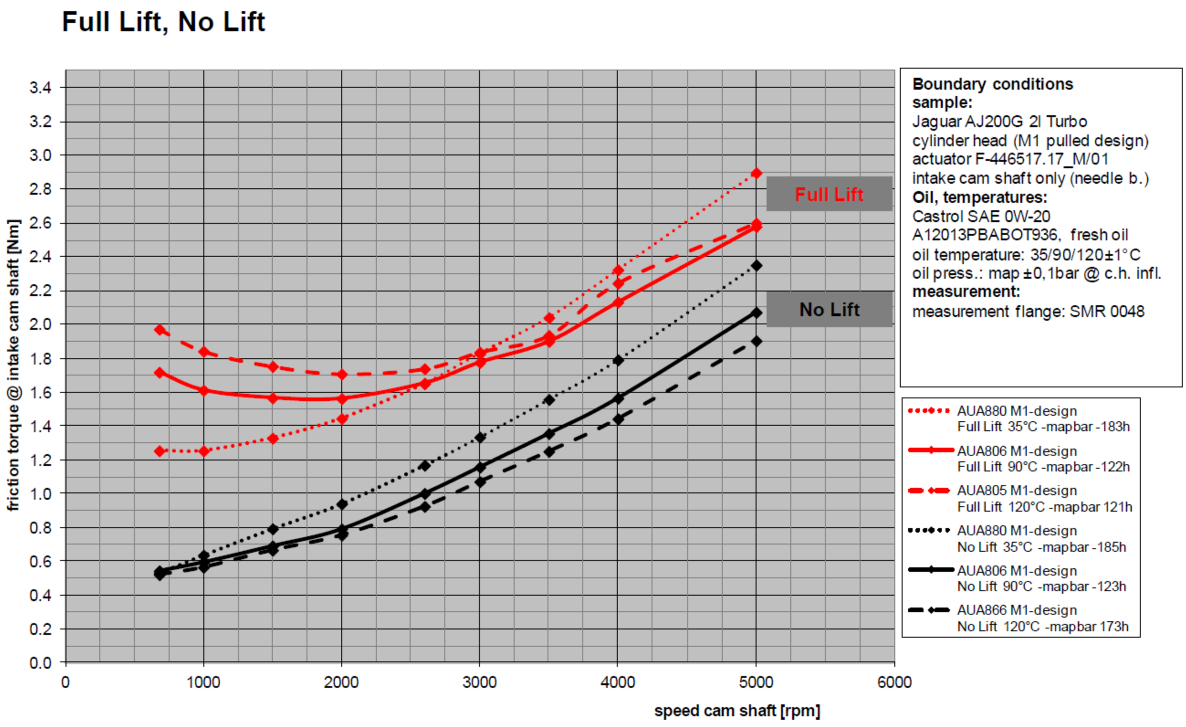


**Figure 31 - Full Lift and No Lift torque values over different temperatures and speeds**

When observing the torque levels for No Lift, Figure 31 demonstrates that the torque levels for 35°C are higher at all speeds. This is assumed to be because there is no energy required to move the valves putting extra emphasis that the torque is due to moving a higher viscosity oil in the UniAir. However as expected, the levels of torque are higher for Full Lift mode as all parts of the UniAir are used to produce valve lift.

### 4.1.1.2 EIVC

It was found from the testing shown Figure 32 that the highest overall levels of friction of approximately 4.3 Nm occurred during the early intake valve closing valve lift cycle at 35°C oil temperature.

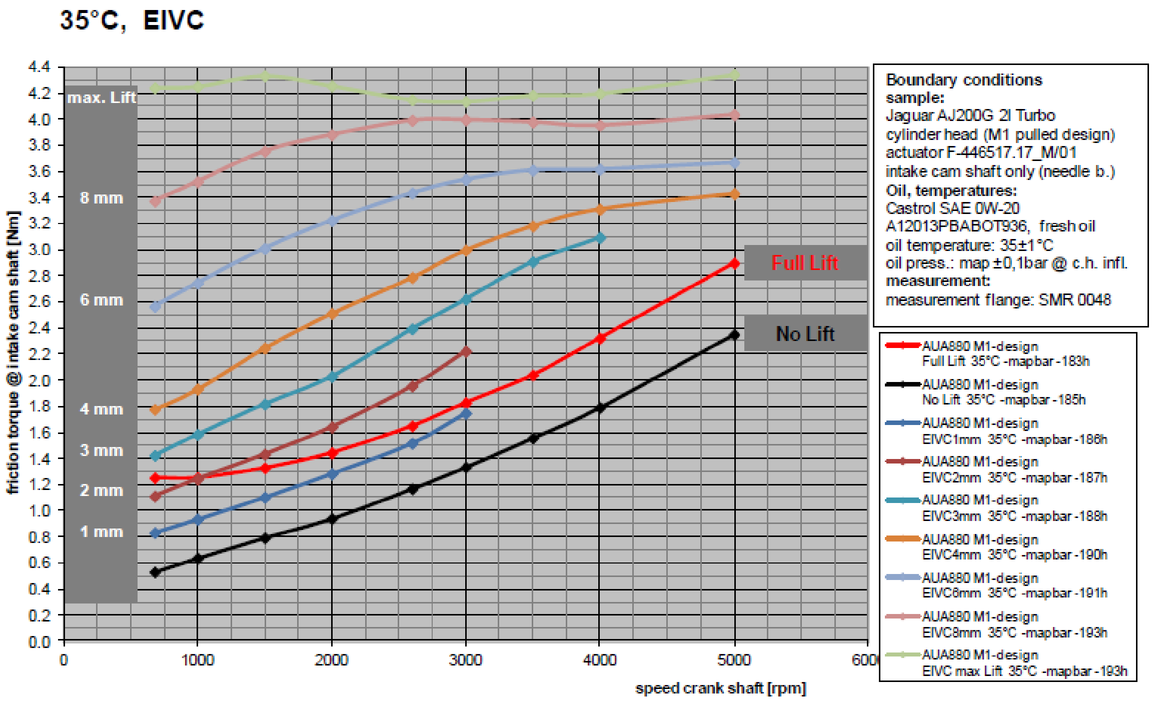


**FIGURE 32 - FRICTION LEVELS OF SCHAEFFLER'S VALVETRAIN TEST RIG AT 35°C DURING EIVC LIFT MODE**

EIVC creates the highest levels of torque because the energy used to create partial lift is entirely lost when the solenoid opens, releasing the oil pressure back into the intermediate pressure chamber. Section 2.4.1.1.3 states that the ECU specifies the intake valves closing point (CA°). The valve opening point is fixed as a result of the cam timing. When comparing this to Full Lift mode; the solenoid remains closed throughout the lift phase of the cam profile. Once the cam has passed the peak lift, the valve spring pushes back onto the cam helping the camshaft rotate and thus reducing overall torque. Figure 32 highlights this further as when higher levels of lift in EIVC mode are used, higher levels of torque are produced. More energy is required to open the valve further, but none of the energy is recouped as the valve spring is decoupled with the camshaft.

### 4.1.1.3 LIVO

It was found when using LIVO Lift mode that the torque levels are similar to that of Full Lift for the same oil temperatures. Figure 33 demonstrates the highest torque levels in LIVO Lift mode were found to be at 35°C oil inlet temperature.

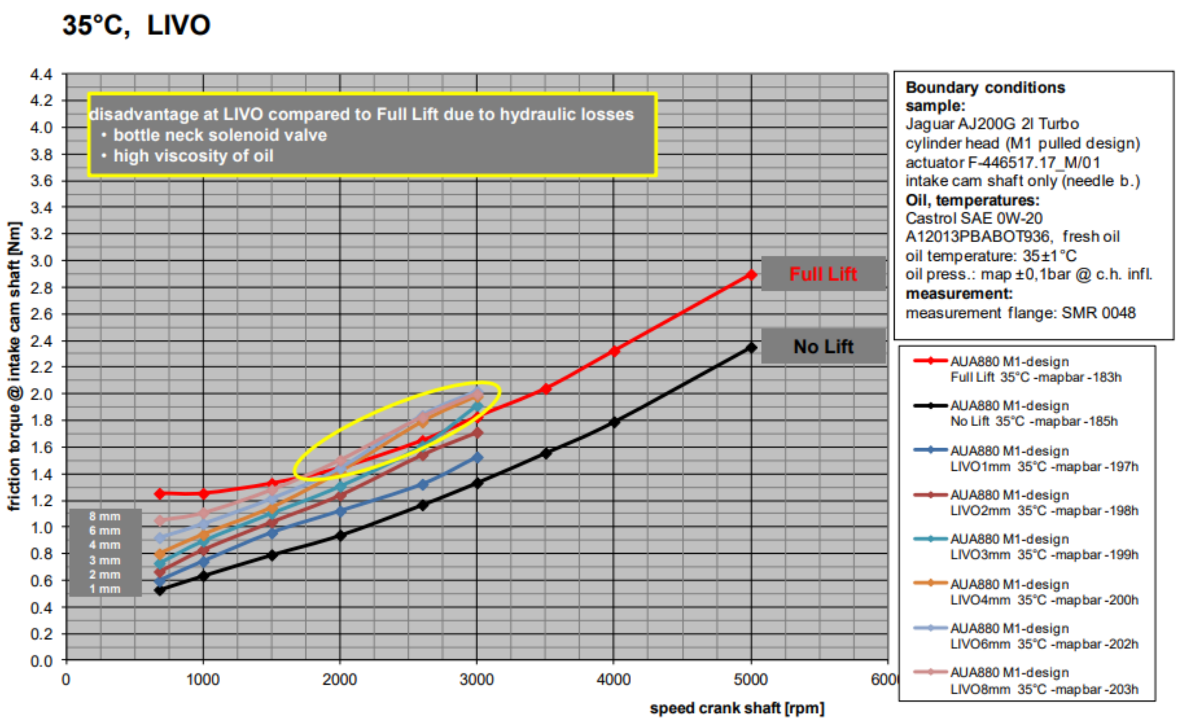


**FIGURE 33 - FRICTION LEVELS OF SCHAEFFLER'S VALVETRAIN TEST RIG AT 35°C DURING LIVO LIFT MODE**

Section 2.4.1.1.4states the ECU specifies the intake valves opening point (CA°). The valve closing point is fixed as a result of the cam profile. This further explains when the torque levels are similar to Full Lift, as the valve spring force is used to help rotate the camshaft as it follows the cam profile. However, Schaeffler had specified that the higher levels of torque at this oil temperature caused bottle necking in the solenoid valve due to the viscosity of the oil. Furthermore, the maximum speed LIVO is used is at 1500rpm camshaft speed, or 3000rpm engine speed as illustrated in Figure 33.

### 4.1.2 Discussion of results & impact on current work

Schaeffler has identified ways to measure frictional forces, running in time and expected frictional peaks throughout a similar UniAir system. The objectives in 1.3 state that to validate the valvetrain test rig in this thesis, comparisons with previous experimental and simulated tests is done. The test rig and results in 4.1 are highly relevant as it is testing a similar non-production prototype UniAir and can be used to help validate and to specify equipment for the valvetrain test rig in this thesis. It is expected that the results from 4.1 will differ to the results in this thesis because Schaeffler used a non-production UniAir opposed to a production UniAir. However, as section 3.5 stipulates, to generate compliant results, a tolerance is required. Schaeffler have not specifically mentioned any tolerance or standard uncertainty values from their results, regardless of demonstrating the test procedure had tested each point three times. The only mention of a value resembling a tolerance is mentioned in (section 4.1) Figure 30, where results were captured once the torque values were "constant" at ±0.1Nm.

Schaeffler tested the UniAir with SAE 0W-20 oil, at temperatures between 35°C and 120°C in an ambient environment. In addition to this, Figure 27 shows the 'tempering fluid' which was not measured and was purely used for regulation of cylinder head temperature. Therefore, this rig does not take into consideration any internal heat generated from the UniAir when in operation, and any heat lost from the measurement point, which is externally from the UniAir shown in Figure 27. Section 1.1 specifies this thesis is to study the influence of oil viscosity on the UniAir operation while varying oil formulation at oil and environmental temperatures from 0°C to 120°C. Therefore, further testing is required to understand the performance of the UniAir system under JLR required temperature ranges of 0°C to +120°C temperatures. Section 3.3.1 highlights that temperature is a critical parameter in viscosity, a more comprehensive temperature measurement system is therefore needed to understand the actual temperature in the UniAir.

## 4.2 JLR SIMULATED TESTS

An internal simulated study from JLR was investigated to see the influence of the engine oil viscosity on valve lift in an AJ200 Jaguar engine. The simulated study used engine oil SAE 0W-20 while varying other quantities:

- Camshaft speed – 375, 1500, 3100 rpm
- Lift modes – Full lift and EVC
- Oil temperatures – 40, 60, 90 and 120°C

### 4.2.1 TEST RESULTS

Figure 34 shows the simulated valve lift at two different camshaft speeds while varying oil temperature. When the camshaft speed is at 375rpm (Figure 34 a), it can be seen that when increasing the oil temperature, valve lift is reduced. This is in comparison to when the camshaft speed is at 3100rpm (Figure 34 b), the effect of oil temperature has a smaller reduction on valve lift. Additionally, the increase in camshaft speed has increased the overall valve lift height across all temperatures when comparing to the max lift at 40°C at 375rpm. This suggests that the influence of oil temperature on the performance of the system is reduced at higher camshaft speeds.

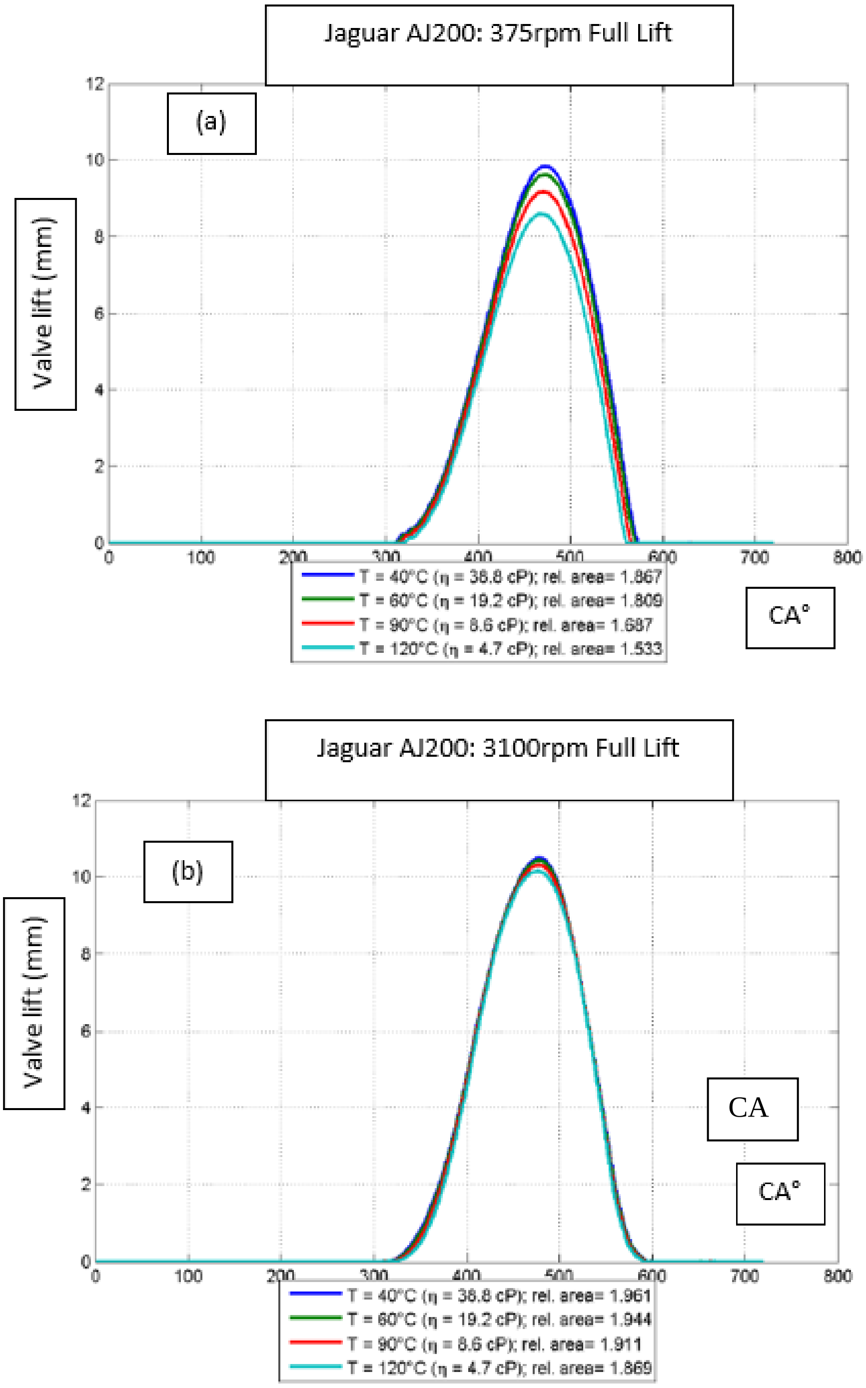


**FIGURE 34 - SIMULATED FULL LIFT AT 375RPM (A) AND 3100RPM (B) USING SAE 0W-20 ENGINE OIL**

When changing the lift mode to EIVC, Figure 35 shows the effects of oil temperature and camshaft speed on valve lift. Comparably to full lift, the effects of oil temperature at 375rpm show that with an increasing oil temperature, valve lift is reduced. Also, with an increase in camshaft speed at 1500rpm, the overall valve lift height increases while the oil temperature effects are reduced. However, when increasing the speed to 3100rpm not only the valve lift height drops significantly, but the effects of oil temperature are reversed. The increased oil temperature increases the valve lift, while lower temperatures reduce the valve lift.

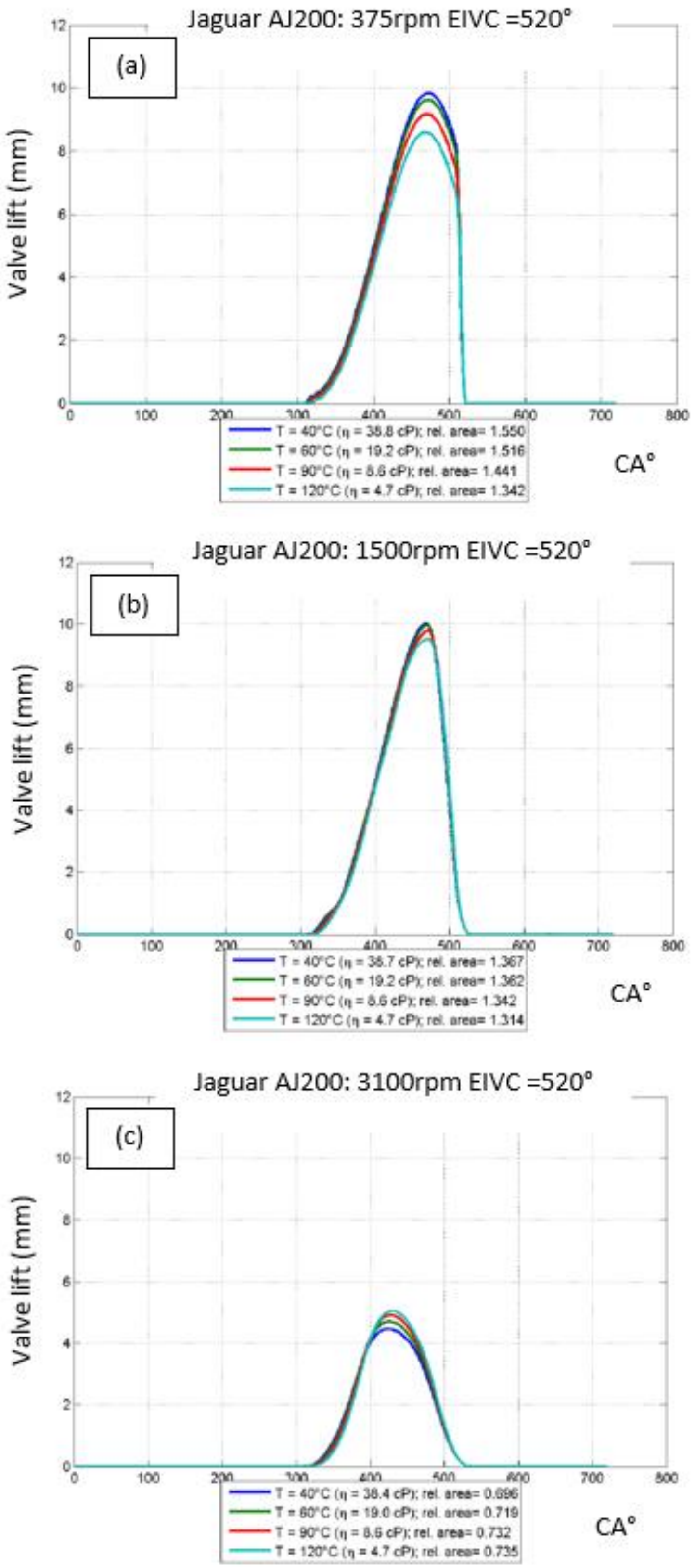


**FIGURE 35- SIMULATED VALVE LIFT OF EIVC AT 520 CRANK ANGLES AT 375RPM (A), 1500RPM (B) AND 3100RPM (C) USING 0W20 OIL**

### 4.2.2 DISCUSSION OF RESULTS & IMPACT ON CURRENT WORK

This simulated study shows how camshaft speed, oil temperature and lift mode have an effect on valve lift. This gives further understanding and evidence to how the oil viscosity influences the UniAir operation and efficiency. However, this is only a simulated study so in order to gain a better understanding of UniAir operation and efficiencies, experimental data is needed. Furthermore, these simulated tests are limited to 40°C to 120°C oil temperatures using a 0W20 oil grade. Whereas the scope for this project is to vary oil formulation and oil temperatures from 0°C to 120°C and environmental temperatures varying from 0°C to 120°C using climate chambers. Additionally, with valve lift changing with oil viscosity, to understand further the impact of lower viscosity fluids to optimise JLR oil specifications is desired.

As stated in Section 4.1, one of the objectives (Section 1.3), is to validate the valvetrain test rig in this thesis, by comparing with previous experimental and simulated tests. In this simulated test, it provides a limited set of data helping understand the performance of valve lift in Full lift and EIVC. However, there is no data for LIVO lift mode, therefore further testing is required. Additionally, there is no mention of lift tolerances or standard uncertainty in the simulated tests.

# 5. LITERATURE REVIEW

A number of different techniques have been used to understand the frictional operation and efficiency of valvetrains. It is therefore important to understand and consider the standard practice when designing the rig. However, because the UniAir is an electrohydraulic valvetrain, there are further parameters which are not found in traditional valvetrain designs and must be considered. These include, but are not limited to, the frictional operation, performance, and efficiency of pumps to understand the effects of the lubricants on the UniAir.

## 5.1 MOTORED VS FIRED ENGINE VALVETRAIN TEST RIGS

Kerres et al. states that a motored test rig offers possibilities to reproduce the behaviour of an engine and avoiding cost and time intensive development loops. Boundary conditions such as oil pressure and temperature can be adjusted to replicate working conditions of the engine. However other parameters such as the influence of combustion gas pressure on valve head during the combustion cycle, and representative formation of tribofilms in the valve-seat contact are not considered. Motored test rigs are also considered when the availability of test parts is an issue. With regards to this, two different setups can be considered (19):

- Individual valve test rigs (can also include multiple valves, just not considered as a combined cylinder head and valvetrain system)
- Cylinder head test rigs

Individual valve test rigs are typically used in the concept phase of development however can offer a very cost-effective insight to the performance of valve lift with a given valvetrain. However, in contrast to the single valvetrain test rig, the cylinder head test rigs offer the ability to demonstrate the entire valvetrain, while getting closer to the replication of an engine. This also then allows the potential to add chain drives, chain tensioners, gears, guides etc (19) to attempt to replicate the dynamic load on the valve train. To take measurements, the cylinder head is pressurized with oil to replicate the working conditions of the engine. The oil temperature and formulation can be adjusted depending on requirements.

Sandoval and Heywood compared mathematical models to estimate individual friction values of valvetrain components to engine test data. They state that measuring friction via a motor makes it easier to obtain the direct measurement of friction on individual components. However, it is also stated that motoring test rig friction differs from fired engine test rigs due to the differences in gas pressure loading on the piston and valves (20).

| Investigation and evaluation of: | Motored | Fired |
|---|---|---|
| Kinematic and dynamic behaviour of valve lift, valve speed and acceleration vs engine speed | Yes | Yes |
| Temperature influence on kinematic and dynamic valve lift, valve speed and acceleration | Limited feasibility | Yes |
| Engine speed and load and drive train influence | - | Yes |
| Real cam timing vs engine speed and load | - | Yes |
| Valve lash during engine durability test | - | Yes |
| Cam and crank angle oscillations with the influence of drive train | Limited feasibility | Yes |
| Analysis of fully variable valvetrains with regard to the engine map | - | Yes |
| Influence of cylinder pressure: | | |
| • Negative valve lift due to valve head deformation caused by cylinder pressure (indication for stress in valve head) | - | Yes |
| • Possible exhaust valve openings due to gas dynamic effects of exhaust system (eg turbo applications) | - | Yes |

**TABLE 5 - PARAMETERS OF MOTORED AND FIRED VALVETRAIN INVESTIGATIONS *(19)***

Kerres et al. agree with this finding, suggesting that although results from motored valvetrain tests do simulate closely to reality, fired engines feature a specific behaviour that cannot be replicated on a motored test rig. The temperature parameter can be adjusted by means of controlling oil temperature on a motored test rig, but the influence of combustion temperatures cannot be replaced. Other parameters such as cam and crank angle oscillation are not easily replicated on motored test rigs. In addition to electric motors having other dynamic behaviours compared to the crankshaft of a fired engine (19).

Table 5 gives an overview of the different parameters that have an influence on valve lift and compare if motored and fired engine tests rigs can achieve them.

Alternatively, Blaxill et al. studied the frictional levels attributed to individual engine systems using both fired and motored rigs. Friction measurements were taken with oil temperatures of both 40℃ and 90℃. It was concluded that friction should not be considered as only a subsystem problem and cannot be properly optimised by simply revising single components. Consideration of the entire engine is recommended as it was found temperature had a significant influence on measured frictional values (21).

However as recognised by Kerres et al., a rig design will be governed by the parts available. In the case of this project, JLR only supplied the cylinder head, this meant the rig would have to be a motored cylinder head test rig.

## 5.2 FRICTION TORQUE MEASUREMENTS

One of the main objectives in the research presented in this thesis was to evaluate the influence of the viscosity of a lubricant, by monitoring the frictional operation and efficiency of the hydraulic valvetrain. Section 4.1 shows Schaeffler's method of measuring the frictional operation and efficiency of the UniAir by capturing friction torque values using a torque transducer between the electric motor and cylinder head. There is numerous motored cylinder head valvetrain friction studies, which all incorporate a torque transducer between the electric motor and cylinder head (21–27).

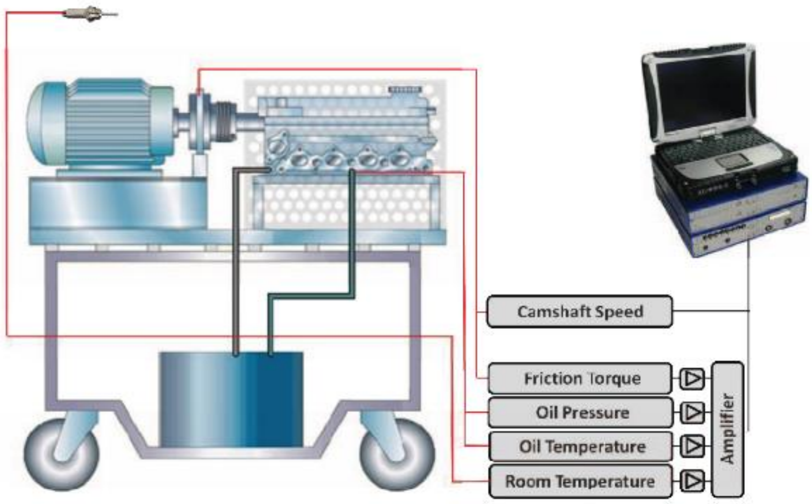


**FIGURE 36 - TEST STAND LAYOUT OF A VALVETRAIN FRICTION TEST *(23)***

All of these tests are monitoring friction torque on a traditional valvetrains, not the UniAir, which has multiple pumping mechanisms and hydraulic power. However, Inaguma et al studied the friction torque characteristics of an internal gear pump. The drive shaft of the pump was rotated by an electric motor and a torque meter was equipped between the electric motor and the driving shaft to the pump, agreeing with the method to capture torque in the traditional valvetrains rigs (28,29).

## 5.3 MOTORED CYLINDER HEAD VALVETRAIN FRICTION

In the UniAir system the intake valves are not controlled directly by the intake camshaft, but indirectly via intermediate hydraulic chambers. However because this hydraulic power is driven by a conventional camshaft and roller finger follower as seen in Figure 13, it is important to understand the frictional characteristics of a more traditional valvetrain test rig.

### 5.3.1 TRADITIONAL VALVETRAIN FRICTION TORQUE

Kovach et al explain that the friction torque in traditional camshafts, were predominantly at the cam-tappet interface and spring loads. They tested various different camshaft types to further understand the frictional torque (30).

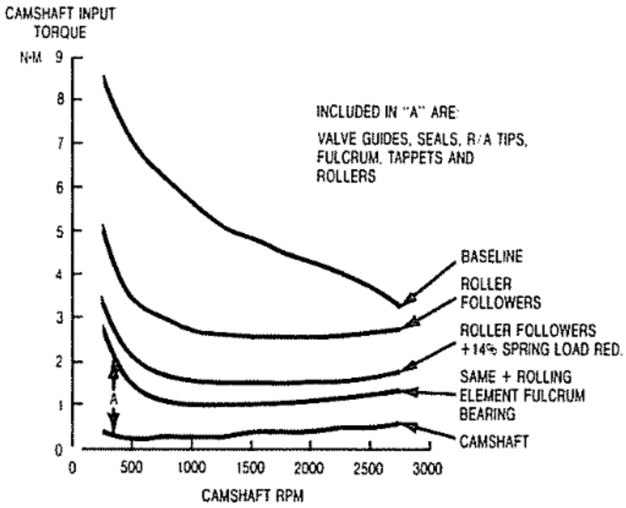


**FIGURE 37 - FRICTIONAL TORQUE OF VARIOUS DIFFERENT VALVETRAIN TYPES*(30)***

With reference to the Stribeck curve (Section 3.3), the effect of friction torque is greater at lower speeds (which can be seen in Figure 37) when the camshaft is in the boundary lubrication condition (21)(30). This high friction is theorized due to incomplete fluid film formation. The entraining velocity at lower speeds does not form sufficiently thick oil films to fully separate the cam and follower, and journal bearings (23). As the engine speed increases the lubrication regime changes to mixed where the friction levels drop considerably before rising again when in the hydrodynamic lubrication region.

The UniAir is actuated by a roller finger follower type camshaft, where in Figure 37 shows a lower friction level than its direct acting camshaft counterpart. This agrees with Brown et al who also looked at the different camshaft interfaces. Test results showed significant benefit of rolling interfaces in comparison to sliding interfaces, by having up to 82% reduced friction torque at certain rpms (27,30).

Additionally, Figure 37 illustrates the friction torque of a roller follower interface with a reduced spring load, this result indicates that lower spring loads decrease valve train friction (30). Brown et al also agree with this as they concluded the magnitude of valvetrain friction is directly related to valve spring force (27). Understanding the behaviour of both the roller follower and spring load friction characteristics will help determine the influence on the CVVL valve train for this rig.

### 5.3.2 CVVL Valvetrain Friction

Kyoung-Pyo et al investigated the friction torque of a CVVL mechanism from Hyundai Motor Company, which also used the roller follower interface as seen in Figure 38.

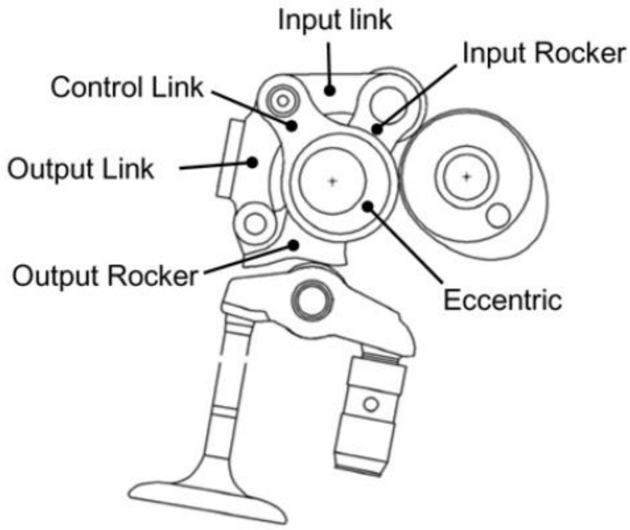


**FIGURE 38 - SCHEMATIC DIAGRAM OF HYUNDAI MOTOR COMPANY CVVL*(31)***

The CVVL was developed based on a fixed valve profile engine by changing only the valvetrain (31). Friction tests were done on the 'base' valvetrain and then using the same cylinder head specifications the CVVL valvetrain was tested. Therefore, the friction torque of the CVVL can be directly compared to the base valvetrain. Three major factors of friction were looked at load (valve lifts), oil viscosity (oil temperature) and velocity (engine speed). Figure 39 shows two oil temperatures tested on both valvetrains while varying the valve lifts on the CVVL.

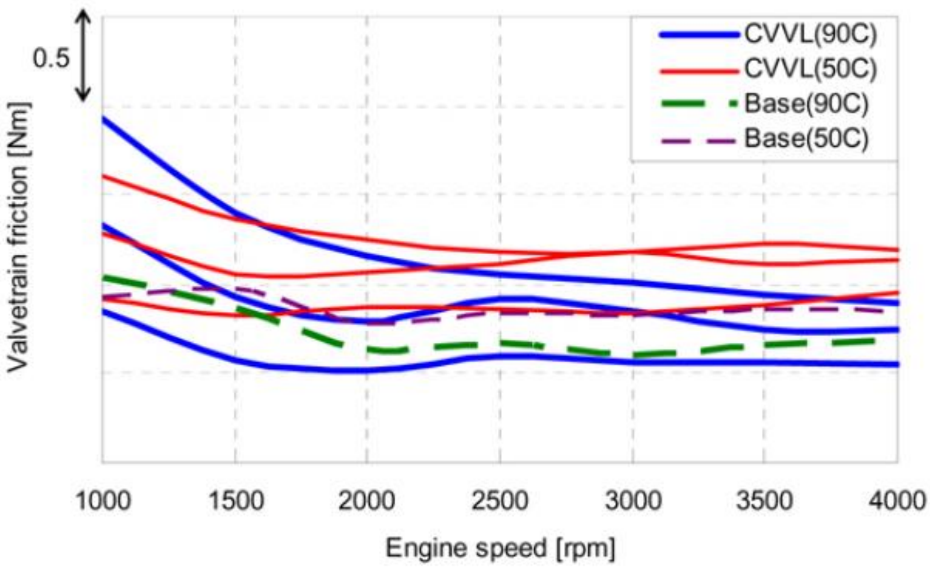


**FIGURE 39 - CVVL AND BASE VALVETRAIN FRICTION OF HYUNDAI MOTOR COMPANY *(31)***

Three different valve lifts were implemented for the CVVL test, 1mm, 6.1mm and 11mm. At low speeds the friction torque of the different valve lifts increased with increasing valve lift height for the relevant temperature. It is explained that the friction of the CVVL at 1mm shows comparable behaviour with that of the base engine in Figure 39, which implies that they are both in a similar lubrication regime. However, the friction for the 11mm valve lift shows signs that the lubrication is dominated by boundary lubrication; this indicates that the metal to metal contacts between the linkages of the CVVL mechanism causes the high friction. In summary, the CVVL has higher levels of friction overall, however the benefit of the flexible valve profile becomes more evident in the fuel consumption (31).

### 5.3.3 Effects of oil temperature on friction torque of valvetrains

Another motored valvetrain friction test was investigated by Leme et al, who investigated the frictional losses of a roller follower valvetrain while varying oil temperature, pressure and spring loads. 32 test conditions were done varying oil pressure, temperature and spring loads. Each varied test was repeated 4 times.

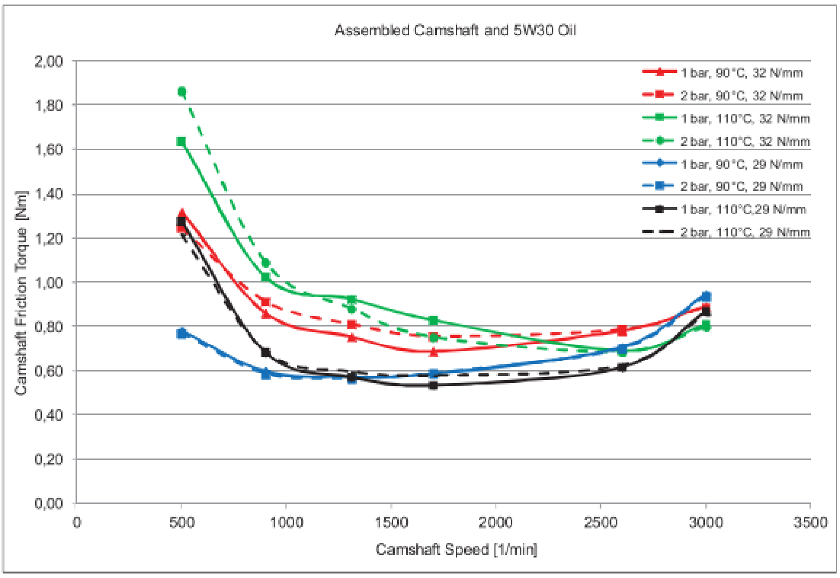


**FIGURE 40 - FRICTION TORQUE RESULTS *(23)***

Figure 40 shows the friction torque results with varied inputs to the test rig. Results found that higher levels of friction were observed at lower camshaft speeds also seen in Figure 37 and Figure 39. Kyoung-Pyo et al similarly found that the higher temperatures created higher levels of friction torque at these speeds. This is due to the oil film thickness which is directly related to the viscosity of the oil. The higher the viscosity, the thicker the oil for a given clearance and entrainment velocity. As the camshaft speed increases, the friction torque drops as the lubrication regime changes, however the friction torque of the higher temperatures decreases further. This is due to the vicious drag caused by the higher viscosity of the oil at 90°C, this is illustrated in Figure 40 as the point at which the torque levels cross for the same valve spring stiffness at different temperatures. Likewise, Figure 40 agrees with Kovach et all that the higher spring stiffness is directly related to higher friction torque.

All of the valvetrain friction studies look at varying oil temperatures typically found when the engines are at peak operational temperatures. Subsequently there are minimal studies of valvetrain friction forces apparent at cold temperatures.

## 5.4 VALVE LIFT

To understand the performance of the UniAir system characterisation of valve lift is required. This is because the UniAir creates a hydraulic link between camshaft and valve. Therefore, monitoring the valve lift profiles under all the different simulated conditions of the engine, will give a vital understanding of how the influence of the viscosity of lubricants effect the operation and efficiency of UniAir.

Kerres et al. investigated various valvetrain test rigs measuring valve lift, to give an overview of differences between motored cylinder head test rigs and fired engine test rigs. On motored test rigs Kerres et al and Calabretta et al. suggested using Laser Doppler Vibrometer (LDV) to measure the valve lift. This is only possible because of the access to underneath the cylinder head so the valve heads can be monitored. Kerres et al. states that using non-contact lasers allows measurements of isolated movement of the valve, without the influence of sprung mass, which allows for more accurate readings of valve lift (19) (24). In the case of this project, this would create more realistic analysis of the valve displacement as the valves are untampered.

The valves in fired engine tests are encased in the engine, therefore an alternative way to measure the valve lift is needed. Kerres et al. suggests using magneto resistive sensors (MRS). In order to measure valve lift, the valve must have a tooth structure machined directly to the valve stem. The MRS is machined and embedded into the valve guide, so as the valve passes by the sensor an external magnetic field is generated. This leads to a change of the sensors electrical resistance which has a linear relationship with the displacement of the valve. In addition to the machining of the valve stem, the steam needs to have a ferromagnetic layer in order to generate the magnetic field. If not, a coating layer can be applied to the valve stem. The MRS and LDV sensors were directly compared for valve lift by measuring the same valve at the same time. It was found to have differences of less than 0.0005mm, Kerres et al. state that the difference was so small that it would not influence the test results (19).

The valve lift was then measured in a motored and fired engine test. Valve clearance of the exhaust valve changed by 0.15mm due to thermal influences of combustion, however there were no differences in the inlet valve clearance. Figure 41 shows the valve lift of the motored engine test rig and fired engine test rig. It can be seen that there is only a marginal difference in valve lift between the two tests.

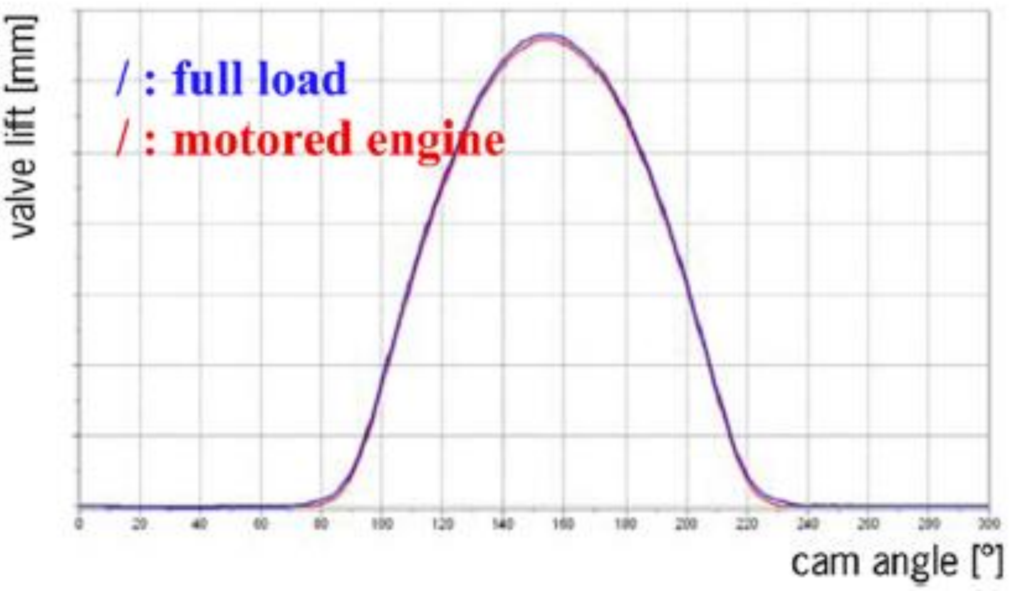


**FIGURE 41 - DIFFERENT VALVE LIFT CURVES COMPARING MOTORED ENGINE TO FIRED ENGINE TEST RIGS. *(19)***

In the case of this project, JLR only supplied the cylinder head, this gives access to the valve heads. LDV is a less complicated method of measuring valve lift, while also having similar accuracy levels of measurement.

## 5.5 Oil

The UniAir system is entirely dependent on the engine oil as the oil is the link between camshaft and valve stem and is therefore the most important parameter in understanding the operational performance and efficiency. This section will look into the different segments of the oil, which will help the design of the rig and analysis of the test results.

### 5.5.1 Hydraulic efficiency

There are two factors affecting hydraulic efficiency, mechanical efficiency and volumetric efficiency. Mechanical efficiency is dependent on the frictional losses within the device and the energy required to generate flow through it (32)(33).

$$n_m = \frac{T_{th}}{T} \quad (12)$$

Where:

$n_m$ is mechanical efficiency

$T_{th} = Theoretical\ Torque$

$T = Measured\ Torque$

Volumetric efficiency is the ratio of actual flow to theoretical flow, or flow losses often due to leakage (33).

$$n_v = \frac{Q}{Q_{th}} \tag{13}$$

Where:

$n_v$ is volumetric efficiency

$Q_{th} = Theoretical\ Flow$

$Q = Measured\ Flow$

Figure 42 shows how the mechanical and volumetric efficiency varies with viscosity and how dependent the hydraulic efficiency is on the viscosity of the fluid. The overall efficiency $(n)$ is the product of mechanical and volumetric efficiencies.

$$n = n_m n_v \tag{14}$$

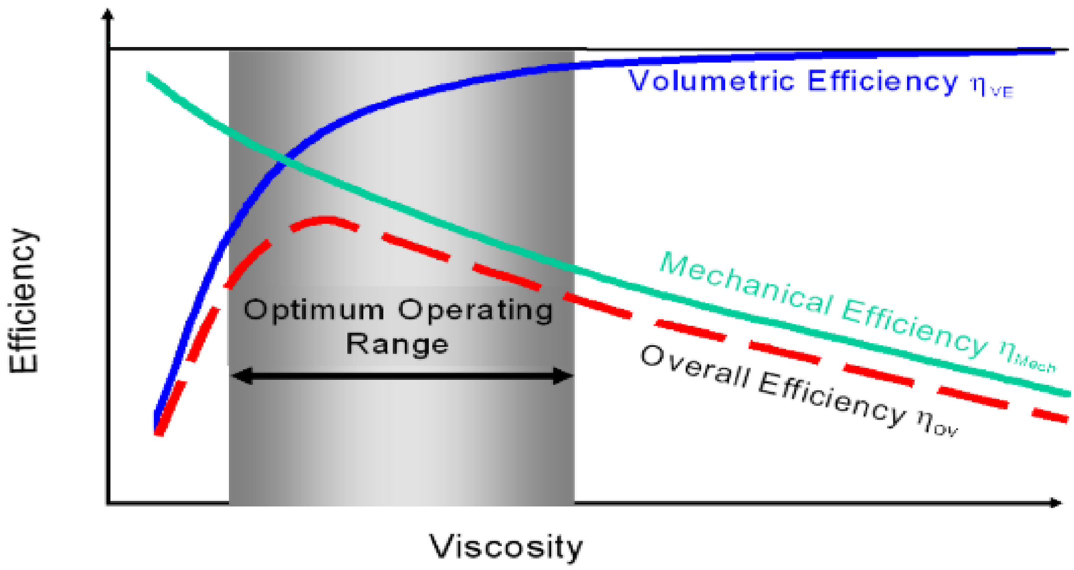


**FIGURE 42 - THEORETICAL HYDRAULIC PUMP EFFICIENCY CURVES *(32)***

At low temperatures when viscosity is high, this negatively affects the mechanical efficiency of the hydraulic system (in addition to higher shear strength) by creating lubricant starvation and cavitation. Viscosity influences cavitation because the high viscosity fluids can create excessive pressure drop at the pump inlet. Cavitation causes metal fatigue and spalling which adds to overall friction values (32,33).

Like any other lubricating fluid (as discussed in Section 3.3), hydraulic fluids also required to provide a lubricating film between surfaces. This film effectiveness depends upon a balance between viscosity, sliding speeds/loads and fluid stability within a hydraulic pump. As temperatures increase the film thins causing the lubricant film to rupture allowing metal to metal contact. As a result of this it generates higher levels of friction and wear within the pump and additional fluid heating because the boundary layer is not strong enough between moving surfaces (33,34).

Wear predominantly occurs in locations within a pump that are critical to volumetric efficiency. Loss of the volumetric efficiency is produced by high temperature low viscosity lubricants bypassing pump clearances causing leakages. Therefore, both high and low temperature situations can cause detrimental effects on hydraulic efficiency (33,34).

#### 5.5.1.1 TEST PROCEDURE FOR EVALUATING HYDRAULIC EFFICIENCY

Koehler and Jackson have developed a test procedure to measure a fluids effect on mechanical and volumetric efficiency. The fluid performance has a direct effect on the efficiency of a hydraulic system (35).

The test procedure had each fluid evaluated for efficiency at 81 combinations of pump speed, outlet pressure load and temperature. This is highly relevant to this project, as this rig will also need to change all three variables during testing.

| Temperature [°C] | Pump Speed [rpm] | Pressure [bar] |
|---|---|---|
| 50<br>55<br>60<br>65<br>70<br>75<br>80<br>85<br>90 | 1200<br>1800<br>2400 | 103<br>155<br>207 |

**FIGURE 43 - KOEHLER AND JACKSON TEST CONDITIONS FOR TESTING HYDRAULIC EFFICIENCY**(35)

A pump was instrumented using a torque meter to measure the speed and torque required by the pump to calculate mechanical power, a flow meter and pressure transducers were used to determine the hydraulic power generated by the pump. Accuracy of the control of the test parameters is critical when measuring efficiency, therefore, the pump speed was controlled by using an electric motor and variable speed drive. The load pressure is controlled by a pressure relief valve at the pump outlet, while the measurement of the inlet temperature was accomplished using a K-type stainless steel thermocouple mounted closely to the pump inlet. By monitoring the required mechanical and hydraulic power to operate the system the hydraulic efficiency of the system can be determined. Subsequently the results found that higher viscosity characteristics greatly improved volumetric efficiency while lowering mechanical efficiency.

Another study by Inaguma investigated the friction torque characteristics of an internal gear pump while varying temperature and pressure (28). Comparably to Koehler and Jackson (35), the pump was operated by an electric motor with a torque transducer between the motor and pump. The inlet and outlet pressures were measured to find the pressure differential across the pump while measuring the oil temperature at the outlet. Figure 44 shows the test results of the friction torque of the internal gear pump while varying pressure, speed and temperature.

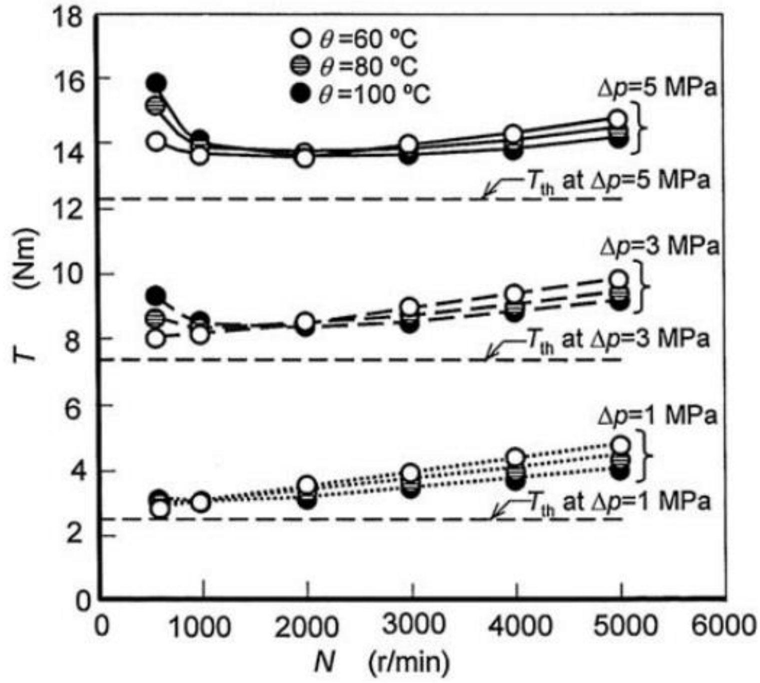


**FIGURE 44 - DRIVING TORQUE CHARACTERISTICS OF INTERNAL GEAR PUMP *(28)***

The pump driving torque was measured twice at various conditions of pump speed and then averaged. It was found that the biggest torque variation for the same test was 0.23Nm.

The mechanical efficiency was investigated by looking at the relationship between the pump speed and driving torque. It was found that pump speed has a substantial effect on the friction torque characteristics of the pump, especially in comparison to the theoretical pump friction torque $T_{th}$. However, the combination of different pump speed, pressure, and temperature all specifically changed the friction torque characteristics. Currently there is no theoretical friction torque for the UniAir, therefore an alternative solution may need to be considered.

Figure 44 illustrates at low pressures, the increase in pump speed causes higher levels of torque. As the pump speed is increased, the torque between different oil temperatures becomes more pronounced. The torque is lower at higher oil temperatures with increasing pump speed. This is explained as the lower oil temperatures cause higher levels of viscous drag resulting in higher levels of friction torque.

However, as the pressure increases, the friction torque characteristics between oil temperatures at low speeds change.

With reference to Figure 24 (Stribeck curve diagram) higher pressures emphasise the boundary lubrication at lower speeds due to a resulting smaller Somerfield number. The higher oil temperatures reduce the viscosity of the oil which subsequently reduces the oil film thickness. With a fixed low speed (or velocity), as the pressure (or load) increases the Somerfield number decreases. This increases the coefficient of friction which results in higher friction torque seen in Figure 44. In the high pressure situation, as the pump speed increases to approximately 1000-2000rpm, the Somerfield number increases. This causes the lubrication regime to change from boundary to mixed causing lower levels of friction torque. As the speed increases further (3000-6000rpm) the lubrication regime changes again (Hydrodynamic) which is signalled by the increase in friction torque.

This study by Inaguma considered a radial internal gear pump in comparison to the axial piston pump of the UniAir seen in Figure 13. There is limited studies on axial piston pump friction torque because they 'have a relatively low horsepower-to-weight ratio and high flow pulsation' (36). The main advantages of axial pumps are

- High pressure capabilities.
- Consistent volume displacement per revolution
- Reduced installation space

It is clear why the axial piston pump is used in the UniAir application due to these advantages, however due to the lack of research in axial piston pump friction torque this emphasises further the need for new test apparatus and associated methodology.

The literature for evaluating hydraulic efficiency is beneficial as it has identified ways to practically evaluate hydraulic efficiency of the system, however one area of interest that was not readily apparent in the literature is that of work involving the colder temperature range. Other studies have established that hydraulic pump efficiency is dependent on hydraulic fluid viscosity (20,32,37,38). The simulated and physical tested studies used fluid viscosities related close to the boundaries of working temperature. Sandoval and Heywood state that with oil viscosity scaling with temperature, resulted in their simulated cold engine friction prediction about twice the value for engines at their typical operating temperature (20). And that this agrees with 'the limited cold engine friction data available in the literature' (20). This gap suggests further research is required in understanding the performance for low temperature applications in valvetrain frictional performance particularly in oil dependant applications like the UniAir.

### 5.5.2 Aged Oils

Gangopadhyay et al studied the effect of oil ageing on valvetrain friction and wear. This was achieved by filling three identical vehicles filled with three different blends of 5W-20 SAE oils. All the engines were new (Ford Crown Victoria with 4.6L 2v engine), and the engine oils were drained and replaced at 3,000,5,000,7,500, 10,000 and 15,000 mile intervals. These oils were then evaluated for their friction and wear characteristics using a motored direct acting cam lobe on a bucket type valvetrain rig. The friction torque was measured by an in-line torque meter. The tests were conducted at three different camshaft speeds of 500rpm, 1000rpm and 1500rpm. Each test ran between 50-100 hours. The fresh oil was evaluated first followed by 3000, 5000, 7500, 10000 and 15000 mile drain samples. A different cam lobe, shim and tappet were used for each blend of oil; however the same cam lobe, shim and tappet were used for fresh oil and subsequent drain samples. When changing the oil, the lubricant was pumped out the entire system and flushed once with the next candidate drain oil (39).

The viscosity of the oils was measured for each oil drain interval.

| **Oil 1** | Fresh Oil | 3000 Miles | 5000 Miles | 7500 Miles | 10000 Miles | 15000 Miles |
|---|---|---|---|---|---|---|
| Viscosity (mm2/s) at $40^oC$ | 48.9 | 49.6 | 55.9 | 62.4 | 62.2 | 110.5 |
| Viscosity (mm2/s) at $100^oC$ | 8.4 | 8.3 | 9.1 | 9.9 | 10.4 | 15.9 |
| **Oil 2** | | | | | | |
| Viscosity (mm2/s) at $40^oC$ | 47.2 | 45.1 | 50.3 | 54.4 | 55.9 | 82.8 |
| Viscosity (mm2/s) at $100^oC$ | 8.7 | 8.3 | 9 | 9.5 | 9.7 | 13.3 |
| **Oil 3** | | | | | | |
| Viscosity (mm2/s) at $40^oC$ | 48.2 | 56.2 | 51.8 | 55.5 | 63.4 | 136 |
| Viscosity (mm2/s) at $100^oC$ | 8.3 | 9.5 | 9.1 | 9.5 | 10.5 | 17.1 |

**TABLE 6 - OIL VISCOSITIES OF THE THREE DIFFERENT OIL BLENDS AT EACH OIL DRAIN INTERVAL**

The oils show a correlation between increasing viscosity with increasing miles. The conclusions of the study are the wear and friction torque of all oils reduced as the oil ageing increased. The friction reduction was in the range of 8-22%. However, the viscosity of the drain oils increased with ageing, therefore suggesting that the reduction in wear and friction is related to changes in oil chemistry due to ageing (39).

This study suggests that the oil ageing will show a reduction in friction torque values for contacts around the camshaft in the AJ200 JLR cylinder head. However, as seen in Schaeffler's UniAir test rig, it is expected that the hydraulic pumping of the UniAir will generate the most losses due to friction in the system. With an increasing viscosity due to oil ageing, it is therefore expected that the overall frictional torque will increase in the UniAir valvetrain test rig.

### 5.5.3 OIL AERATION AND SPEED INFLUENCES

Koch et al studied oil aeration in combustion engines. It is explained that dissolved gasses have an influence on the lubricant, especially on compressibility behaviour. Part of the study looks at the effects of oil aeration on hydraulic lash adjusters, which are used in their conventional manner in the UniAir system. A hydraulic lash adjuster is a small piston and housing, which includes a high pressure space filled with oil. Its function is to automatically adjust the valve lash ensuring precise valve seating, optimized air flow and precise fuel management. Koch et al go on to describe that if the high pressure space is supplied with dissolved aerated oil, that the oil may not consist of the right properties to provide the required rigidity of the hydraulic system. "The extreme case in this situation is a failure of the valve mechanism resulting in extreme component stresses due to the high touch down speed of the valves"(40).This needs to be considered for the UniAir system as it could be thought to be a scaled up hydraulic lash adjuster. This could potentially cause a lack of pressure or not enough rigidity to fully open the valve, causing a reduced valve lift.(40)

Koch et al explains that oil aeration increases with increasing speed as illustrated in Figure 45. (40)

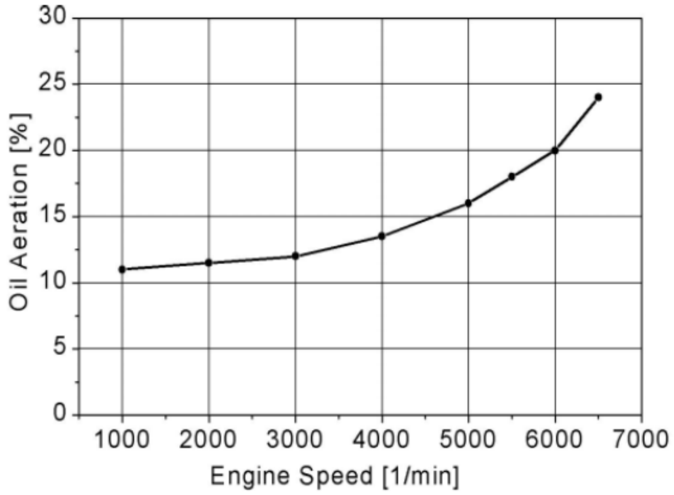


**FIGURE 45 - OIL AERATION AT DIFFERENT ENGINE SPEEDS *(40)***

It is identified that the increase in engine speed increases the oil flow, which subsequently has a shorter time for air bubbles to escape from the oil while being pumped. This may be a problem for this rig because there will be a variation of different speeds. Therefore, the oil aeration may be inconsistent at different speeds.

Theisen et al explains there is a limit to the amount of air that can be dissolved into oil(41). There are three different forms of gas in engine oil as illustrated in Figure 46:

- Dissolved gas
- Dispersed gas
- Surface foam

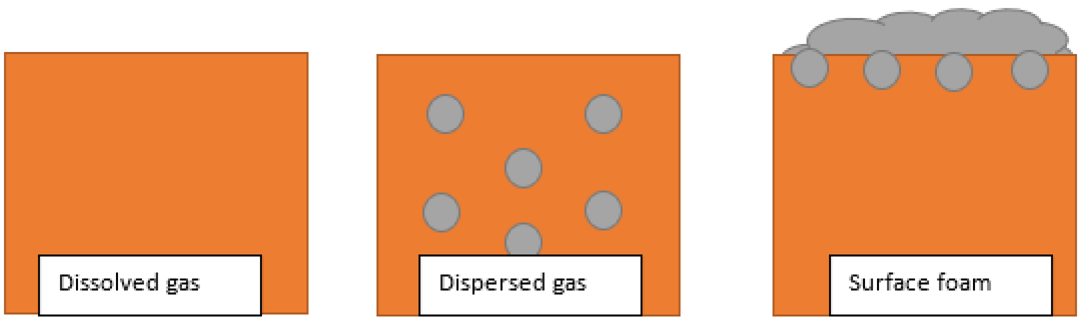


**FIGURE 46 - DIFFERENT STAGES OF GAS IN OIL**

It is explained that the dissolving power is influenced by various physical parameters, however the main contributions come from oil temperature/viscosity and pressure(41). Increasing the oil temperature and pressure allows more gas to be dissolved. Koch et al describe there is a strong correlation between dissolved air in oils and the pressure level. There is a “saturation limit” where the pressure dictates how much air is dissolved in the oil as shown in Figure 47 (40).

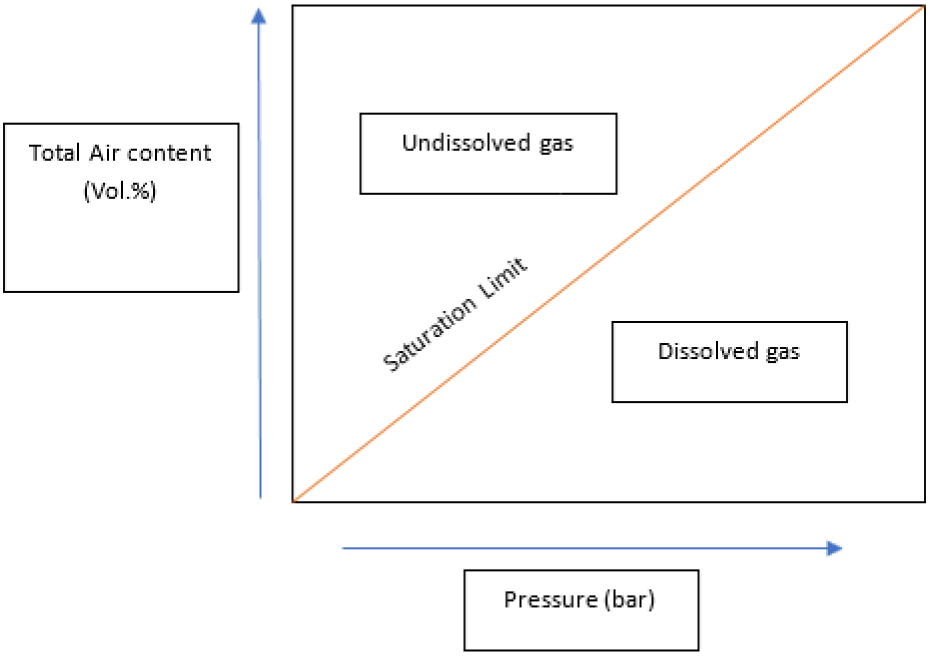


**FIGURE 47 - AIR DISSOLVING POTENTIAL OF OIL**

Theisen et al continues to explain dissolved form of gas has no considerable influence on the physical parameters of an oil. However, undissolved gas fractions (dispersed gas and surface foam) have a direct influence on the physical properties of an oil. Undissolved gas fractions have a significant change in the density, thermal conductivity and a strong influence of the compressibility of a fluid. Furthermore, dynamic effects and flow restrictions in an oil circuit can cause a drop below saturation pressure limit, increasing the amount of undissolved gas in the oil system (41).

### 5.5.4 Oil Dilution

Oil dilution from fuel generally occurs when unburnt fuel impinges on the cylinder walls in a combustion engine, which then flows down past the piston rings and piston skirt, towards the crankcase where it mixes with the oil. This causes the oil to lose viscosity which reduces the lubricating and hydraulic properties of the oil (42).

Although there has recently been a significant interest in investigating the effects of fuel dilution in oil and how it changes the oils performance in the engine due to the demands of carbon emissions (42)(43)(44), there seems to be a severe lack of research on fuel dilution in oil and its effects on frictional performance in internal combustion engines. This is primarily due to the operating conditions of the internal combustion process generating temperatures that largely control the levels of fuel dilution, due to evaporation or being able to absorb the fuel into the oil (43). This lack of literature of fuel dilution in oil and its effects of frictional performance seems to be even more sparse for cold temperatures, therefore strengthening the need for a new test rig to test various oil parameters, including dilution in the UniAir (and other modern valvetrain systems).

Artmann et al explain that fuels containing ethanol has a greater chance of fuel dilution in oil under cold start conditions due ‘to lower calorific value of ethanol a higher amount of fuel is require and the preconditions for mixture formation compared to regular gasoline are worse because ethanol has a higher evaporation enthalpy, a lower vapor pressure and no low boiling components' (42). In addition to this Kleiner et al state that there is a large dependence on oil temperature and the desorption behaviour of the engine oil (43). After reaching the saturation region, either the oil has to change, or a heating process has to be carried out to have the lowest possible fuel content in the oil (43). Without the combustion process and the heat generated in a cylinder head valvetrain test rig, this enables a more controlled testing platform to analyse of the influence of oil dilution across a broad range of temperatures.

## 5.6 Summary & Scope of Research

The literature reviewed supports the intended direction of the work presented in this thesis. It has highlighted the advantages of a motored test rig over fired engine test rigs, which is especially relevant here due to the focus on the valvetrain. It also emphasises the critical importance of how the oil temperature affects the levels of friction within an engine. This strengthens the need to use a motored test rig, because oil temperature can be controlled in a closed system with no heat generated from other events in a fired engine, such as combustion. A more in depth analysis of how the oil temperature influences the operation and efficiency of the UniAir is therefore possible.

In addition to monitoring the levels of friction, because the UniAir is hydraulically driven, valve lift will also be analysed to understand the operation and efficiency of the UniAir. The literature suggests that a likely method to capture valve lift, is to use LDV. This captures the true valve lift operation due to its non-contact application, therefore not influencing any sprung mass of the valve, which allows more accurate readings of valve lift.

The oil is the link between camshaft and valve stem, therefore hydraulic efficiency is a key parameter in understanding the UniAir's performance. The literature has highlighted how the effect of viscosity will drastically change the hydraulic efficiency, which gives an understanding of how the UniAir may perform with changing viscosity. However, the literature has also revealed a lack of consensus concerning hydraulic efficiency at more extreme temperatures. This research will consider at temperatures between 0°C to 120°C whereas the literature typically considers (what would be classed as operating temperatures) temperatures between 40°C to 100°C. This gap in the research emphasises the need to investigate colder and hotter temperatures as oil viscosity is such an important parameter in hydraulic efficiency. Likewise, with varying oil formulations the literature has given an understanding of how the UniAir will perform with these parameters. However once again, the literature is limited in discussing the influence of more extreme temperatures, which creates a critical unknown in the operation and efficiency of the UniAir at these temperatures.

# 6. Design methodology and rig development

In alignment with the objective of this thesis to design, set-up and instrument a valvetrain test rig the literature has given confidence in the direction for the design and development of the rig. This chapter will include the detailed design methodology and rig development for the thesis. Before designing the rig, the supplied materials from JLR are considered. These included:

- Oils
- Supplied hardware
- Design parameters

## 6.1 Oils

In order to successfully design the valvetrain test rig, the selection of oils needed to be considered. The test rig was designed around using the standard SAE 0W-20 oil intended to be used in the production level Jaguar Land Rover AJ200 engine (Appendix 1).

### 6.1.1 Lubricant viscosity

The effect of lubricant viscosity on valve lift deviation and CVVL power losses were studied over the whole temperature range defined in Table 7. Two viscosity grades were considered in the first instance 0W20 and 5W30. Particular attention was given to the potential viscosity compensation needed by the control of the CVVL unit to provide the desired lift for all viscosities considered, and the effect of this compensation on overall system efficiency. From these results, thought can be given to the design of CVVL and on possible optimizations to minimize valve lift deviation and power losses for varying oil viscosities.

### 6.1.2 Oil formulations

A comparison of CVVL operation and efficiency using varying oil formulations within a similar viscosity grade was studied. Castrol Hyspin AWH-M 46, which is typically used in hydraulic applications was chosen as a comparison of the current JLR 0W20 automotive grade as the viscosity profile is very similar. Using a specific hydraulic oil and comparing to the 0W20 will show the potential benefits of using varying base stocks and/or additives (friction modifiers, viscosity index improvers, anti-oxidants, and emulsifiers). To further illustrate the difference of oil viscosities at specific temperatures, Figure 48 shows this relationship.

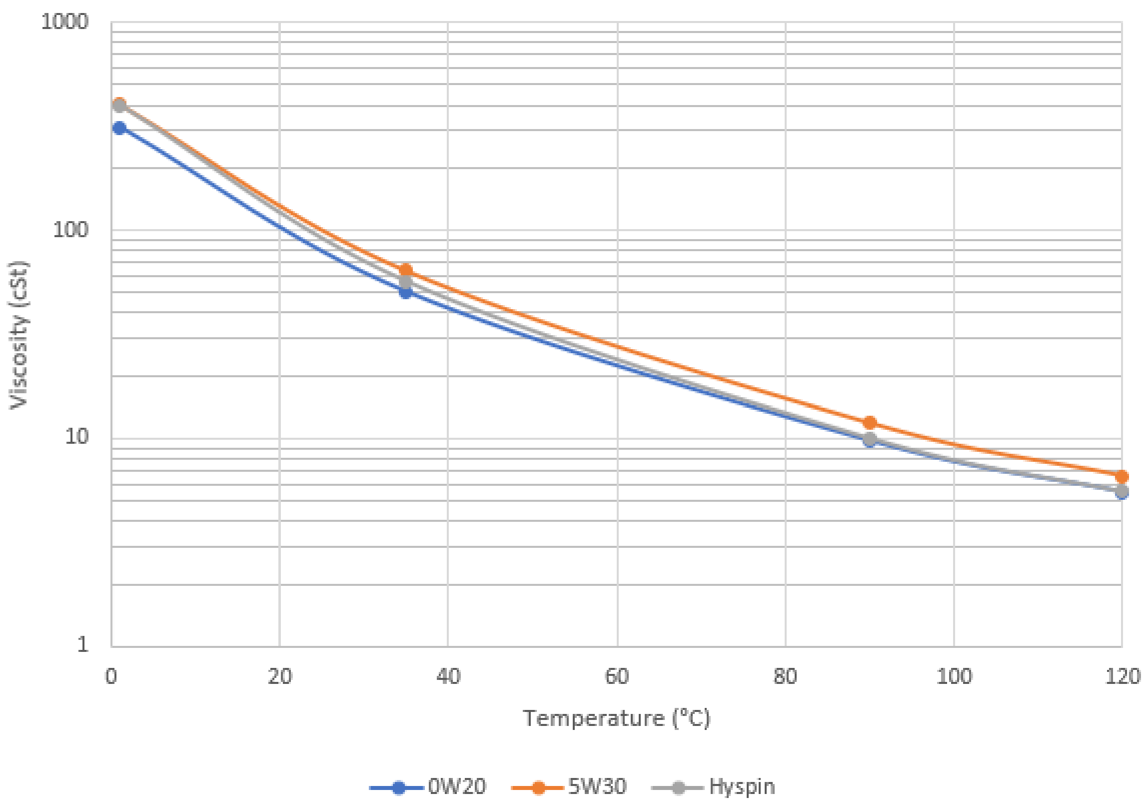


**FIGURE 48 - THE RELATIONSHIP BETWEEN VISCOSITY AND TEMPERATURE FOR 0W20, 5W30 AND HYSPIN OILS**

## 6.2 Supplied hardware

The rig is designed around the supplied hardware from JLR and Schaeffler, which includes:

- JLR cylinder head – this includes all relevant fasteners, camshafts, valve gear and gaskets
- Customized UniAir – JLR will supply a UniAir.
- ECU and calibration – In order to use the UniAir, this must be controlled via the ECU. Engineering support was received from JLR to setup their hardware.

## 6.3 Preliminary boundary conditions

With engineers at JLR the preliminary boundary conditions of the project were selected in Table 7 as these values had potentially the most meaningful output.

| | |
|---|---|
| Camshaft speed (rpm) | 500, 1000, 1500 and 3000 rpm |
| Oil temperature in the UniAir | 0, 35, 90 and 120°C ± 1°C |
| Oil pressure supplied to the UniAir | 3 bar ± 0.1bar |

| | |
|---|---|
| Oil formulations | Castrol SAE 0W-20<br><br>Castrol SAE 5W-30 C1<br><br>Castrol Hyspin AWH-M |

**TABLE 7 - PRELIMINARY BOUNDARY CONDITIONS OF INPUTS**

## 6.4 General Considerations for Test Rig Design

Before designing the specifics of this rig, more general features needed to be further analysed.

### 6.4.1 Foundation

The first main feature to be taken into consideration is the structural foundation of the rig. Baseplates are the most used design for industrial rotating machinery. They are typically either cast or fabricated by using structural steel such as I-beams, channel iron, angle and structural tubing (45). These design styles give high second moment of areas which provides the baseplate a high level of stiffness and mass. However, as discussed in section 5.4, access is required to the valve heads. A frame is therefore needed to lift the cylinder head off the baseplate to enable the required access to the valve heads. The frame is then attached to the baseplate and consequently are not as rigid as equipment mounted to baseplates and frequently will exhibit higher level of vibration amplitude. Due to the welded or bolted connections the surfaces that the machinery attaches to are often not coplanar or parallel to each other (45).

When the equipment is placed onto the frame, the base or 'feet' of the equipment can have poor or no surface contact made between the underside of the equipment to the frame (45). This can be attributed to warped or bowed frames or improper machining of the equipment and is referred to as 'soft foot' (45). Soft foot leads to unsecure equipment which causes residual vibration due to improper shaft alignment positions.

### 6.4.2 Alignment

The primary objective of accurate alignment is to increase the operating life span of rotating machinery. Rotating equipment is specifically designed to be operated within an alignment limit. Operating these components in dynamic situations outside their limits will cause excessive axial and radial forces on the bearings, seals, coupling and shafts. This will reduce the life expectancy of the components as well as generating vibrational forces across the rig (45). Particularly in test equipment, reducing or eliminating vibrational forces is critically important, as they can cause unwanted electrical noise, which can interfere with a sensors (for example a torque transducer) output giving an incorrect value.

Ideal shaft alignment occurs when the centrelines of two (or more) rotating shafts are concentric. Shaft misalignment can occur in two ways: parallel and angular as seen in Figure 49 (45).

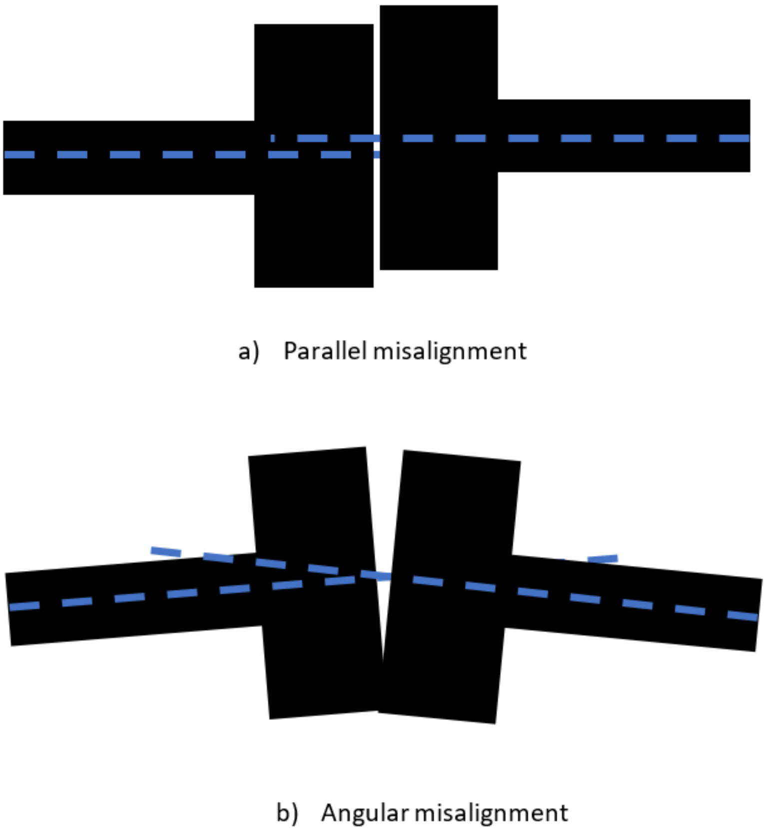


FIGURE 49 - PARALLEL (A) AND ANGULAR (B) MISALIGNMENT BETWEEN SHAFTS

### 6.4.3 RUNOUT

Runout describes a condition when a rotating object is not concentric with its own centreline of rotation as illustrated by Figure 50. This is caused by improper manufacturing and will inevitably cause shaft misalignment.

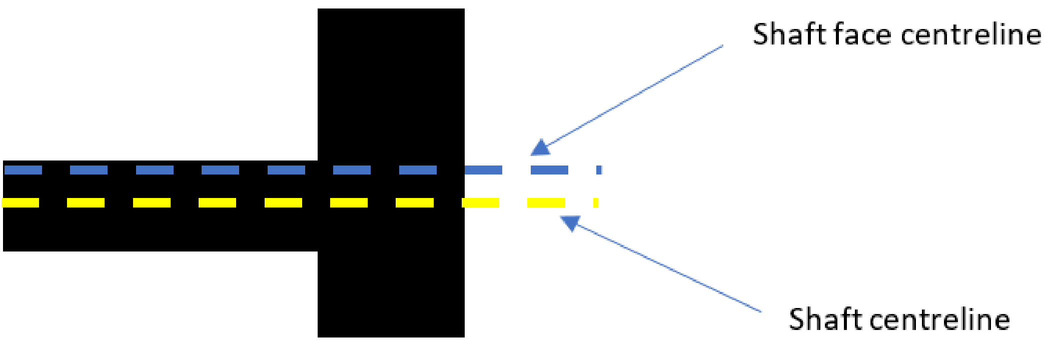


FIGURE 50 - RUNOUT ON SHAFT

### 6.4.4 MISALIGNMENT TOLERANCES

To understand the magnitude of a good alignment a general guide for misalignment tolerances is specified in Figure 51. This gives an indication for which the rig must achieve to negate any alignment issues.

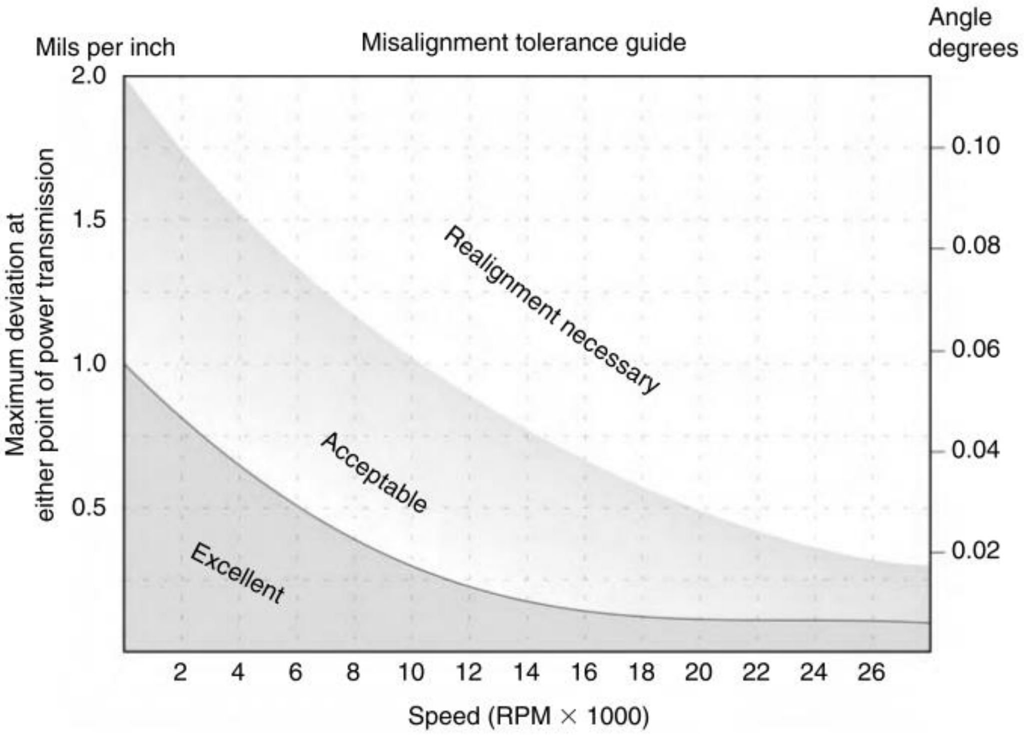


**FIGURE 51 - RECOMMENDED MAXIMUM MISALIGNMENT OF FLEXIBLY CONNECTED ROTATING MACHINERY *(45)***

### 6.4.5 COUPLINGS

Rotating equipment have shaft alignment and runout are manufactured to their designed tolerances; it is nearly impossible to maintain perfect colinear centrelines of rotation between two shafts. Flexible couplings are designed to provide a certain degree of yielding to allow for initial or running shaft misalignment (45). However, these designed 'allowable misalignments' in couplings are specific to the coupling and not the equipment it is coupled to (45). Coupling misalignment tolerances quoted by the manufacturers typically specific the mechanical or fatigue limits of the coupling or components of the coupling (45). These misalignment tolerances are frequently excessive compared to the misalignment tolerances specified for the shafts (45).

Table 8 highlights what a perfect coupling would include with the following design features:

- Allow limited amounts of parallel and angular misalignment
- Be easy to install and disassemble
- Accept torsional shock and dampen torsional vibration
- Minimise lateral loads on bearings from misalignment
- Stay rigidly attached to the shaft without damaging of fretting the shaft
- Withstand temperatures from exposure to environment
- Produce minimum unbalance forces
- Have a minimal effect on changing system speeds
- Be a material capable of long life in the environment in which installed

**TABLE 8 - LIST OF PERFECT COUPLING DESIGN FEATURES *(45)***

### 6.4.5.1 Replicating the firing action of the pistons with a universal joint coupling

An alternative coupling that can be used to mimic the nature of a fired motor are universal joints. They can be used as the input to the camshaft to replicate the firing action of the pistons (46). To reproduce the torsional oscillation of the crankshaft, the universal joints are misaligned. The misalignment of the universal joint then creates an oscillation when rotated (46). The greater the angle of the universal joint the greater the induced oscillation.

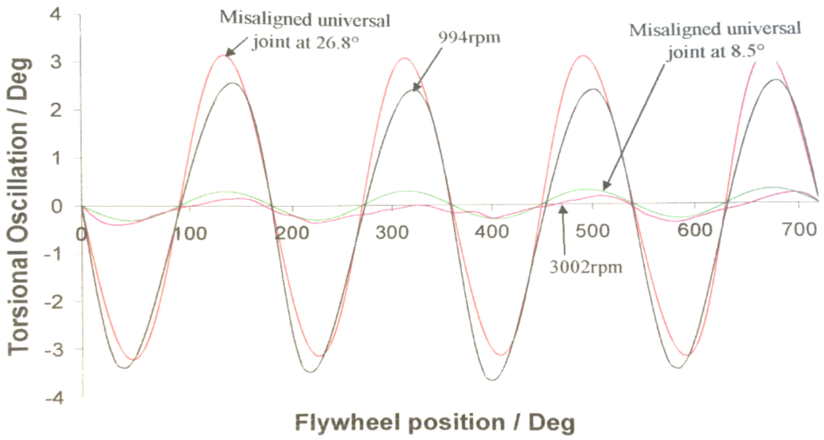


**FIGURE 52 - THE FIRING ACTION OF THE PISTONS VS UNIVERSAL JOINT MISALIGNMENT *(46)***

The angle between the shafts giving the desired torsional oscillation:

$$Tan(\beta) = Cos(\alpha) * Tan(A) \tag{15}$$

Where:

$$\alpha = angle\ between\ shafts$$

$$\beta = angle\ of\ output\ rotation$$

$$A = angle\ of\ input\ rotation\ from\ neutral\ position$$

Using a universal joint can be used to replicate the torsional oscillation of the crankshaft providing a more realistic input to the UniAir. This can potentially identify other variables effecting the UniAir operation and efficiency.

### 6.4.6 Measuring techniques

To measure the parallel and angular misalignment of shafts two techniques are used.

#### 6.4.6.1 Parallel misalignment measuring technique

With reference to Figure 53, a dial gauge is attached to shaft 1, while the dial gauge plunger measures shaft 2. With the dial gauge on the top position, the dial can be set to zero.

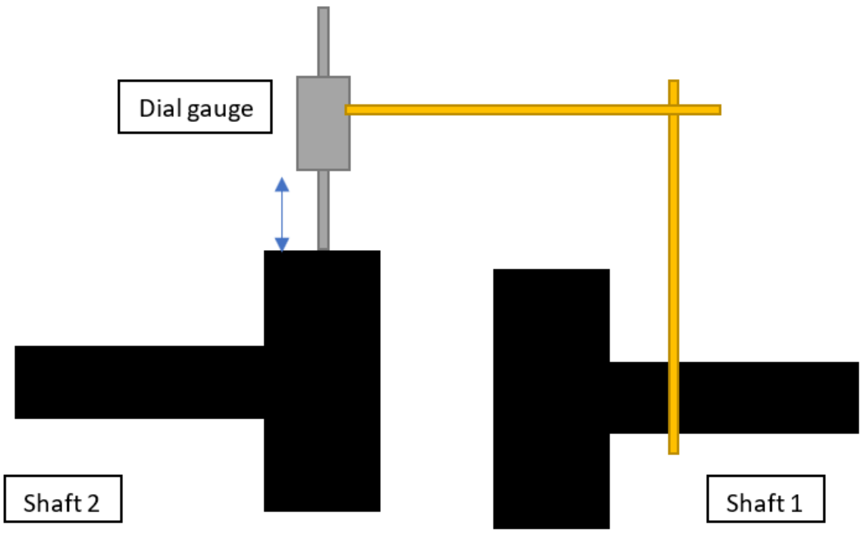


**FIGURE 53 - MEASURING TECHNIQUE FOR MEASURING SHAFT PARALLEL MISALIGNMENT IN STARTING POSITION**

By rotating shaft 1 by 180° as illustrated in Figure 54, the dial gauge will then show how much parallel misalignment there is in vertical plane. Either shaft 1 or 2 can then be adjusted with shims to put it in the correct position. This measuring technique can be also used for measuring the horizontal plane misalignment. The starting position is now 90° from the top position in Figure 53. The dial set to zero and then rotating shaft 1 to 270° to a finished position will give the horizontal misalignment measurement.

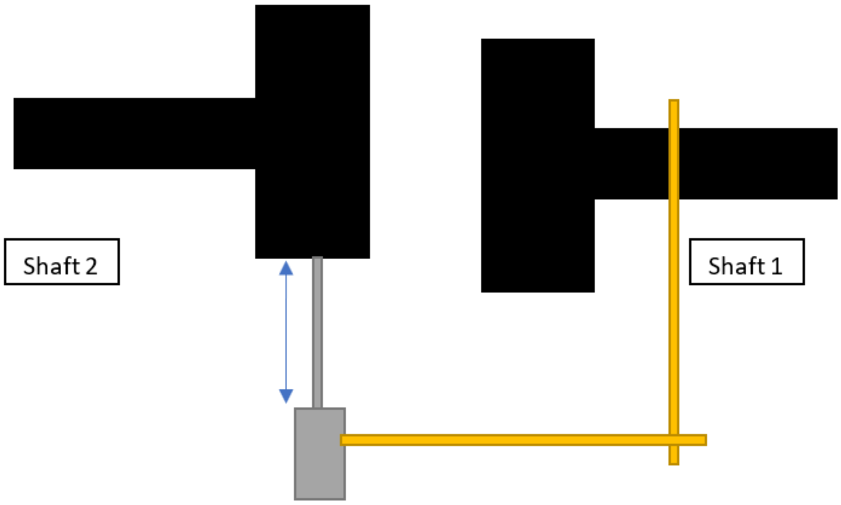


**FIGURE 54 - MEASURING TECHNIQUE FOR MEASURING PARALLEL MISALIGNMENT IN FINISH POSITION**

#### 6.4.6.2 ANGULAR MISALIGNMENT MEASUREMENT

For angular misalignment, the measuring technique is similar, but instead of measuring the displacement on the shaft circumference it measures the displacement on the shaft face. Figure 55, shows the dial gauge is attached to shaft 1, and the dial gauge is placed on the face of shaft 2. In the current position at 0°, the dial gauge is set to zero.

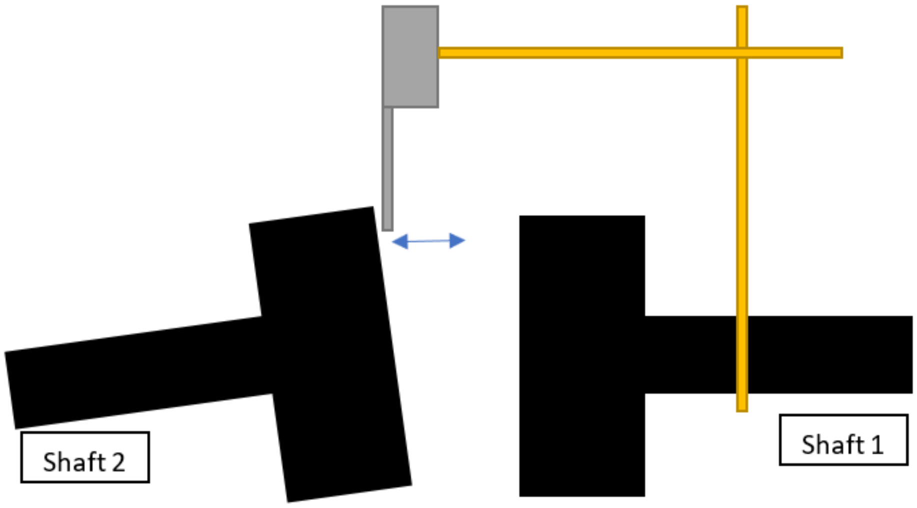


FIGURE 55 - MEASURING TECHNIQUE FOR ANGULAR MISALIGNMENT IN STARTING POSITION

When shaft 1 is rotated 180° the dial gauge will measure the linear displacement of the face allowing to calculate the vertical angular misalignment.

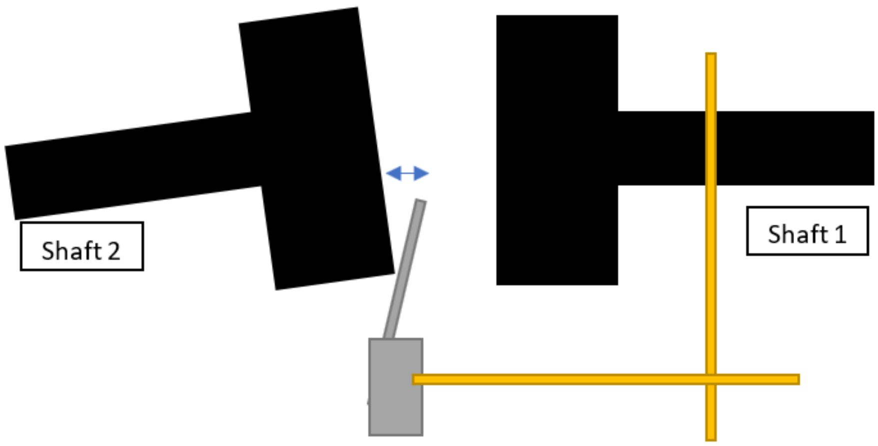


FIGURE 56 - MEASURING TECHNIQUE FOR ANGULAR MISALIGNMENT IN FINISH POSITION

This technique can be used to calculate the horizontal angular misalignment by having the start position at 90° from the top position in Figure 55. Then rotating shaft 1 to 270° to a finished position.

## 6.5 INITIAL TEST RIG DESIGN

The rig attempts to satisfy all the aims, objects, and design parameters of the project, while taking into consideration the previous valvetrain testing and surrounding literature. An electric motor is connected to the camshaft to generate motion of the valves. Between the camshaft and electric motor are various couplings dependent on the setup and a torque transducer. The couplings allow the rig to satisfy the misalignment tolerances while using the torque transducer to capture the required torque to operate the UniAir. Furthermore, the cylinder head and drivetrain of the rig is lifted in the air to gain access to the valve heads to measure the displacement. Additionally, the room created underneath the cylinder head and drivetrain allow additional components to be stored, such as the oil circuit and associated parts.

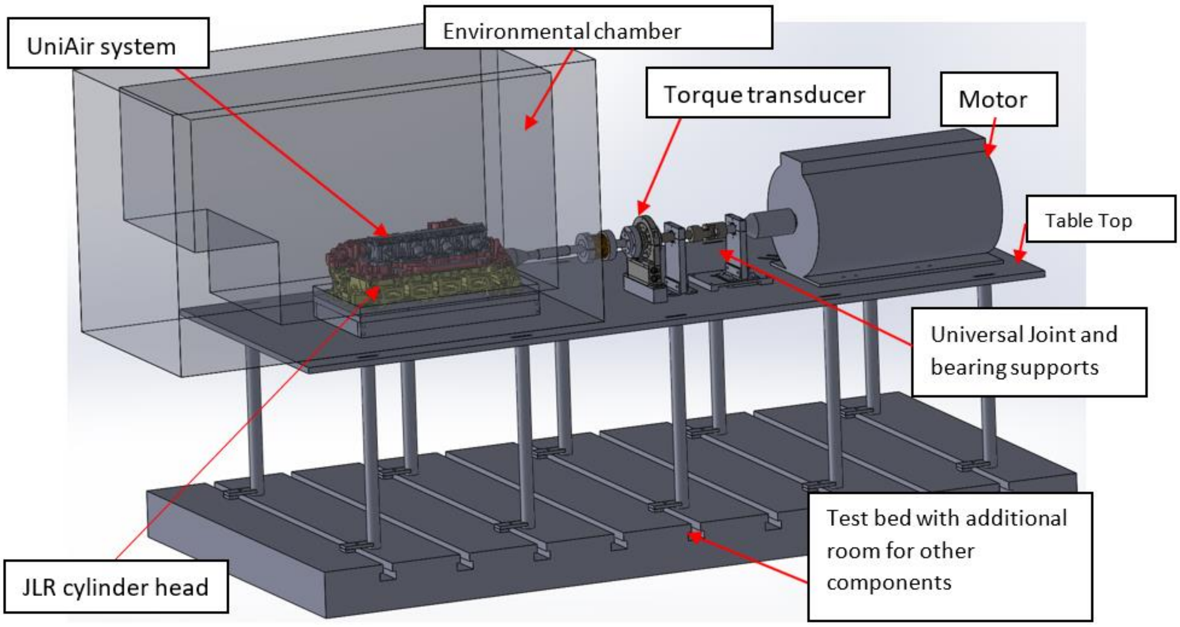


**FIGURE 57 - A COMPUTER AIDED DESIGN VIEW OF THE INITIAL TEST RIG DESIGN**

### 6.5.1 VALVE LIFT

The operation of the UniAir is dependent on oil. Monitoring the valve lift profiles under all the different simulated conditions of the engine, will give a vital understanding of how the oil viscosity effects the operation and efficiency of the UniAir system.

The cylinder head was supplied as part of the equipment by JLR, therefore the rig is a motored cylinder head test rig. The valve lift will be monitored by measuring the linear displacement of the valves during the cycle of the engine. Due to the speed of the valves, mechanical devices will create inaccurate readings due to the inertial forces of the sensor (19). Therefore, a non-contact sensor is used to ensure measurements are uninfluenced by the valve.

With continual commutation with JLR they had specified the shortest timed valve opening and closing speed is at 1500 revolution per minute (rpm) over a 50° Crank Angle for 2mm valve lift.

This means:

$$\frac{1500rpm}{60seconds} = 25\,revolution\;per\;second \tag{16}$$

And that:

$$\frac{50°}{360°} = 13.8\% \tag{17}$$

Therefore:

$$\frac{1\ revolution}{25\ revolution\ per\ second} = 0.04\ seconds\ for\ 1\ revolution \tag{18}$$

Which means the valve opens and closes in:

$$Open\ and\ closes = 0.04 seconds\ \times 13.8\% = 5.52ms \tag{19}$$

To improve accuracy of the valve lift measurement, the sensor must measure the valve head displacement as many times as possible in 5.52ms. The valve lift was measured using an ultra-high-speed laser displacement sensor, LK-H057 from Keyence as they can measure up to $2.55\mu s$. The sensors are situated underneath the cylinder head where it has access to the valve heads. The measurement principle of the LK range is based on triangulation. Given a known relative position, a laser is emitted and then reflected. The valve lift is then calculated by measuring the time of flight of the reflected beam (47). Therefore, to measure the valve movement in the engine, the valves must be visible to the sensors head. They are used to understand influence of oil viscosity on the UniAir operation (valve lift deviation and leak rates) and to calculate hydraulic/volumetric efficiency. The budget only allowed 5 lasers to be purchased, therefore, two lasers were used on cylinder 1 to compare valve to valve variability, two lasers were used on cylinder 3 to compare valve to valve variability and cylinder to cylinder variability. The fifth laser was used to monitor the table top frequency and deduct it from the measured valve lift to get a true representation of valve lift.

To measure the valve movement in the engine, the valves must be visible to the sensors head. Figure 58 illustrates a section of the Table Top is cut away in order to have access to the valve heads for measurement. Figure 59 highlights the sensors are located externally and independently to the table and any temperature control components. Although this position is ideal for both measurement and sensor environmental requirements, this causes additional problems with vibration.

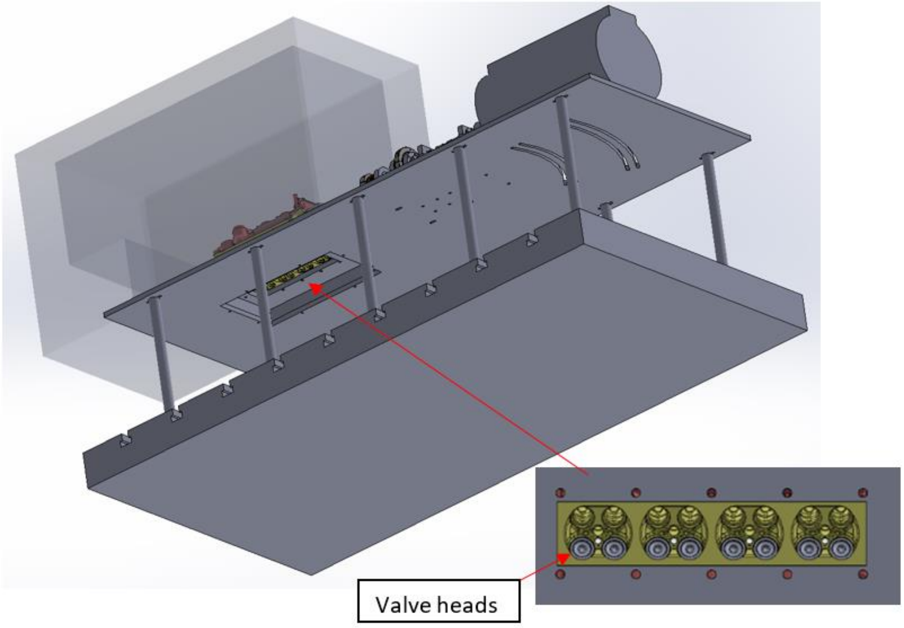


**FIGURE 58 - AN UNDERNEATH VIEW OF THE VALVETRAIN TEST RIG HIGHLIGHTING THE ACCESS TO THE VALVE HEADS**

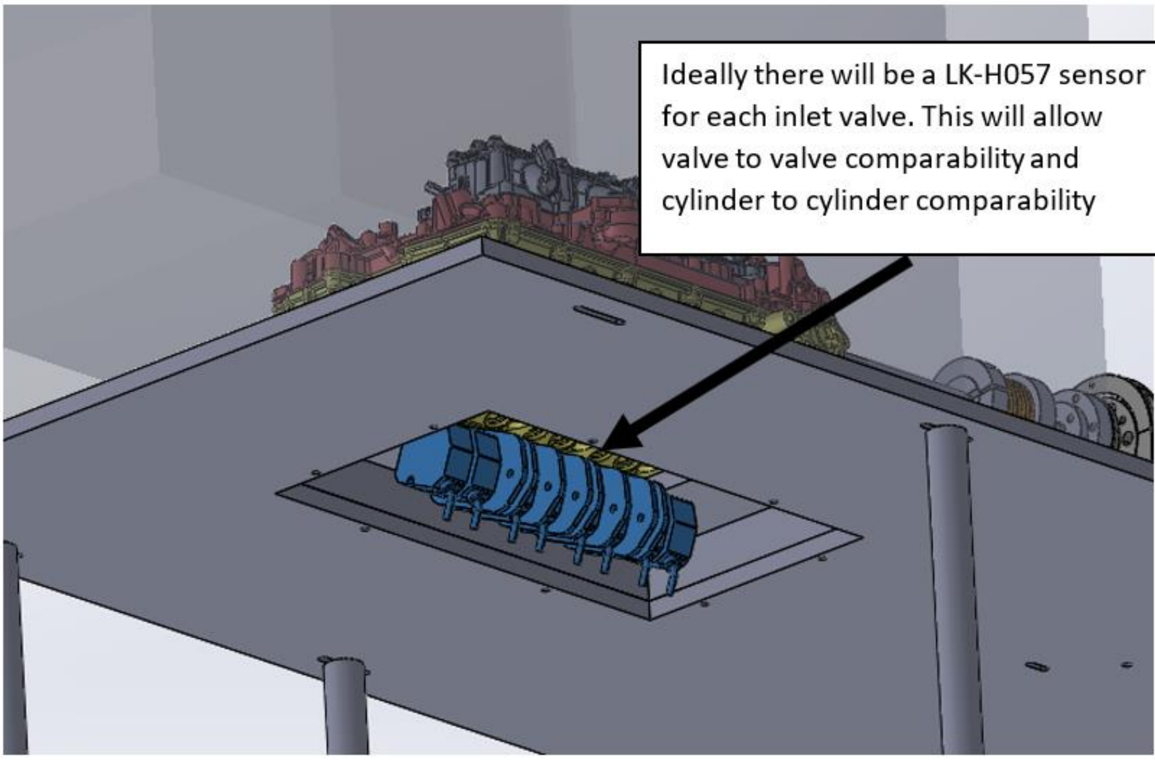


**FIGURE 59 UNDERSIDE VIEW OF SENSORS MONITORING VALVE MOVEMENT**

The dynamic behaviour and vibration from other components such as the motor can potentially cause exaggerated valve lift measurements (19). Figure 60 illustrates how the laser is used to measure the valve displacement, whilst also highlighting the laser is independently mounted. Ideally an additional LK-H057 sensor will be used to monitor the Table Tops frequency, as the moving parts of the rig will cause vibrations. This frequency will be monitored in-situ with the valve measurements, which can then be deducted from the valve lift measurements to get a raw sample of the valve displacement. This will help create reliable comparable data between different tests on the rig.

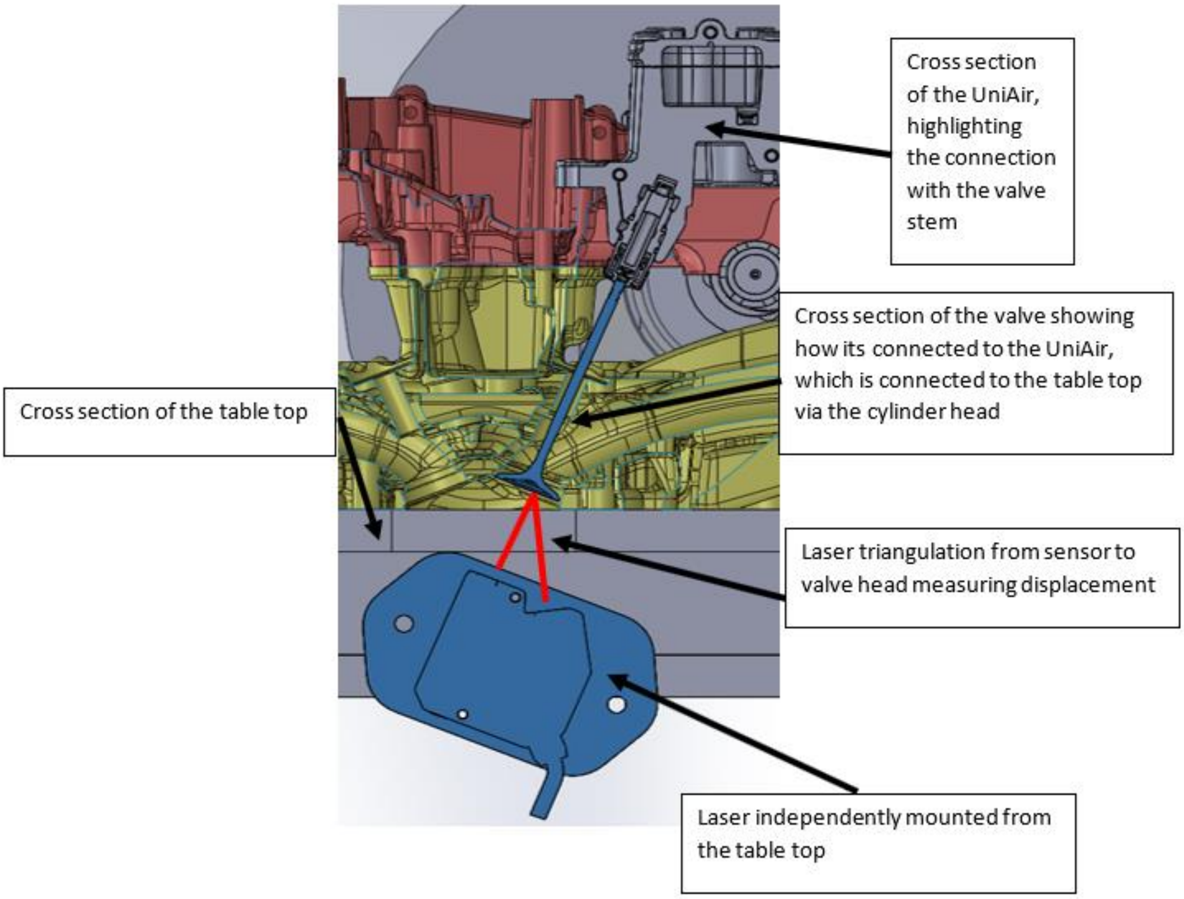


**FIGURE 60 - HIGHLIGHTING HOW THE LASER MEASURES THE VALVE DISPLACEMENT AND INDEPENDENT MOUNTING**

### 6.5.2 TEMPERATURE CONTROL

The literature has shown that the temperature of the oil affects the operation and efficiency of both valvetrain and pump friction torque (15). Therefore, the temperature control is crucial to the performance of the rigs test results. It is expected that the supplied oil temperature will gain/lose temperature due to the ambient air conditions around the UniAir. Therefore, to control the oil temperature inside the UniAir an environmental chamber will also be used. This section looks at the methods of controlling the temperature for the oil and the environment.

#### 6.5.2.1 ENVIRONMENTAL TEMPERATURE

As well as controlling the oil temperature, the entire UniAir needs to replicate the operational temperatures. An environmental chamber will be located around the engine to simulate the under-bonnet air conditions. JLR specified that the under-bonnet air temperature range is 0°C to 120°C.

#### 6.5.2.1.1 COLD SETUP

To simulate the cold environment an insulated box with a heat extraction system is used. To do this a laboratory specified deep chest freezer LGT 3725 Chest Freezer is modified to surround the cylinder head and reduce the environmental temperature(48). This particular freezer has the volume to fit the required components inside, while also being able to achieve the minimum temperatures of the simulated environment.

Discussion with the freezer manufacturer Liebherr (a large freezer manufacturer), it revealed that there are no refrigerant pipes on the bottom of the freezer or on the side where the compressor is. This then allowed a section to be cut out of the freezer floor, so the cylinder head can sit inside the freezer when on the Table Top.

Figure 61 shows how the cylinder head was placed inside of the freezer while trying to maximise the original insulation. This is done by creating a step on the table, which raises the engine level with the insulation on the floor. The step up also aligns the camshaft to the motor axis.

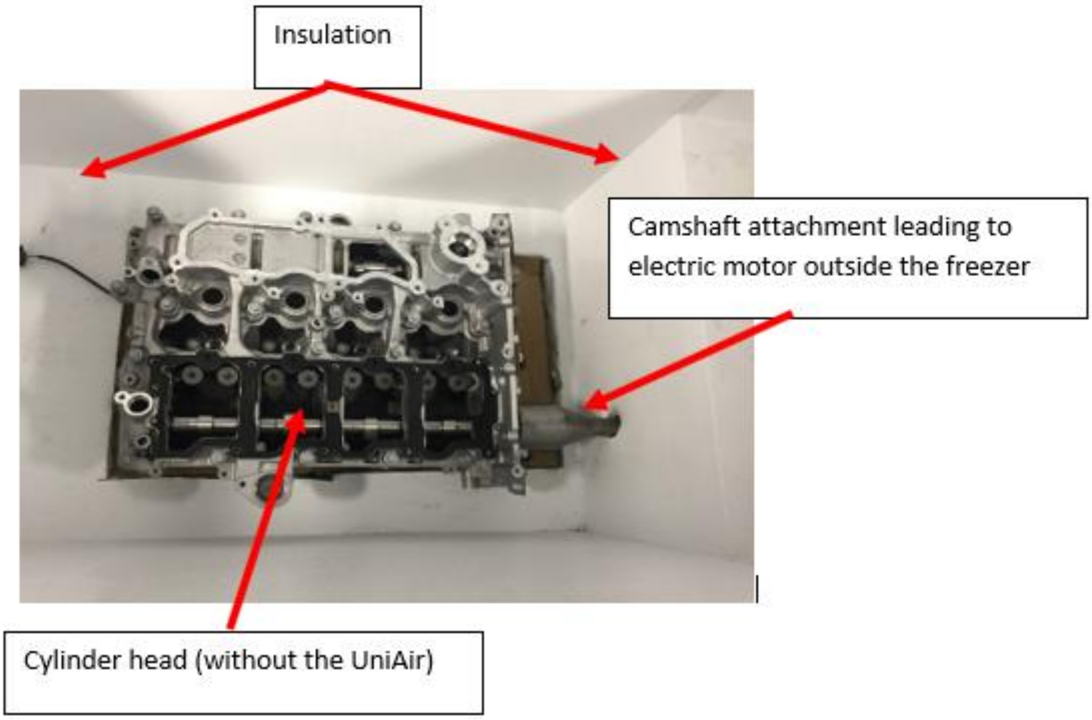


FIGURE 61 - INTERNAL VIEW OF COLD CHAMBER HIGHLIGHTING THE INSULATION AROUND THE CYLINDER HEAD

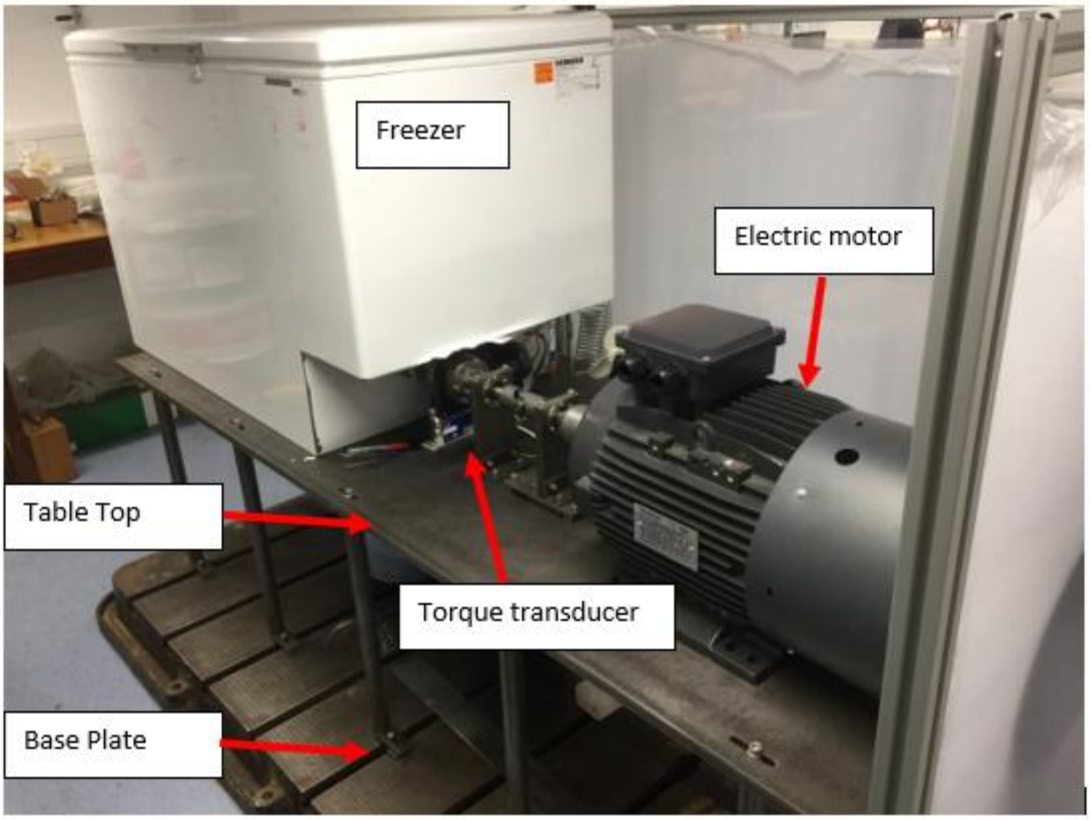


**FIGURE 62 - VIEW OF RIG HIGHLIGHTING COLD CHAMBER**

The test rig objective is to evaluate oil and environmental temperatures varying from 0°C to 120°C. The heat transfer of the oil and environment needs to be calculated to see if the freezer is powerful enough to maintain its temperature.

To create a condition that represents the worst case, it will be assumed that the air in the freezer is at -40°C and the entire UniAir unit will be instantaneously put inside the freezer at 120°C. This particular problem is heat transfer by convection (49). This can be calculated by the following equation:

$$\dot{Q} = hA(T_s - T_\infty) \quad (20)$$

Where:

$\dot{Q} - Rate\ of\ convective\ heat\ transfer\ (J.s^{-1}\ or\ W)$

$h - Convective\ heat\ transfer\ coefficient\ (W.m^{-2}.K^{-1})$

$A - Area\ perpendicular\ to\ the\ direction\ of\ heat\ flow\ (m^2)$

$(T_s - T_\infty) - is\ the\ temperature\ difference\ between\ the\ surface\ and\ free\ stream\ (K)$

The convective heat transfer coefficient (h) is a value used to combine all the difficult to determine parameters about the fluid flow (49). As a result, these values are usually determined through empirical or computational methods (49). To gain an understanding of the performance of the freezer an approximation of 10 $W.m^{-2}.K^{-1}$ will be used (49).

Using the approximate dimensional values of the UniAir unit (as seen in Appendix 3) and equation 20, allows the following calculation:

Front and back

$$\dot{Q}_{FB} = 2 \times 10\ W.m^{-2}.K^{-1} \times 0.08m \times 0.4m \times 190K = 121.6W \quad (21)$$

Sides

$$\dot{Q}_{S} = 2 \times 10\ W.m^{-2}.K^{-1} \times 0.08m \times 0.14m \times 190K = 42.56WF = \mu N \quad (22)$$

Top

$$\dot{Q}_{T} = 10\ W.m^{-2}.K^{-1} \times 0.14m \times 0.4m \times 190K = 106.4W \quad (23)$$

The total convective heat transfer:

$$\dot{Q} = \dot{Q}_T + \dot{Q}_S + \dot{Q}_{FB} = 121.6W + 42.56W + 106.4W = 270.56W \quad (24)$$

Liebherr state that the LGT 3725 requires 125W to achieve its maximum performance of -45°C (228.15 K) in 43°C (316.15 K) ambient conditions (48). The coefficient of performance (COP) of Carnot refrigerators is determined using the following equation (49):

$$COP = \frac{1}{\frac{T_H}{T_L} - 1} \quad (25)$$

Where:

$$T_H = Hot\ temperature$$

$$T_L = Low\ Temperature$$

Therefore:

$$COP = \frac{1}{\frac{316.15}{228.15} - 1} = 2.6 \tag{26}$$

This means that for every 1 joule the refrigerator uses, the coolant can transfer 2.6 joules out of the refrigerator. With a maximum power of 720W, this means the refrigerator can transfer 1.872kW out of the freezer.

#### 6.5.2.1.2 HOT SETUP

The most cost-efficient method to simulate the hot environment is to use the freezer as environmental chambers for hot and cold. This will be done by having the freezer surrounding the cylinder head in the same way as the cold setup. However, the freezer will be turned off and purely used for its insulation to capture the heat generated from using the rig.

### 6.5.2.2 OIL PATH AND TEMPERATURE MONITOR

Figure 25 highlighted the oil path for the entire engine for the AJ200. However, because this study is only interested in the performance of the UniAir, all the relevant oil galleries were blocked off as well as parts that are not used in the function of the UniAir on the cylinder head.

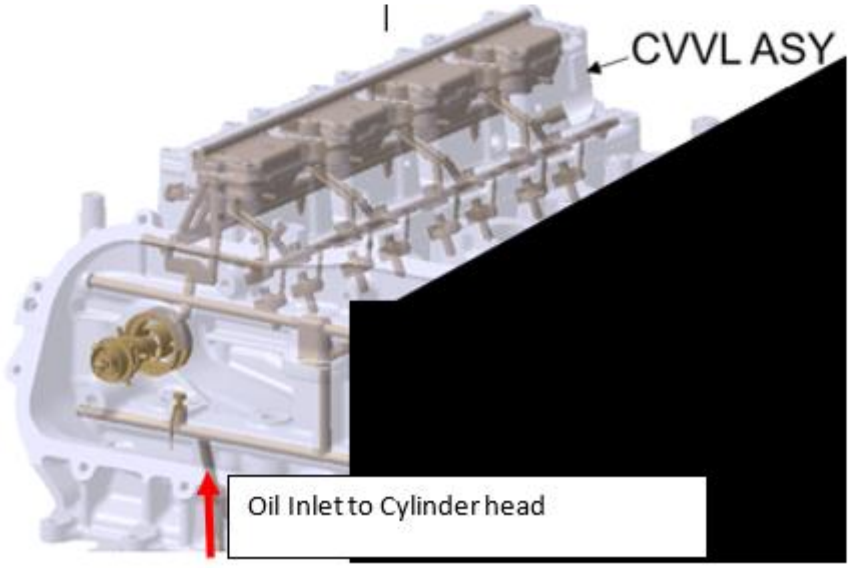


**FIGURE 63 - OIL PATH TO UNIAIR**

The oil galleries that were not used for the UniAir were blocked with metal epoxy that was rated to the temperatures of the rig. The following oil circuit in Figure 64 is created.

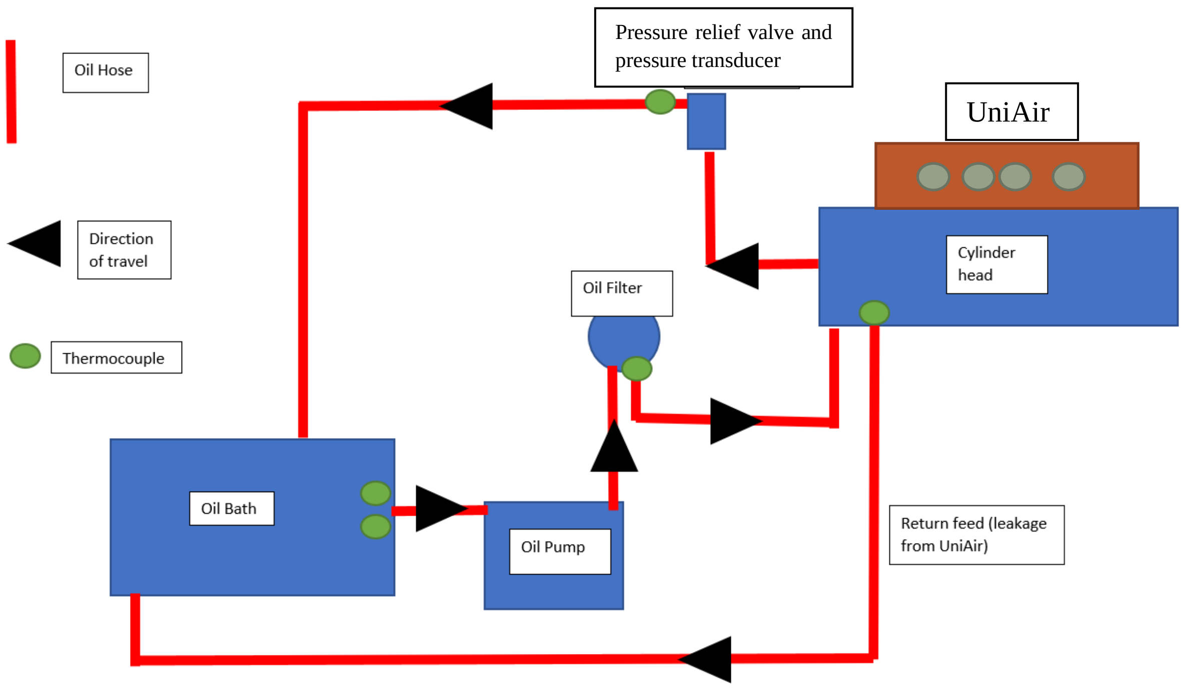


**FIGURE 64 - SCHEMATIC DIAGRAM OF OIL CIRCUIT**

Figure 64, shows the oil path as well as the temperature monitoring positions across the rig. As highlighted in Figure 15, the UniAir has an intermediate pressure chamber for each cylinder in the engine. The AJ200 engine has four cylinders therefore has four intermediate pressure chambers. Consequently four k-type thermocouples are placed on the outside of the UniAir intermediate pressure chambers (as seen in Figure 65) to monitor the temperature distribution across the UniAir.

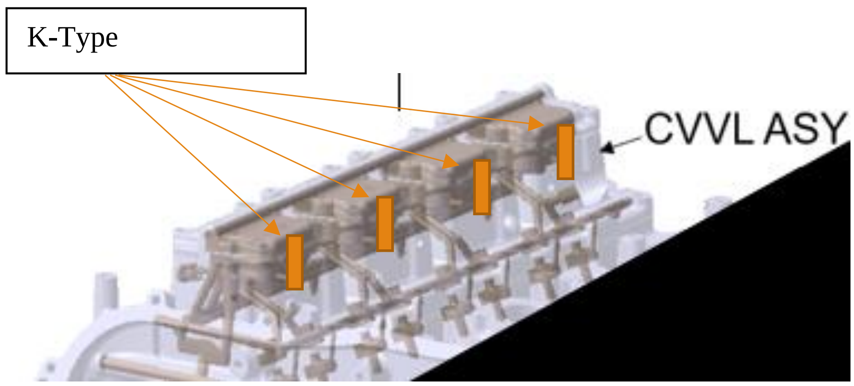


**FIGURE 65 - K-TYPE THERMOCOUPLE POSITIONS ON UNIAIR**

### 6.5.3 Power

The test rig needed to be designed to measure the valve lift profiles and the power supplied to test the operational efficiency (relative power losses) of the CVVL system.

$$Power = Torque \times Angular\ Speed \tag{27}$$

To monitor the power supplied to the UniAir system, the angular speed of the camshaft and the torque required to overcome the systems friction are needed. To understand what magnitude the friction forces of the camshaft are, the valvetrain study by Schaeffler was analysed.

It was found that the lift mode Early Intake Valve Closing (EIVC) at 35°C oil temperature generated the highest amounts of friction torque in the system. Figure 32, indicates that the highest value of friction torque was approximately 4.3 Nm. In addition, Figure 29, demonstrates that the camshaft experienced spikes in torque when testing at approximately every 1000rpm increase. The torque peaks at approximately 30 Nm during the ramp up of the AJ200 test rig.

This test rig is designed to test oil temperatures down to 0°C, which will make the viscosities of the oil much higher (15). Previous experimental tests (section 4.1) suggest that higher viscosities will generate much higher frictional forces throughout the camshaft and UniAir. The torque required to operate the test rig will need to be significantly over specified as the torque at this temperature to operate the UniAir is unknown. With discussions between Schaeffler and JLR, a Figure of 60 Nm was suggested as a worst-case torque requirement.

#### 6.5.3.1 Torque transducer

With reference to Equation 27, to calculate the supplied power the torque and angular speed of the camshaft is needed. To do this, a torque transducer was used to capture the required torque to operate the valvetrain. This is in addition to the evidence found in evaluating the hydraulic efficiency using torque transducers to calculate mechanical power (35).

The T40B torque transducer from HBM monitors the static and dynamic torque, rotational speed, and position of the camshaft. The T40B torque transducer is capable of measuring torque values up to 100N.m, which is above the required value of 60N.m stated in 6.5.3. Furthermore, the T40B is able to provide high accuracy of 0.03% in terms of linearity including hysteresis.

The position encoder which tracks the camshaft rotational position, which is essential to determine the torque and speed for a specific camshaft position. The torque transducer will be positioned as close to the camshaft as possible to capture the true torque characteristics of the CVVL system.

### 6.5.4 Couplings

In order to attempt to replicate the dynamics resulting from the torsional oscillation of the crankshaft in 6.4.5.1 an arc is cut into the table top to allow the motor to rotate so an angle can be applied to the universal joint illustrated in Figure 66.

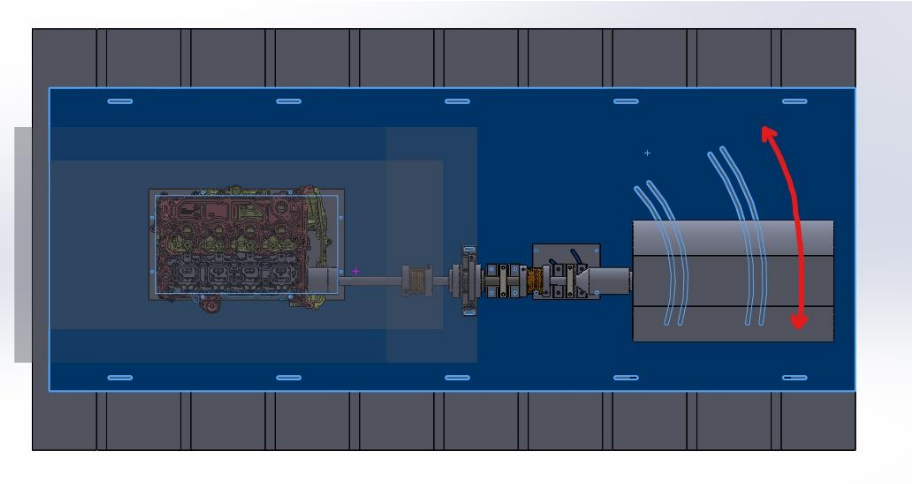

FIGURE 66 - TOP VIEW OF RIG LAYOUT SHOWING UNIVERSAL JOINT MOVEMENT

Alternatively, when universal joint angles are not required, a bellow coupling called smartflex from Mayr will be used in its place to compensate for any misalignment between the camshaft and electric motor. The smartflex is capable of nominal torque values up to 700N.m and will maintain torsional rigidity up to 10000rpm, both within the specified tolerances for this rig found in section 6.5.3 and 6.3 respectively (50).

## 6.6 CONTROL AND INSTRUMENTATION

To gain a better understanding of the rig and its components, this section will cover all the controls and instruments used to measure, monitor and supply the inputs and outputs to the rig. Figure 67 illustrates the final layout of the rig while highlighting all of the components on the rig. Table 9 identifies the control components while Table 10 identifies the measurement and instrumentation components in Figure 67.

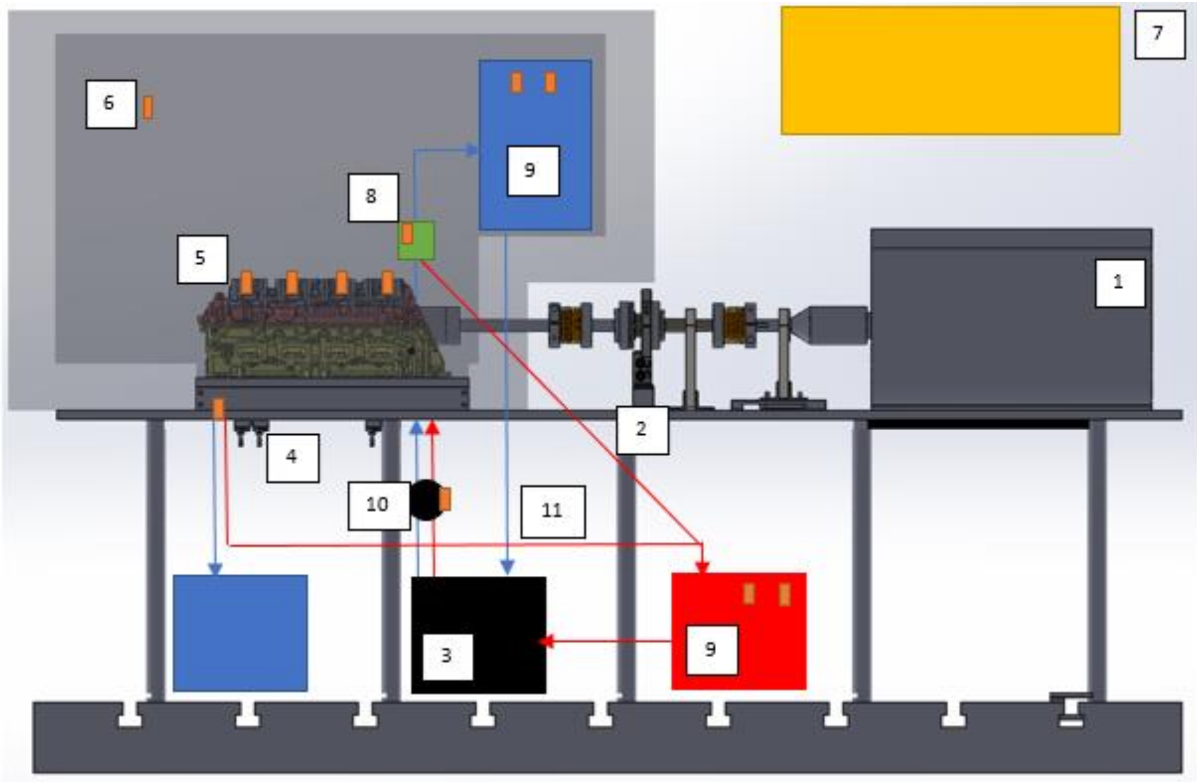


**FIGURE 67 - SCHEMATIC OF RIG AND ITS COMPONENTS**

| | |
|---|---|
| 1 | 22kW 3 phase 2 pole motor. Capable of delivering expected highest levels of torque (>200Nm) and speed range (500-3000rpm) |
| 3 | 7.5kW oil pump |
| 5 | JLR supplied UniAir system for AJ200G engine |
| 7 | National instruments control system NI CDAQ-9133 Controller and Inverters for motors for speed variation |
| 8 | Pressure relief valve and Pressure transducer. Pressure relief valve set to 5.5 Bar just below maximum operating pressure of UniAir to ensure components can't be damaged. Pressure transducer monitoring supplied pressure to the cylinder head. |
| 9 | For the hot setup, a Grant 150R oil heater was used, which is capable of heating upto temperatures of 150°C. However for the cold setup, the most cost efficient way to cool the oil is to put the oil bath inside the freezer with the cylinder head. The difference in oil circuit is symbolised with red (hot setup) and orange (cold setup) lines |
| 10 | Oil filter transferring oil supply to cylinder head/UniAir. |
| 11 | Oil return hoses. Due to the cooling capabilities of the freezer the cold setup does not recirculate the oil (blue lines) and is captured separately to the oil supply. However, the hot setup does recirculate the oil as the heating capabilities of the Grant 150R can recondition the oil to the desired temperature before supplying back to the cylinder head. |

**TABLE 9 - LIST OF CONTROL COMPONENTS IN FIGURE 67**

| | |
|---|---|
| 2 | HBM K-T40B Torque Transducer. Used to measure torque and speed output for frictional operation, and mechanical efficiency. |
| 4 | The valve lift was measured using an ultra-high-speed laser displacement sensor, LK-H057 from Keyence. The sensors are situated underneath the cylinder head where it has access to the valve heads. |
| 6 | k-type thermocouples monitoring air temperature in chamber, oil temperature across the whole system from oil bath to exit from cylinder head. Particular attention was given to the oil temperature in the UniAir with 4 thermocouples situated on the UniAir for each cylinder as seen in Figure 64 |

**TABLE 10 - LIST OF MEASUREMENT/INSTRUMENTATION COMPONENTS IN FIGURE 67**

# 7. TEST METHODOLOGY

## 7.1 INTRODUCTION

In order to create a robust and optimal test procedure, a systematic method to determine the relationships between inputs affecting the process and the outputs of that process will be investigated (51). This systematic method is also known as Design of Experiments (DoE) and can be split into three main objectives:

- Screening
- Optimisation
- Robustness

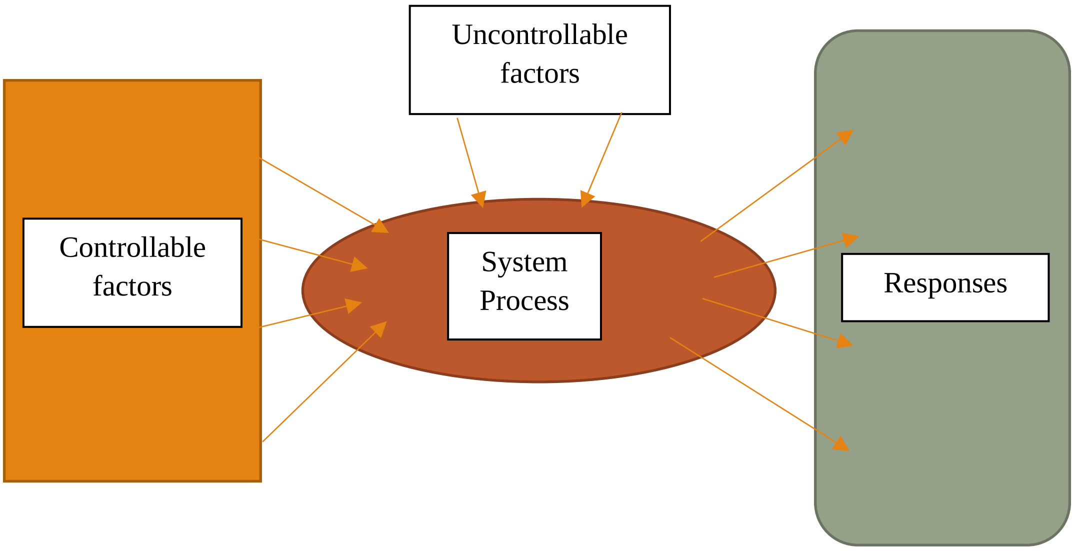


FIGURE 68 - APPROACH TO EXPERIMENTAL TEST PROCEDURE

Section 6.5.2 highlights this rig has two different setups, hot and cold. Since there are two different setups, both setups are considered when creating the test methodology. To create a data set for comparisons so the rig can be validated, the hot setup tests temperature conditions 35°C, 90°C and 120°C to replicate Schaeffler's testing points seen in Section 4.1. Alternatively, the cold setup will test novel temperature conditions at 0°C to further understand the influence of oil viscosity on the UniAir operation and efficiency.

## 7.2 SCREENING

The screening process is used at the beginning of experimental procedures to explore as many factors as possible. Using the rig and setting the current known controllable factors, highlights what uncontrollable factors and the responses of the system. Section 1.3 stipulates the objective of this thesis is to validate the valvetrain test rig by comparing results with previous experimental and simulated tests. Section 4.1 explores the test procedure used by Schaeffler which presented a systematic method for testing on this rig and having a direct comparison to Schaeffler's test for validation. However, Schaeffler's test procedure only measured camshaft speeds up to 2500rpm. The boundary conditions in Table 7 highlight this rig is to observe the response of the UniAir at 3000rpm (6.3). Therefore, before adopting Schaeffler's test procedure, the understanding of the uncontrollable factors and response of the rig at 3000rpm is required.

Figure 27 shows Schaeffler's rig and the last oil temperature measurement point, which is at a distance away from the cylinder head and UniAir. Since this distance is unknown the test procedure cannot accurately replicate Schaeffler's test condition. However similarly to the Schaeffler rig, Figure 64 shows this rig has an oil temperature measurement point at the oil filter which is an equivalent distance.

### 7.2.1 SCREENING TEST 1 – INTERNAL HEAT

The first screen test was conducted using the hot setup while observing the temperatures on the rig found in Figure 64. The temperatures captured on four k-type thermocouples on the UniAir are averaged and called 'UniAir Avg', while the two thermocouples in the oil bath are averaged and called 'Oil Bath Avg' so there is one temperature to represent a specific time.

To help validate the rig the chosen controllable factors are used to replicate Schaeffler's testing where possible, for this screening test the controllable factors in Table 11 are used:

| | |
|---|---|
| Camshaft Speed | 3000rpm |
| Oil Type | Castrol SAE 0W-20 (Matching Schaeffler oil) |
| Oil Bath temperature | 35°C (Matching Schaeffler temperature test point) |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar (Matching Schaeffler pressure map) |
| Test time | 2 Minutes/120 seconds (replicating Schaeffler's test time for each speed) |

**TABLE 11 - CONTROLLABLE FACTORS FOR SCREENING TEST 1**

Using the controllable factors in Table 11, Figure 69 shows the temperature response of the system. Figure 69 illustrates the temperatures captured on the rig at 3000rpm with the oil bath temperature set at 35°C. The oil is transferred through the oil circuit, where there is a heat loss of approximately 10°C to the oil filter over a 120 second test. This is due to the oil hose, pump and oil filter being at ambient temperature, the heat loss occurs from the oil heating these components up before it gets to the oil filter. This is shown by the oil filter (brown line on Figure 69) temperature increasing over time as these components warm up. Therefore, to achieve the required oil temperature at the oil filter, the oil bath temperature needs to be set higher. This also suggests that further heat would be lost from the oil between the oil filter and the UniAir.

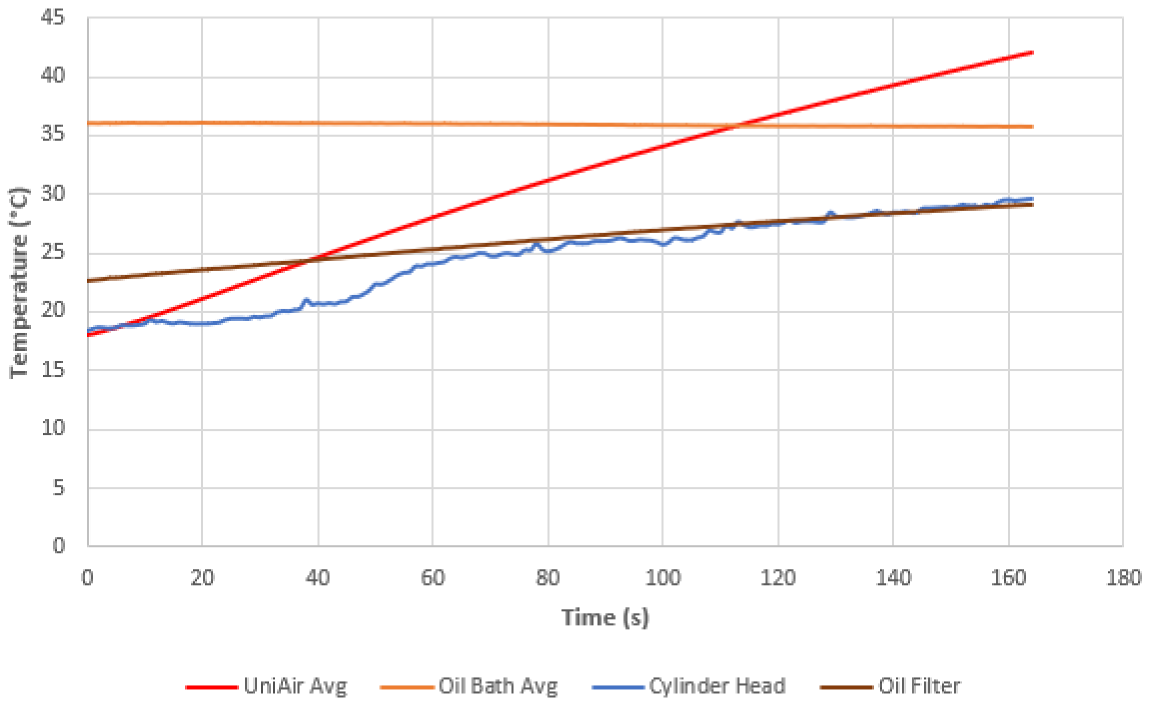


**FIGURE 69 - SCREENING TEST OBSERVING TEMPERATURE RESPONSE OF SYSTEM AT 3000RPM**

Section 1.3 highlights one of the objectives of this thesis is to understand the influence of oil viscosity on the UniAir operation and efficiency. The literature explains that oil viscosity is directly related to temperature, therefore the positions of the thermocouples are critical to understand the influence of the oil viscosity on the UniAir operation and efficiency.

Section 4.1 highlights Schaeffler's test procedure, where the test duration is 2 minutes (120 seconds) at each speed increment. Figure 69 details that the oil temperature in the UniAir (UniAir Avg) actually exceeds the oil temperature in the bath over 120 seconds test at 3000rpm. This provides evidence that the oil temperature in the UniAir is more dependent on the camshaft speed than the oil inlet temperature at higher engine speeds.

The UniAir temperature achieved 35°C after approximately 110 seconds, which is shorter than the 120 second test time Schaeffler implemented for each speed increment seen in 4.1. Therefore, using the UniAir temperature as the setpoint for analysing the UniAir performance would suggest that the test time used by Schaeffler is not appropriate for this test, which gives further justification for creating a new rig and test methodology.

Using the controllable factors in Table 11, this test has identified two uncontrollable factors on the rig, internal heat generated from the UniAir operation and heat loss from the oil across the rig from the oil bath.

### 7.2.2 SCREENING TEST 2 – CAMSHAFT SPEED INFLUENCE

Screening test 1 (7.2.1) at 3000rpm identified an uncontrollable factor of internal heat generated within the UniAir. To further understand the influence of the heat generated, Screening test 1 is replicated however using the lowest testable camshaft speed at 500rpm (Table 7). Therefore, the controllable factors for Screening test 2 are shown in Table 12.

| | |
|---|---|
| Camshaft Speed | 3000rpm |
| Oil Type | Castrol SAE 0W-20 (Matching Schaeffler oil) |
| Oil Bath temperature | 35°C (Matching Schaeffler temperature test point) |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar (Matching Schaeffler pressure map) |
| Test time | 2 Minutes/120 seconds (replicating Schaeffler's test time for each speed) |

**TABLE 12 – CONTROLLABLE FACTORS FOR SCREENING TEST 2**

Using the controllable factors in Table 12, Figure 70 shows the temperature response of the system.

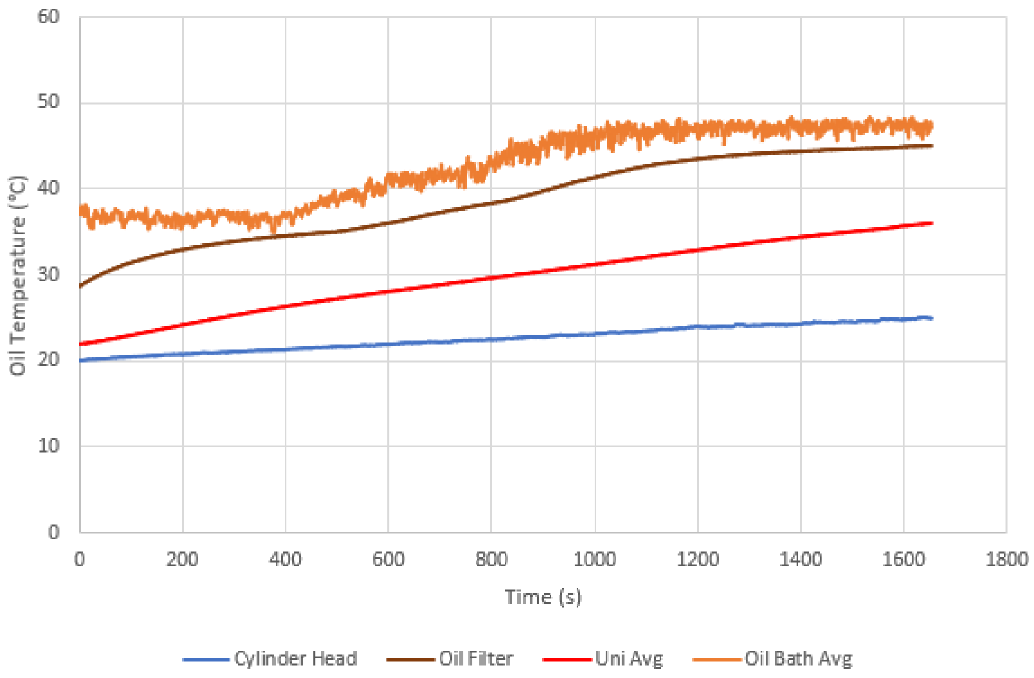


**FIGURE 70 - SCREENING TEST OBSERVING TEMPERATURE RESPONSE OF SYSTEM AT 500RPM**

The initial 120 second test time stated for Screening test 2 was extended due to the findings during the test. It was found from Figure 70 that after 120 seconds of testing, the oil filter temperature (brown line on Figure 70) had not achieved 35°C, and the UniAir Avg temperature (red line on Figure 70) was not increasing as fast as it did at 3000rpm. After 300 seconds, the oil filter temperature had achieved 35°C however the UniAir Avg temperature was still approximately 10°C below the set point. Therefore, the oil bath temperature (orange line on Figure 70) was increased to 45°C and after approximately 900 seconds of testing, the oil bath temperature had achieved its new set point. However, the UniAir Avg temperature had still not achieved the original temperature set point of 35°C, even with the increased oil entry temperature. It was after approximately 1500 seconds of test time that the UniAir Avg temperature had reached the original temperature set point of 35°C.

Figure 64 highlights the cylinder head temperature is the outlet temperature from the UniAir. Figure 69 in Screening Test 1, shows the oil filter temperature was closely related to the cylinder head temperature over the test duration. However, in Figure 70 the oil filter temperature was significantly higher than the cylinder head temperature. This shows that for higher speeds, the UniAir and cylinder head temperature increases at a faster rate in comparison to lower speeds. This suggests that for tests at higher speeds, the UniAir Avg temperature is less dependent on the oil bath temperature. For tests at lower speeds, the UniAir Avg temperature is more dependent on the oil bath temperature. Therefore, to achieve a temperature setpoint through various speeds, it is a combination of heat generated from camshaft speed and oil bath temperature.

### 7.2.3 Screening test 3 – Temperature Distribution (fixed inlet oil temperature)

Screening test 1 and 2 (Section 7.2.1 and 7.2.2 respectively) highlighted that the UniAir Avg temperature was heavily dependent on camshaft speed. To help understand the influence of the oil inlet temperature on the UniAir temperature, the maximum speed is used while maintaining an ambient oil bath temperature. The following controllable factors in Table 13 are applied.

| | |
|---|---|
| Camshaft Speed | 3000rpm |
| Oil Type | Castrol SAE 0W-20 (Matching Schaeffler oil) |
| Oil Bath temperature | 17°C (Ambient oil conditions) |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar (Matching Schaeffler pressure map) |
| Test time | Until UniAir Avg temperature reaches 35°C |

**TABLE 13 - CONTROLLABLE FACTORS FOR SCREENING TEST 3**

Using the controllable factors in Table 13, Figure 71 shows the temperature response of the system. Figure 71 highlights the oil filter temperature (brown line on Figure 71) maintained a temperature of 17°C throughout the test duration. UniAir Avg temperature reached the temperature setpoint in approximately 110 seconds from ambient. Comparably to Figure 69 the UniAir Avg temperature achieved the setpoint temperature in similar time. This is in addition to the oil inlet temperature remaining at constantly lower ambient temperature. This provides further evidence that oil temperature in the UniAir is highly dependent on the camshaft speed in comparison to the oil inlet temperature at higher engine speeds.

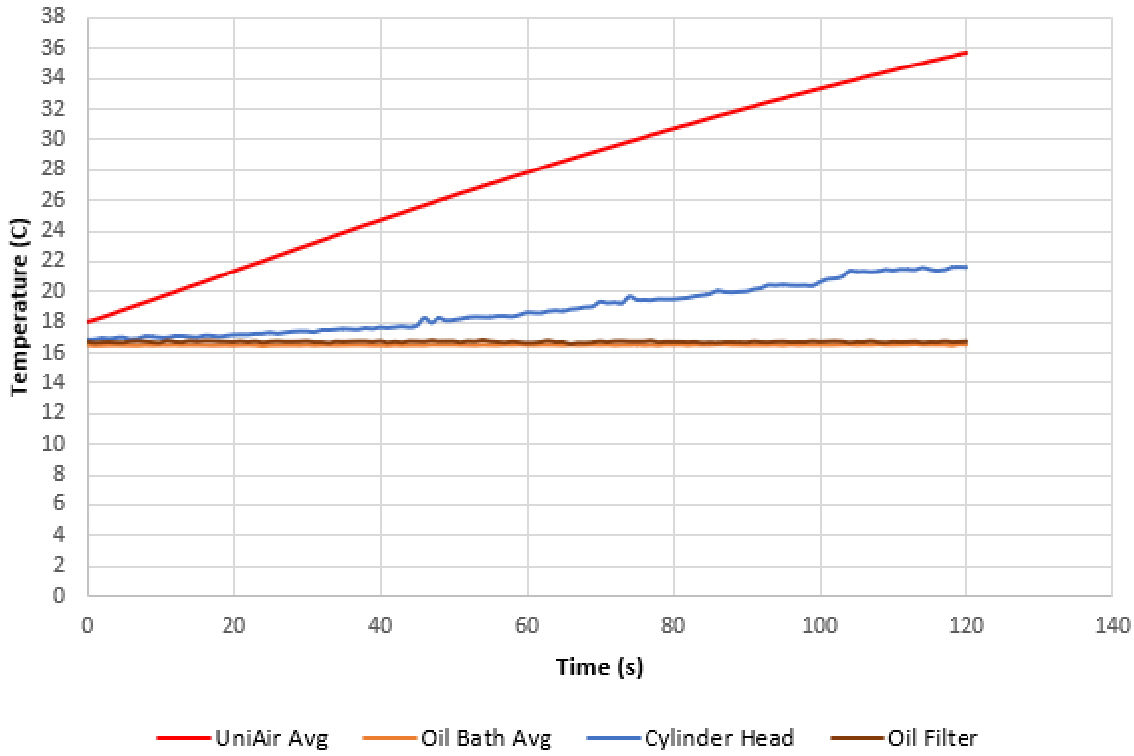


**FIGURE 71 - SCREENING TEST OBSERVING TEMPERATURE RESPONSE OF SYSTEM AT 3000RPM WITH AMBIENT OIL INLET**

### 7.2.4 SCREENING TEST 4 – TEMPERATURE DISTRIBUTION AND COOLING CAPABILITIES

Screening tests 1-3 identified uncontrollable factors of heat loss and internal heat generated in the hot setup. Therefore, to create a testing procedure for all temperatures it is essential to understand how the uncontrollable factors are affected in the cold setup.

The modifications to the freezer reduced the amount of insulation inside and replaced it with the cylinder head, therefore it is expected that the cooling efficiency of the freezer will decrease. The first cold setup screening test is used to understand the capabilities of the freezer and what temperatures it can achieve. Due to health and safety restrictions at the University of Sheffield, the freezer could only be turned on during the operational hours of the laboratory (8:30.a.m – 4:30.p.m). The combination of reduced insulation and restricted times severely restricted the capabilities of the freezer. It was found that after 6 hours, the maximum cooling potential of the UniAir achieved -5°C, and the oil bath achieved 0°C. The findings of heat loss to ambient air/components in the hot setup screening tests, created the expectation that the cold oil would gain heat from the ambient air/components using the oil circuit before supplying the UniAir. Therefore, to achieve an oil supply temperature to below the required set point the oil needed to be cooled for longer. As an alternative method, the oil bath was cooled in a separate freezer which is capable of reaching -27°C and could be used without restricted time.

To understand the limits of the cooling capabilities of the freezer, a camshaft speed of 3000rpm was chosen to generate the highest levels of internal heat. Therefore, the controllable factors in Table 14 are used.

| | |
|---|---|
| Camshaft Speed | 3000rpm |
| Oil Type | Castrol SAE 0W-20 |
| Oil Bath temperature | -27°C |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar (Matching Schaeffler pressure map) |
| Test time | Until UniAir Avg temperature reaches 35°C |

**TABLE 14 - CONTROLLABLE FACTORS FOR SCREENING TEST 4**

The temperature distribution for Screening test 4 in the cold setup at 3000rpm can be seen in Figure 72.

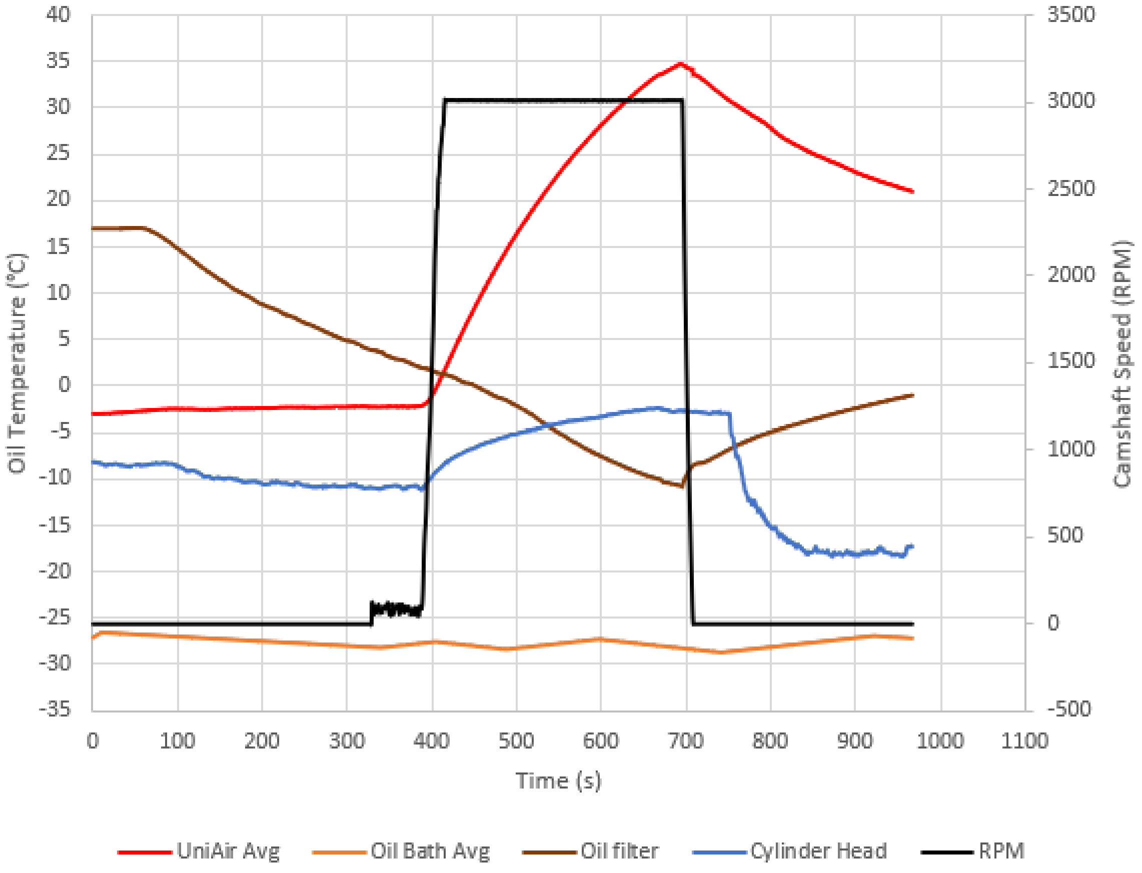


**FIGURE 72 – SCREENING TEST OBSERVING TEMPERATURES IN COLD SETUP AT 3000RPM CAMSHAFT SPEED**

The pump is turned on at approximately 60 seconds, which is illustrated by the oil filter temperature (brown line in Figure 72) decreasing in temperature over time as its pumping -27°C oil from the oil bath (orange line in Figure 72). When the motor is turned on at approximately 400 seconds, the internal heat generated by the UniAir at 3000rpm increases the UniAir Avg temperature (red line in Figure 72). However, contrasting to Screening test 1 and 3 (Section 7.2.1 and 7.2.3 respectively), when the UniAir is at ambient temperature (around 500seconds) it takes approximately 200 seconds for the UniAir Avg temperature to reach 35°C, which is longer than both Screening test 1 and 3 (Section 7.2.1 and 7.2.3 respectively). This suggests that the oil temperature/environment does influence the temperature in the UniAir.

At the point the UniAir Avg temperature reaches 35°C the rig is turned off. The UniAir Avg and Cylinder head temperatures start to decrease due to the colder temperature inside the freezer. Alternatively, the temperature at the oil filter starts to increase as the oil filter is on the outside of the freezer and open to the ambient temperatures in the laboratory.

Schaeffler's test methodology does not measure the oil temperature in the UniAir to analyse the torque characteristics. To fully understand which temperature point is the most relevant in studying the influence of the oil viscosity on the UniAir operational performance and efficiency, a comparison exploring the oil temperature at the oil filter (the equivalent distance of Schaeffler's temperature test point) and the UniAir Avg temperature against torque and valve lift is essential.

## 7.3 OPTIMISATION

The optimisation typically used to "predict the response values for all possible combinations of factors within experimental regions"(51). Specifically, for the work presented here, the screening process has identified uncontrollable factors of heat loss and heat generation on the UniAir, both of which are dependent on oil bath temperature and camshaft speed. The optimal test procedure needs a specific oil bath temperature for a specific camshaft speed. To generate a temperature map of oil bath temperatures, the worst-case situations are considered to account for all testing parameters.

### 7.3.1 HOT SETUP

The worst-case scenario in the hot setup is the lowest camshaft speed of 500rpm (i.e. smallest internal heat generation) at the highest setpoint oil temperature of 120°C (highest heat loss). It was found that the UniAir Avg temperature could not achieve 120°C due to the combination of heat loss from the oil bath/oil filter to the UniAir, and the low heat generated due to the camshaft speed. Therefore, without another source of external heat supplied, 120°C UniAir Avg temperature could not be tested at 500rpm.

One way to achieve the desired oil temperature in the UniAir of 120°C at 500rpm, is to heat the oil to the desired temperature, circulate the oil until the entire cylinder head and UniAir is thermally saturated. However, this is time inefficient and an alternative solution is therefore needed. The most time efficient method is to increase the camshaft speed, so that the internal heat of the UniAir increases the UniAir Avg temperature beyond the setpoint. Subsequently the camshaft speed is reduced to the required testing speed, so the torque and valve lift can be captured as the oil cools into the desired oil temperature range.

Schaeffler's test rig (4.1) has a "cooling water" tempering fluid used for regulation of the cylinder head temperature, which "was not measured". This additional cooling would make the higher temperatures significantly harder to achieve, especially at lower speeds. The screening and optimisation process have identified four important parameters:

- Internal heat
- Heat loss
- Testing time

- Optimised temperature distribution

Due to the position where Schaeffler measure the oil temperature, the testing time of 2 minutes for all speeds, in addition to the unmeasured tempering fluid, would strongly suggest that the temperatures in the UniAir are unpredictably different. This emphasises another methodology is required to fully understand the influence of oil viscosity on the UniAir operation and efficiency.

### 7.3.2 COLD SETUP

The cold setup presented its own set of challenges due to the internal heating of the UniAir. The minimum temperature to be analysed on the rig is 0°C as specified in 6.3. To achieve this parameter, the cylinder head, UniAir and oil inlet (filter) temperature needs be at a starting temperature below 0°C. Therefore, during testing, any internal heat generated during start-up in the UniAir will be accounted for. As discussed in 7.2.4, the cooling capabilities of the freezer enables the components inside (UniAir and cylinder head) to be cooled down to -5°C. However, because of heat loss/gain in the oil circuit, the oil bath is cooled in a separate freezer which is capable of cooling to -27°C. This combination will allow all components to be cooled and tested at the desired temperature of 0°C.

## 7.4 ROBUSTNESS

The robustness objective is typically used to establish that a method can produce stable, reliable and repeatable values. It is also used to understand how to alter the bounds of the factors so that robustness may still be claimed (51). Section 3.5 demonstrates how to calculate a measurement, the confidence level of that measurement and if there is a statistical difference between data sets. However before exploring the final measurement, how the measurement is captured needs to be understood.

### 7.4.1 DATA ANALYSIS

Having identified the controllable/uncontrollable factors and how to gain confidence in the measured systems responses, this section investigates how to process and analyse the raw data. Part of the test methodology requires the data captured for each test to be processed and analysed in the same way, so that direct comparisons can be made. DIAdem, a data analysis software suite developed by National Instruments is used to inspect, analyse and report on measurement data from the rig (52). Figure 73 illustrates an example screenshot from DIAdem showing all of the raw data from one test.

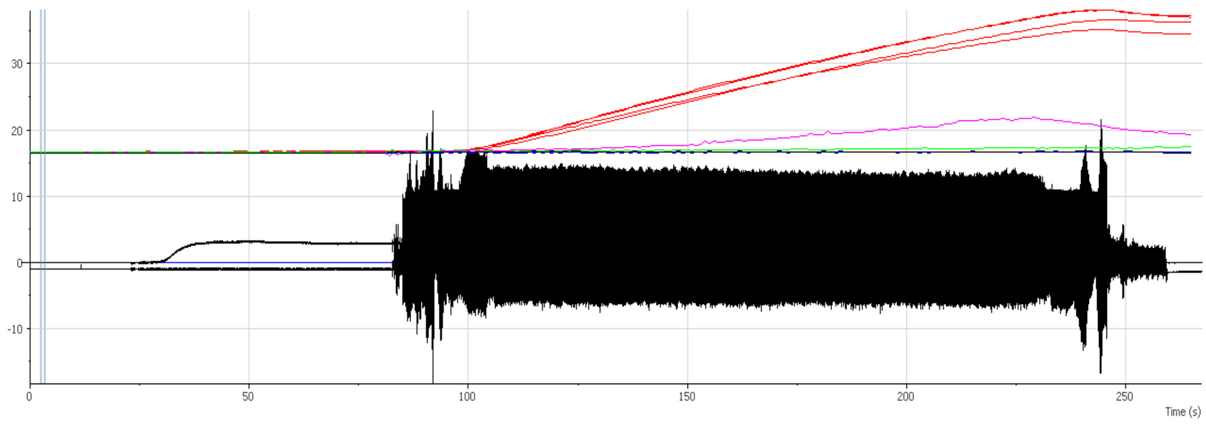


**FIGURE 73 – SCREENSHOT OF EXAMPLE RAW DATA OVER TIME IN DIADEM**

#### 7.4.1.1 CAPTURE TIME

The 'capture time' is the amount of time used to capture a relevant section of the raw data. To ensure that all the tests can be captured at the same time, the worst case scenarios are considered. The capture time is dictated by the test that provides the fastest time to surpass the UniAir Avg temperature setpoint.

##### 7.4.1.1.1 CAPTURE TIME TEST 1

The first worst case scenario shown in Section 7.2.3 is the rate of heat generated at 3000rpm in the hot setup. To capture data at an average UniAir temperature of 35°C, the tolerance of the temperature setpoint is ± 1°C replicating the value previously used by Schaeffler in Section 4.1 (Unless otherwise stated all temperatures will assumed to be ± 1°C). Therefore, the capture time of this test will be dictated by how long the UniAir temperature is in the region of the setpoint ± 1°C. Table 15 shows the controllable factors for capture time test 1.

| | |
|---|---|
| Camshaft Speed | 3000rpm |
| Oil Type | Castrol SAE 0W-20 |
| Oil Bath temperature | Ambient |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar |
| UniAir Avg temperature setpoint | 35°C |

**TABLE 15 - CONTROLLABLE FACTORS FOR CAPTURE TIME TEST 1**

Figure 74 shows the temperature and rpm of the camshaft (black line on Figure 74) against time. It can be seen that the UniAir Avg temperature increases with camshaft speed. At approximately 240 seconds, the UniAir Avg temperature is at the setpoint of 35°C. Using DIAdem, the temperature distribution on the UniAir can be explored further.

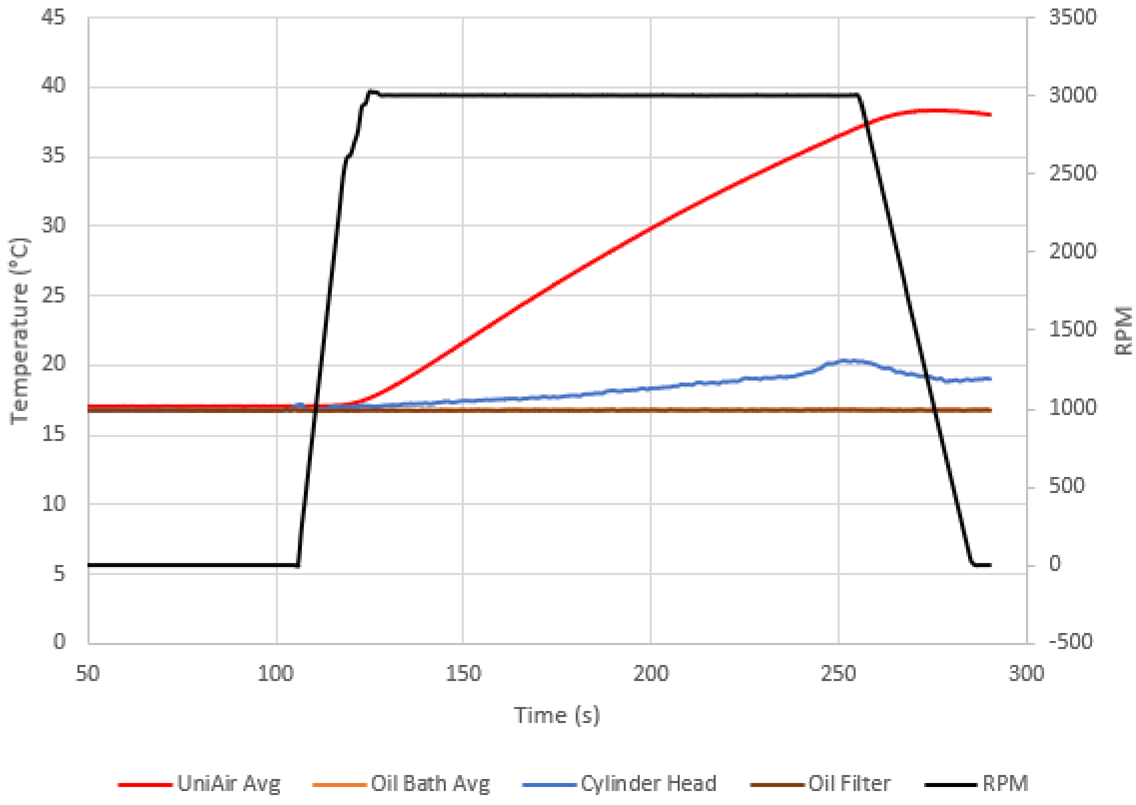


**FIGURE 74 – TEMPERATURE AND RPM AGAINST TIME WITH 3000RPM CAMSHAFT SPEED AND AMBIENT OIL BATH TEMPERATURE**

Figure 75 highlights the time cursor, a tool in DIAdem which is used to capture the raw data between two specific points. In this example, the Y axis is fixed to the UniAir temperature setpoint of 35°C. The time cursor is synchronised across all the raw data, so when isolating the UniAir temperature between these two points it will capture all of the data for this specific temperature.

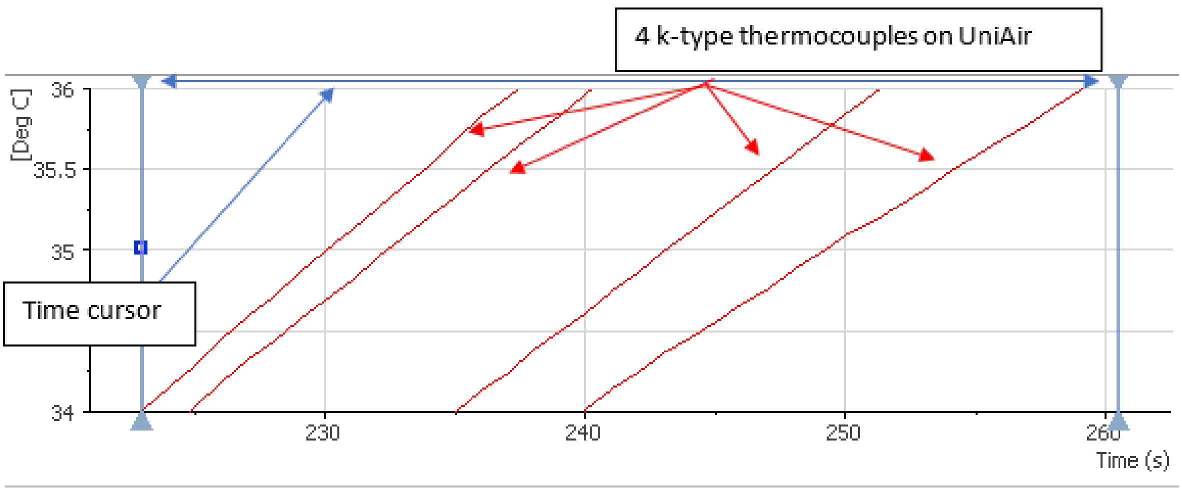


**FIGURE 75 - ISOLATED VIEW SHOWING THE TEMPERATURE OF THE FOUR UNIAIR THERMOCOUPLES AT 3000RPM AT AMBIENT OIL BATH, TEMPERATURE OVER TIME**

Figure 75 shows the temperature distribution of the four individual thermocouples on the UniAir. To attain an average value of 35°C on the UniAir, it takes approximately 35 seconds. The entire data range is then captured during this time so there is a specific data set to the UniAir temperature of 35°C.

This test has identified the capture time at 3000rpm at ambient oil temperature to be approximately 35 seconds. It also reiterates the findings in Section 7.2.1 that the temperature of the oil in the UniAir is heavily dependent on the increase in frictional heating due to the speed of the camshaft, not just the oil inlet temperature.

#### 7.4.1.1.2 Capture time test 2

Capture time test 2 is used to investigate another worst case scenario identified in Section 7.3, which is the rate of heat lost in the hot setup. The controllable factors to replicate this worst case scenario are found (Table 16) at 500rpm while capturing data at 120°C UniAir Avg temperature.

| | |
|---|---|
| Camshaft Speed | 500rpm |
| Oil Type | Castrol SAE 0W-20 |
| Oil Bath temperature | 130°C |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar |
| UniAir Avg temperature setpoint | 120°C |

**TABLE 16 - CONTROLLABLE FACTORS FOR CAPTURE TIME TEST 2**

To achieve the a UniAir Avg temperature of 120°C at 500rpm, the speed of the camshaft was increased to approximately 2000rpm, as this generates excessive internal heat in the UniAir as illustrated in Figure 76. Once the UniAir Avg temperature (red line on Figure 76) is above the setpoint of 120°C, the camshaft speed is changed to 500rpm and the data is captured as the temperature in the UniAir cools.

Figure 76 shows the temperature distribution of the rig when varying the camshaft speed from 500rpm to 2000rpm (black line on Figure 76). When the camshaft speed is increased to 2000rpm, the UniAir Avg temperature increases to above 120°C at approximately 310 seconds. The speed is then reduced to 500rpm at approximately 325 seconds so the temperature in the UniAir cools over the setpoint at the desired speed. In the same process used in Capture time test 1 (Section 7.4.1.1.1), DIAdem is used to analyse the temperature distribution at this time.

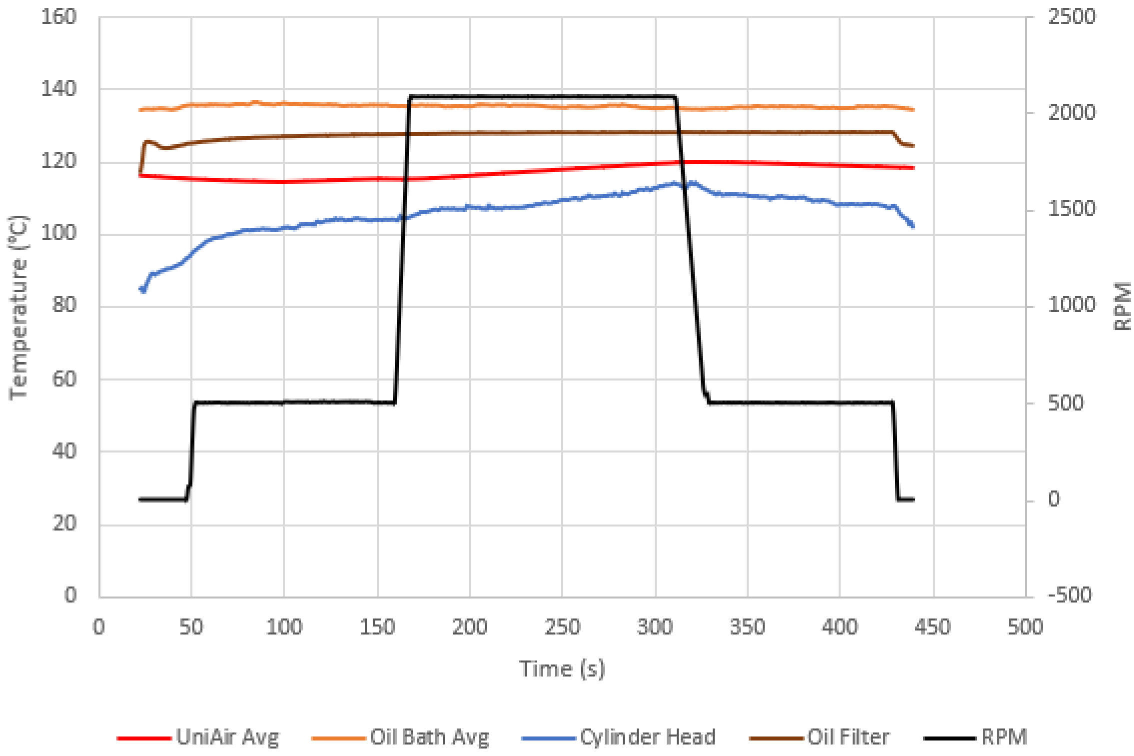


**FIGURE 76 - TEMPERATURE AND RPM AGAINST TIME WHEN VARYING CAMSHAFT SPEED TO ACHIEVE 120°C UNIAIR AVG TEMPERATURE SETPOINT**

Figure 77 indicates the temperature distribution of the UniAir at 120°C as it cools beyond the set point in approximately 55 seconds. Therefore, the capture time is 55 seconds, which is longer than the capture time seen in Capture time test 1 (Section 7.4.1.1.1).

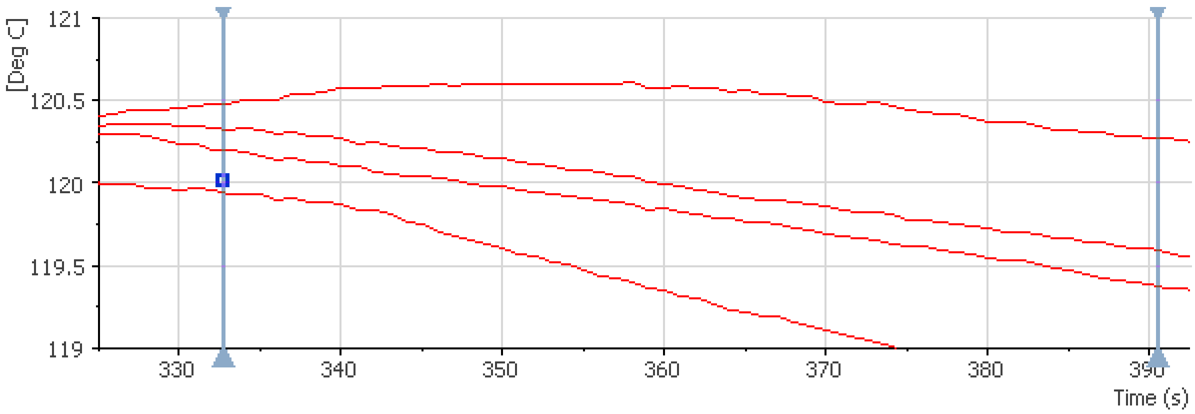


**FIGURE 77 – ISOLATED VIEW USING DIADEM TO LOOK AT THE TEMPERATURE DISTRIBUTION OF THE FOUR UNIAIR THERMOCOUPLES AT 500RPM AT 130°C OIL BATH TEMPERATURE, TEMPERATURE OVER TIME**

This capture time test demonstrates how the UniAir temperature increases and decreases with camshaft speed. This also highlights how the UniAir temperature of 120°C is achieved at 500rpm.

#### 7.4.1.1.3 Capture time test 3

Capture time test 3 is used to investigate another worst case scenario identified in Section 7.2.4 in the cold setup. Similarly to Capture time test 1 (Section 7.4.1.1.1), this test is looking at the heat generated and the time it takes the temperature to exceed the temperature setpoint of 0°C. Therefore, 3000rpm is used to generate the highest levels of heat. The controllable factors used for Capture time test 3 are found in Table 17.

| | |
|---|---|
| Camshaft Speed | 3000rpm |
| Oil Type | Castrol SAE 0W-20 |
| Oil Bath temperature | -27°C |
| Lift mode | Full lift |
| Oil Pressure | 3 Bar |
| UniAir Avg temperature setpoint | 0°C |

**TABLE 17 - CONTROLLABLE FACTORS FOR CAPTURE TIME TEST 3**

Figure 78 illustrates the motor is turned on (camshaft speed represented by the black line in Figure 78) when the oil filter temperature (brown line on Figure 78) was at the same temperature as the UniAir Avg (red line on Figure 78) . At 1020 seconds the UniAir Avg temperature is at the setpoint of 0°C. DIAdem is used to look at the temperature distribution on the UniAir to understand the capture time.

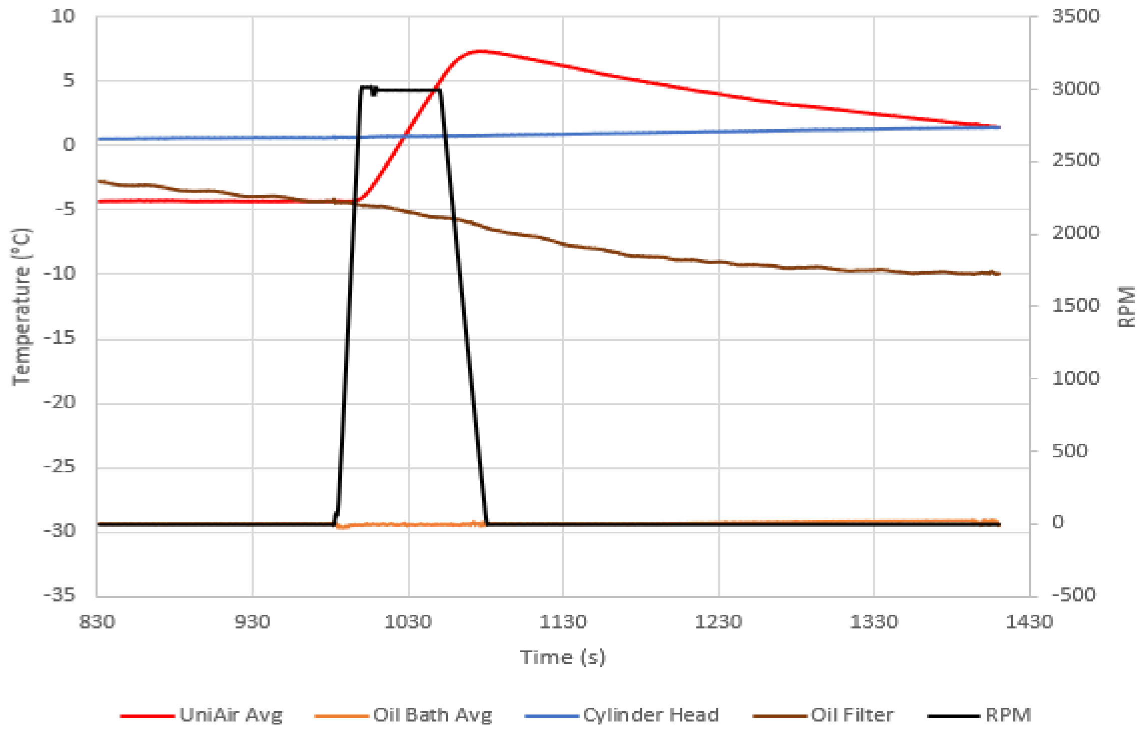


**FIGURE 78 - TEMPERATURE AND RPM AGAINST TIME AT 3000RPM CAMSHAFT SPEED AT -30°C OIL BATH TEMPERATURE**

Figure 79 indicates that the capture time is approximately 15 seconds and therefore the fastest capture time in comparison to Capture time tests 1 and 2 (Section 7.4.1.1.1 and 7.4.1.1.2 respectively). This dictates that the capture time in the test procedure for each test is 15 seconds, as this is fastest time for the UniAir temperature to exceed the setpoint. During this capture time all of the sensors are measured. This creates a data set specific to a UniAir temperature of torque, valve lift, pressure, and camshaft speed. This helps to understand the influence of oil viscosity on the UniAir operation (valve lift deviation and leak rates) and efficiency (relative power losses), described in the objectives defined in Section 1.3.

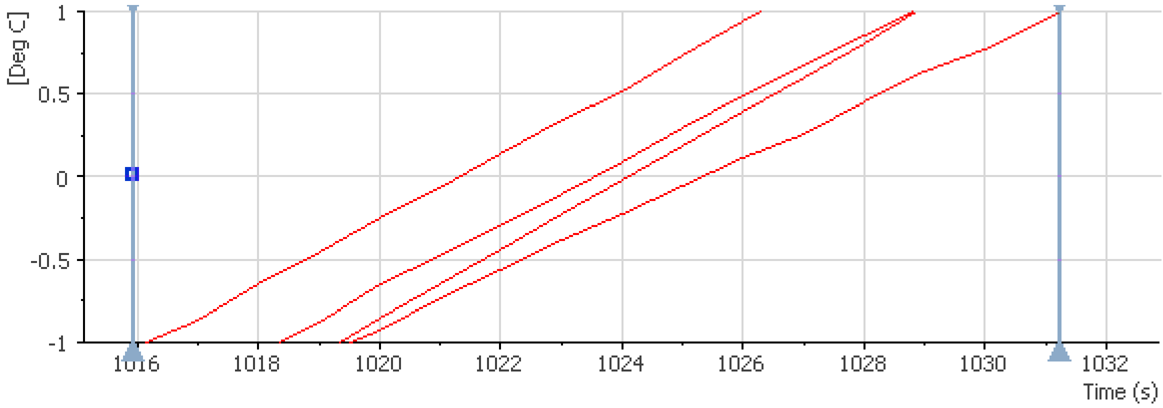


**FIGURE 79 - TEMPERATURE DISTRIBUTION OF UNIAIR AT 3000RPM CAMSHAFT SPEED AND -30°C OIL BATH TEMPERATURE**

#### 7.4.1.2 Torque Characteristics

The motivation for the objectives (Section 1.3) is to understand the influence of oil properties on the UniAir efficiency. This is achieved by analysing the torque response of the UniAir over various input parameters. Understanding the torque characteristics facilitates how the data is captured for comparisons between tests. Using DIAdem, Figure 80 shows the torque characteristics of the UniAir at 3000rpm in Full Lift mode.

Figure 80 illustrates how the torque characteristics change when the speed of the camshaft increases to 3000rpm. It can be seen the torque curve fluctuates above and below zero which was also observed in Schaeffler's results (Section 4.1). This is because of the way the camshaft interacts with the UniAir in Full Lift mode. As each individual cam lobe pushes against the pumping element in the UniAir, there is a required torque to push the oil and open the valve, which results in a positive torque response. As the cam lobe goes beyond peak lift as discussed in Section 2.4.1, the valve spring pushes the oil back against the cam lobe resulting in a negative torque response. This process is replicated for every individual cam lobe and repeated for every camshaft revolution resulting in the varying sinusoidal torque characteristics observed in Figure 80. As the torque curve changes with speed it is essential to understand at what point the torque becomes stable or consistent once the target speed of the camshaft is achieved.

Reducing the X axis in Figure 80 to a smaller time scale emphasises the torque characteristics further. Figure 81, illustrates the smaller time scale of the torque characteristics at 3000rpm. At 913 seconds, the camshaft speed achieves the target speed of 3000rpm. However, it takes approximately 2 seconds for the torque to stabilize after the speed is constant. This agrees with the findings in Section 4.1, Figure 29. Schaeffler discovered that during a speed ramp there are regular spikes of torque in the UniAir. To ensure repeatability in the results, the data was captured when the torque is stable.

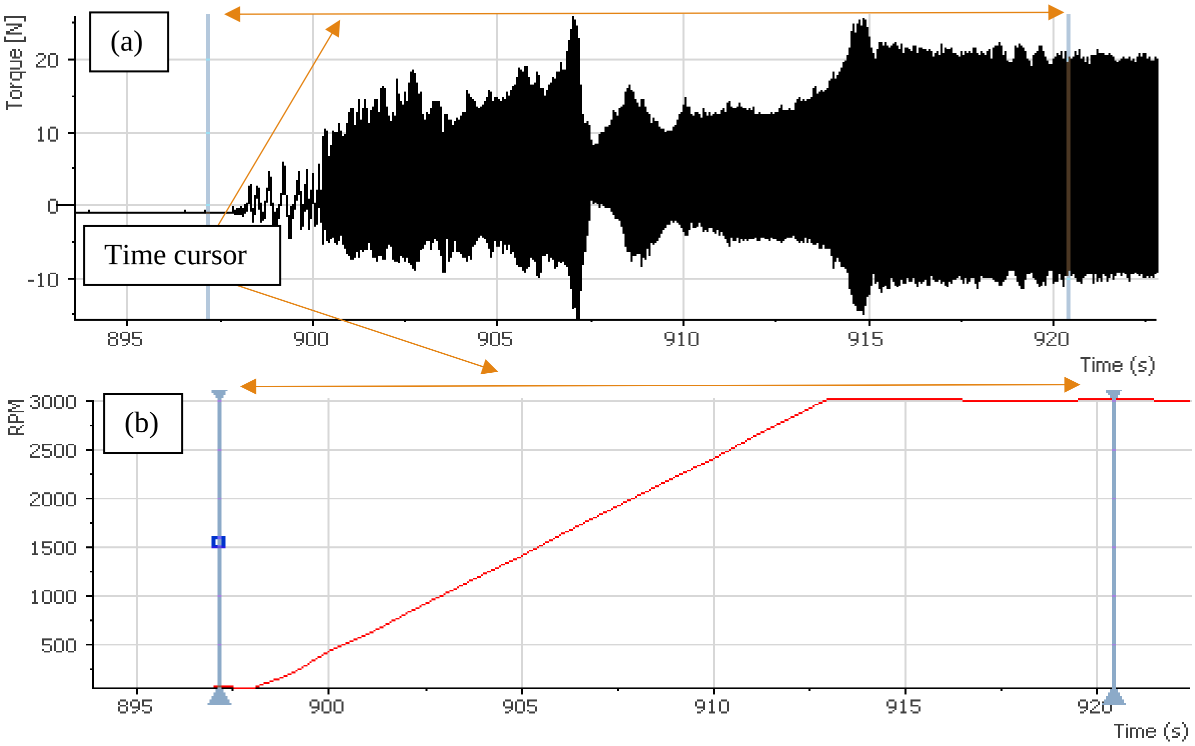


**FIGURE 80 - TORQUE CHARACTERISTICS OF UNIAIR (A) AND SPEED OF CAMSHAFT (B) AT 3000RPM WITH FULL LIFT.**

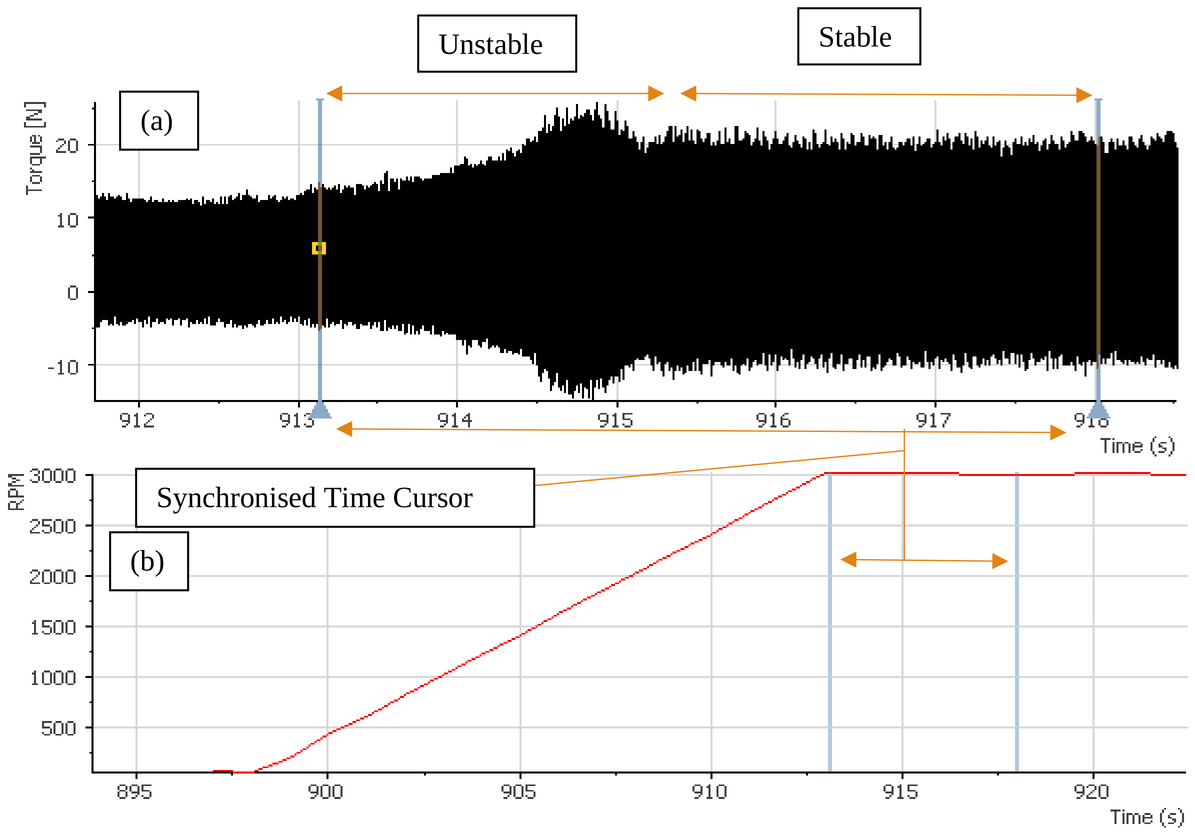


**FIGURE 81 - TORQUE CHARACTERISTICS OF UNIAIR (A) AND CAMSHAFT SPEED (B) AT 3000RPM FULL LIFT MODE**

To further understand how to determine the stability of the torque and UniAir, the torque profile at 500rpm and Full Lift was then analysed as demonstrated in Figure 82. Figure 82 shows the speed and torque characteristics of the UniAir at 500rpm. It can be seen that the speed becomes stable at 790 seconds, at which point the torque also appears to be stable. In this example the stability of the torque is identified when the peak torque values are within 5%. Once stability is achieved, the torque selected in the capture time was then averaged to give a single value of torque for a specific data set.

This section highlights two parameters (stable speed and torque) which are considered when capturing a valid data set for a specific UniAir Avg temperature. The UniAir Avg temperature must be within $\pm 1°C$ of the temperature setpoint and the torque must be stable.

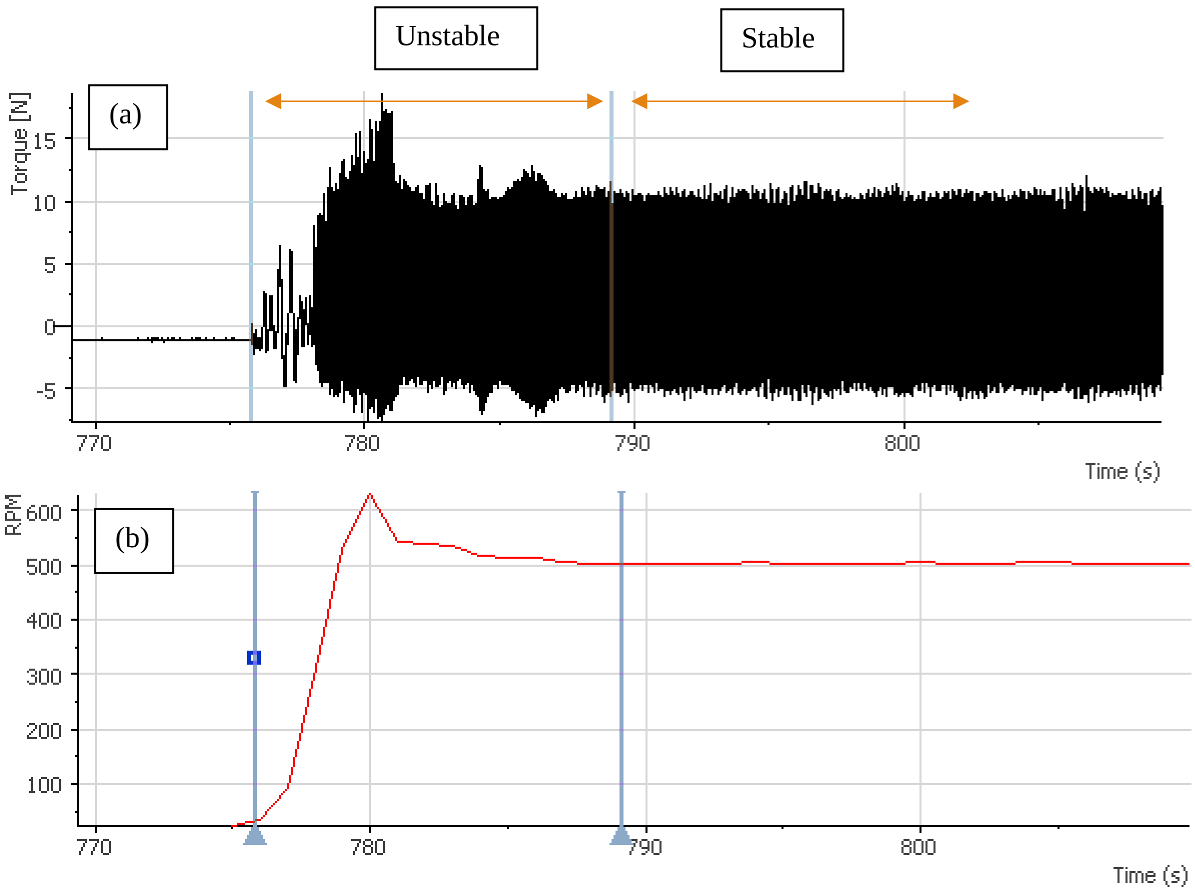


**FIGURE 82 - TORQUE CHARACTERS OF UNIAIR (A) AND CAMSHAFT SPEED (B) AT 500RPM FULL LIFT MODE**

7.4.1.3 VALVE LIFT

To achieve the objective of understanding the influence of oil viscosity on the UniAir valve lift deviation and leak rates, it is fundamental to ensure the valve lift is captured in a robust and repeatable way. There are eight valves on the cylinder head test rig to actuate the UniAir. However, only five Keyence lasers were purchased due to the budget as mentioned in Section 6.6, Table 9. Consequently, only 4 valves are measured per test. In the interest of time, the same valves are measured every test so that direct comparisons are made. Every revolution will generate four valve lift profiles from the four lasers (6.6). Looking at the data from capture time test 2, using DIAdem the four valve lift profiles from one revolution can be seen in Figure 83.

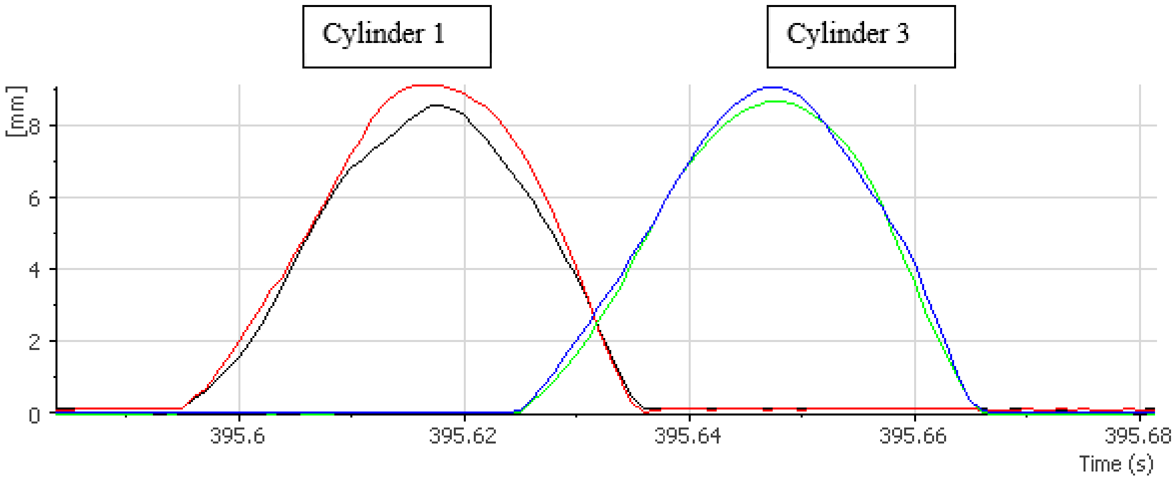


**FIGURE 83 - FULL LIFT PROFILES FROM CAPTURE TIME TEST 2 DATA, EACH COLOUR REPRESENTS A DIFFERENT VALVE LIFT PROFILE FOR ONE REVOLUTION**

Figure 83 displays two full lift profiles on cylinder 1 and two full lift profiles on cylinder 3 at 500rpm and 120°C UniAir oil temperature. Section 3.5.2.1 states how to understand the measured value in terms of variation and repeatability. To do this, the maximum lift of each valve lift profile is required. Figure 84 demonstrates how DIAdem is used for the calculation of "Find Peaks", which finds the maxima peak value of every valve lift profile.

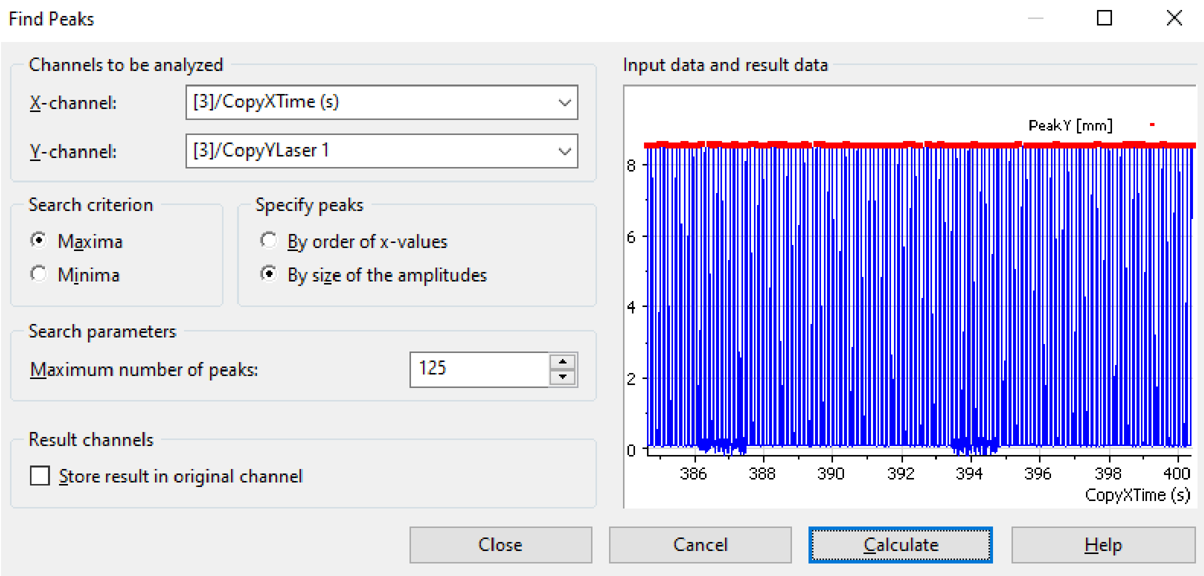


**FIGURE 84 - DIADEM 'FIND PEAKS' TOOL USED TO FIND THE PEAK LIFT OF EACH VALVE LIFT PROFILE**

Dependent on the camshaft speed, this dictates how many peak values are found. For example, at 500rpm with a 15 second capture time will have 125 valve lift profiles on each laser. At 3000rpm with a 15 second capture time, will have 750 valve lift profiles on each laser.

In Figure 84, 125 peaks were found as it is looking at the Capture time test 2 data, which is at 500rpm. DIAdem then produces a 1x125 matrix of all the peak valve lift profiles in this data set. This value is then averaged to give an averaged peak valve lift profile for a specific laser. Repeating this procedure for each laser will then give the required information to calculate the variation and repeatability (14.4).

Table 18 shows the averaged peak valve lift profile for a specific laser. Laser 1 (L1) and laser 2 (L2) measured valve lift on cylinder 1, while laser 3 (L3) and laser 4 (L4) measured valve lift on cylinder 3. Using Schaeffler's technique to determine valve lift variation and repeatability (14.4), the average valve lift for cylinder 1 (L12) and cylinder 3 (L34) was calculated. Variation is defined as the maximum difference of the mean valve lift of two valves on one cylinder against the mean valve lift of two valves on another cylinder. In this particular example, a value of 0.08mm variation was calculated which is within the tolerance of 0.8mm variation for full lift at 120°C oil temperature and 375rpm (14.4).

| L1 (mm) | L2 (mm) | L3 (mm) | L4 (mm) | L12 (mm) | L34 (mm) | Variation (mm) | LA (mm) |
|---|---|---|---|---|---|---|---|
| 8.41 | 8.94 | 8.51 | 9.00 | 8.68 | 8.75 | 0.08 | 8.71 |

**TABLE 18**

To enable more holistic comparisons between tests, L12 and L34 are averaged (LA in Table 18). This generates one valve lift value for a specific camshaft speed, oil type, oil pressure and UniAir oil temperature.

### 7.4.2 Testing for Robustness

As discussed in Section 7.2, Schaeffler's test methodology does not measure the oil temperature in the UniAir as part of their analysis of the torque characteristics. To fully understand which temperature point is the most appropriate in studying the influence of the oil viscosity on the UniAir operational performance and efficiency, a comparison exploring the oil temperature at the oil filter (the equivalent distance along the lubricant flow path to Schaeffler's temperature test point) and the UniAir Avg temperature against torque and valve lift is essential.

#### 7.4.2.1 Standard uncertainty and confidence in test results

Section 3.5.1 explains to gain confidence in the robustness of the rig and its results, The National Physics Laboratory suggest finding the uncertainty of measurement with at least 3 individual data points (17). Therefore, the data acquired from Screening tests 1, 3 and 4 (Sections 7.2.1, 7.2.3 and 7.2.4 respectively) will be used, as it considers the performance of the UniAir in the same conditions, other than varying oil inlet temperatures. This determines which temperature point is the most appropriate by studying the influence of the oil viscosity on the UniAir performance and efficiency. To achieve this, the average peak valve lift, torque and oil filter temperature from each test, is captured at the average UniAir temperature points of 25°C, 30°C and 35°C. Using Equation 10 and 11 the standard uncertainty and confidence level is calculated. The results confirm the rig's robustness by statistically demonstrating the confidence and significance of the results.

Table 19, Table 20 and Table 21 show the results from Screening test 1, 3 and 4 (Section 7.2.1, 7.2.3 and 7.2.4 respectively) of average valve lift, torque and oil filter temperature at 25°C, 30°C and 35°C UniAir Avg temperature. There are three important parameters to consider when analysing the results:

- Standard uncertainty and confidence level in comparison to provided data and literature
- Relationship between torque and valve lift against oil filter and UniAir temperature
- Relationship between each UniAir temperature point, showing effect on torque and valve lift

| Screen Test | Oil Filter temperature (°C) | UniAir Temperature (°C) | Average Valve lift (mm) | Average Torque (Nm) |
|---|---|---|---|---|
| 1 | 24.488 | 25.119 | 10.785 | 3.848 |
| 3 | 16.714 | 25.203 | 10.783 | 3.851 |
| 4 | -7.353 | 25.275 | 10.788 | 3.868 |
| Mean | - | - | 10.785 | 3.856 |
| Stand Deviation | - | - | 0.002 | 0.011 |
| Standard uncertainty | - | - | 0.001 | 0.006 |
| Confidence level (95%) | - | - | 0.003 | 0.013 |
| Result | - | - | 10.785 ± 0.003 | 3.856 ± 0.013 |

**TABLE 19 - COMPARISON OF SCREEN RESULTS AT 25°C UNIAIR TEMPERATURE SHOWING TORQUE AND VALVE LIFT**

| Screen Test | Oil Filter temperature (°C) | UniAir Temperature (°C) | Average Valve lift (mm) | Average Torque (Nm) |
|---|---|---|---|---|
| 1 | 25.846 | 30.061 | 10.770 | 3.707 |
| 3 | 16.723 | 30.017 | 10.777 | 3.728 |
| 4 | -9.045 | 30.013 | 10.772 | 3.748 |
| Mean | - | - | 10.773 | 3.728 |
| Stand Deviation | - | - | 0.004 | 0.02 |
| Standard uncertainty | - | - | 0.002 | 0.012 |
| Confidence level (95%) | - | - | 0.004 | 0.024 |
| Result | - | - | 10.773 ± 0.004 | 3.728 ± 0.024 |

**TABLE 20 - COMPARISON OF SCREEN TEST RESULTS AT 30°C UNIAIR TEMPERATURE SHOWING TORQUE AND VALVE LIFT**

| Screen Test | Oil Filter temperature (°C) | UniAir Temperature (°C) | Average Valve lift (mm) | Average Torque (Nm) |
|---|---|---|---|---|
| 1 | 27.253 | 35.135 | 10.710 | 3.570 |
| 3 | 16.731 | 34.902 | 10.723 | 3.550 |
| 4 | -10.478 | 34.895 | 10.733 | 3.610 |
| Mean | - | - | 10.722 | 3.577 |
| Stand Deviation | - | - | 0.011 | 0.031 |
| Standard uncertainty | - | - | 0.007 | 0.018 |
| Confidence level (95%) | - | - | 0.013 | 0.035 |
| Result | - | - | 10.722 ± 0.013 | 3.577 ± 0.035 |

**TABLE 21 - COMPARISON OF SCREEN TEST RESULTS AT 35°C UNIAIR TEMPERATURE SHOWING TORQUE AND VALVE LIFT**

##### 7.4.2.1.1 Optimal oil temperature measurement position

To understand which position is the most appropriate in studying the performance of the UniAir, in both torque and valve lift, the results in Table 19, Table 20 and Table 21 were analysed. To do this the magnitude of the valve lift and torque results needed to be understood. It was expected that the results in this rig will differ to Schaeffler's in Section 4.1 (as Schaeffler used a non-production UniAir in their test programme in comparison to the production level UniAir used in this thesis), however, Schaeffler (Section 4.1.2) highlights there is no specific tolerance for the results in torque on the UniAir. It was found that Schaeffler's rig (Section 4.1) was classified as "run in" once the results of torque were "constant" at ±0.1Nm. Therefore, this value will be considered to validate the rig in this thesis.

When comparing this value of ±0.1Nm to the results in Table 19, Table 20 and Table 21, it can be seen that the highest level of error in torque found in this rig, is ±0.035N.m in Table 21. This would suggest that this rig is very robust, as it is within Schaeffler's tolerance for constant torque values. Furthermore, this data is captured entirely based on the temperatures in the UniAir. Therefore, the low level of error in the torque results in Table 19, Table 20 and Table 21 would strongly suggest that the thermocouples on the UniAir are the most appropriate position when analysing the performance of the UniAir. In addition to this, the alternative position at the oil filter has a difference in oil temperature of approximately 30°C, suggesting that there is little to no correlation with torque.

Likewise with the torque values, in Schaeffler's test results for valve lift (Section 4.2.2), no tolerances were specified. The only supplied information regarding valve lift tolerances are found in the Section 14.4. It can be seen that the repeatability tolerances in the Full Lift table in Section 14.4 are based on oil entry temperature of 120°C. Therefore, direct comparisons cannot be made as the results (in Table 19, Table 20 and Table 21) are observed at temperatures of 25°C 30°C and 35°C. However, the magnitude of these tolerances can be used to understand the repeatability in the results.

The Full lift table in Section 14.4 highlight that the valve lift tolerance is ±0.2mm at 6200rpm (i.e. 3100rpm camshaft). In comparison to the results in Table 19, Table 20 and Table 21, the highest level of valve lift error is ±0.013mm. Again, this would suggest that the rig is very robust, as it is within Schaeffler's tolerance. However, it is important to note that there is no alternative tolerance for valve lift, at the testing conditions observed in Table 19, Table 20 and Table 21. This further reiterates the need for a new rig as the thermocouples on the UniAir are the most appropriate position when analysing the performance of the UniAir.

For both valve lift and torque the standard uncertainty will be displayed in the results via error bars. This enables the visual reference of the rigs robustness when comparing results.

##### 7.4.2.1.2 Statistical difference in the results

Using the data in Table 19, Table 20 and Table 21, this allows the calculation of a t test, which as explained in Section 3.5.4, is used to create a quantitative value to understand if there is a significant difference in the results. A t test requires two sets of data to create a p value. Therefore, comparing the results of valve lift and torque in:

1. Table 19 and Table 20
2. Table 19 and Table 21
3. Table 20 and Table 21

Using Microsoft Excel, a paired t test can be performed to generate the data required to understand the significant difference. For example using a paired t test on the valve lift data samples in Table 19 and Table 20 creates the following information:

| | *Variable 1* (Table 19) | *Variable 2* (Table 20) |
|---|---|---|
| Mean | 10.785 | 10.773 |
| Variance | 0.000 | 0.000 |
| Observations | 3.000 | 3.000 |
| Pearson Correlation | -0.606 | |
| Hypothesized Mean Difference | 0.000 | |
| df | 2.000 | |
| t Stat | 3.879 | |
| P value (T<=t) one-tail | 0.030 | |

**TABLE 22 - EXAMPLE OF T TEST ON VALVE LIFT DATA FROM TABLE 19 AND TABLE 20**

Table 22 illustrates that a p value of 0.03 is calculated, which is lower than a commonly used threshold of 0.05 alpha and therefore is 95% confident that the results are significantly different.

To fully understand the statistical difference between test results, t tests are done on all combinations of the results found in Table 19, Table 20 and Table 21.

| | Table 10 and Table 11 | | Table 10 and Table 12 | | Table 11 and Table 12 | |
|---|---|---|---|---|---|---|
| | Valve Lift | Torque | Valve Lift | Torque | Valve lift | Torque |
| P value | 0.03 | 0.001 | 0.004 | 0.001 | 0.007 | 0.004 |

**TABLE 23 - T TEST ON EVERY COMBINATION FROM RESULTS IN TABLE 10, 11 AND 12 ON VALVE LIFT AND TORQUE**

Table 23 highlights the p values from using a t test between the data found in Table 19, Table 20 and Table 21. It can be seen that every p value is lower than 0.05, which defines a 95% confidence that all results are significantly different. This reiterates that the rig is robust and repeatable.

## 7.5 TEST PROCEDURE FOR UNIAIR "RELATIVE HYDRAULIC EFFICIENCY"

To satisfy the requirements of the overall project, the UniAir operation and efficiency needed to be determined. Section 5.5.1 stipulates there are two factors affecting hydraulic efficiency, mechanical efficiency, and volumetric efficiency.

Mechanical efficiency is dependent on the frictional losses within the device and the energy required to generate flow through it. This is calculated using equation 12. However, to calculate the mechanical efficiency, a theoretical efficiency is required. Due to the complexity of the UniAir's design, theoretically calculating the torque over the various parameters tested is not plausible. Therefore, the theoretical torque is replaced by a measured 'baseline torque' from the results. The baseline torque value is the lowest torque value found for a specific speed. This demonstrates the relative efficiency across the viscosities tested at each speed compared to the lowest captured torque value. Therefore, by adopting equation 12, a 'relative mechanical efficiency' can be calculated to understand the difference in mechanical efficiencies for a specific speed at different UniAir Avg temperatures.

$$n_m = \frac{baseline\ torque}{measured\ torque} = relative\ mechanical\ efficiency\ (\%) \tag{28}$$

Volumetric efficiency is the ratio of actual flow to theoretical flow, or flow losses often due to leakage. To calculate the volumetric efficiency, equation 13 is used. However, the UniAir is not like a traditional pump as the volume of oil being displaced, is used to push a valve, and not specifically circulate oil. Consequently, the theoretical flow is actually a theoretical valve displacement. However, as described in Section 7.4.2 there is no published data to dictate a theoretical valve displacement. Hence, similarly to the theoretical torque the theoretical valve lift is replaced by a measured 'baseline valve lift' from the results. The baseline valve lift value is the largest valve lift measured for a specific speed. This enables the calculation of the 'relative volumetric efficiency' which demonstrates a quantifiable way of comparing valve lift against changing oil viscosities. Therefore, by adopting equation 13 the relative volumetric efficiency is calculated to understand the difference in volumetric efficiencies for a specific speed at different UniAir Avg temperatures.

$$n_v = \frac{baseline\ valve\ lift}{measured\ valve\ lift} = relative\ volumetric\ efficiency\ (\%) \tag{29}$$

The hydraulic efficiency of a hydraulic component (Section 5.5.1) is the product of mechanical and volumetric efficiencies (equation 14). Therefore, adopting equation 14, the 'relative hydraulic efficiency' can be calculated:

$$relative\ hydraulic\ efficiency = n_v \times n_m \tag{30}$$

Calculating the relative hydraulic efficiency combines both parameters torque and valve lift which are used to understand the influence of oil viscosity on the UniAir operation and efficiency.

## 7.6 Development of Test procedure and plan

From the evidence presented in Chapter 7, this has demonstrated sufficient information to create a test procedure to ensure repeatability and robustness. It has also described how the captured data will be analysed and processed across all tests so that direct comparisons can be made. The test procedure is found in 14.8.

The test plan highlights the order and number of tests carried out. Table 24 illustrates for 0W20 oil type, the number of tests studied with the symbol ✓. As described in Section 7.4.2.1, at least three tests are required to generate a standard uncertainty and statistical difference within the results. Consequently, each test (✓) is completed three times. This results in a total of 180 tests per oil type.

Table 24 illustrates the test plan for one oil type, and the test plan is replicated for every oil tested. As described in Section 6.3, three oils are studied in this thesis, therefore, this equates to a minimum of

540 tests in total. Due to high number of tests the decision was made to reduce this amount by fixing inlet oil pressure and limiting testing camshaft speeds. If an additional two oil pressures points and two camshaft speeds were considered a further 1728 tests would need to be conducted to generate standard uncertainty and statistical difference within the results. Consequently, the overwhelming number of tests was beyond the timescale of this thesis.

Table 3 in Section 4.1, highlights the speed related pressure map used in Schaeffler tests. Therefore, the oil pressure of 3 bar was selected as it provides enough oil pressure for the highest camshaft speed tested at 3000rpm in this thesis. Subsequently having a fixed oil pressure allows direct comparisons between camshaft speed and oil temperatures.

Furthermore, LIVO lift mode (Section 4.1.1.3) is limited to camshaft speeds of 1500rpm or less. Therefore, the camshaft speeds above 1500rpm were less of a priority than those under 1500rpm. 3000rpm had not been tested by Schaeffler, therefore is required for this thesis in generating novel results.

| 0W20 | | | | |
|---|---|---|---|---|
| Camshaft speed (rpm) | Full Lift | No Lift | EIVC | LIVO |
| 0°C | | | | |
| 500 | ✓ | ✓ | ✓ | ✓ |
| 1000 | ✓ | ✓ | ✓ | ✓ |
| 1500 | ✓ | ✓ | ✓ | ✓ |
| 3000 | ✓ | ✓ | ✓ | N/A |
| 35°C | | | | |
| 500 | ✓ | ✓ | ✓ | ✓ |
| 1000 | ✓ | ✓ | ✓ | ✓ |
| 1500 | ✓ | ✓ | ✓ | ✓ |
| 3000 | ✓ | ✓ | ✓ | N/A |
| 90°C | | | | |
| 500 | ✓ | ✓ | ✓ | ✓ |
| 1000 | ✓ | ✓ | ✓ | ✓ |
| 1500 | ✓ | ✓ | ✓ | ✓ |
| 3000 | ✓ | ✓ | ✓ | N/A |
| 120°C | | | | |
| 500 | ✓ | ✓ | ✓ | ✓ |
| 1000 | ✓ | ✓ | ✓ | ✓ |
| 1500 | ✓ | ✓ | ✓ | ✓ |
| 3000 | ✓ | ✓ | ✓ | N/A |

Each point (✓) tested 3 times

**TABLE 24 - TESTING PLAN HIGHLIGHTING NUMBER OF TESTS PER OIL TYPE**

### 7.6.1 ENGINE CONTROL UNIT – LIFT MODES

Table 24 has highlighted the test plan, identifying how many tests per individual lift modes. As presented in Section 2.4.1.1, this describes how the lift modes are controlled. The engine control unit (ECU) is used to control the UniAir and govern which lift mode is enabled. Full Lift and No Lift have limited parameters to control the lift, as it is either full lift, or no lift. However, EIVC and LIVO have more parameters to govern the lift by changing at which point of the cam event the solenoids open or close for a certain crank angle (CA°).

As highlighted in Table 24 there are a high number of tests required for this work. Therefore, to maintain the number of tests outlined in Section 7.6, the CA° for both EIVC and LIVO must be fixed. Furthermore, to validate the rig for these lift modes, the results will be compared to previous experiment and simulated tests. However, there is no previous experimental or simulated tests for valve lift in LIVO mode, therefore the CA° will be dictated by the UniAir lift tolerances stated in Section 14.4. This leads to the opening CA° for LIVO will be set at 420°, while EIVC will be set at 540°. This will enable the results to be compared to the UniAir tolerances and therefore validating the rig for these lift modes. Furthermore, by limiting the testing conditions for these 4 lift modes only initially will identify potential conditions of specific interest for future work.

## 7.7 SUMMARY & IMPACT ON CURRENT WORK

Chapter 7 has systematically illustrated the novel test methodology used in the work described in this thesis. It has highlighted the main uncontrollable factors of the system process while demonstrating the controllable factors and responses of the rig. The uncontrollable factor of heat loss is a critical area of interest for analysing the performance of the UniAir. It has been observed from the preliminary screening tests that there is a substantial amount of heat lost from the oil between components on the rig. This suggests that due to this heat loss, the most appropriate place to measure the temperature of the oil is as close to the component being tested. Consequently, four k-type thermocouples were instrumented on the UniAir to gather more information with regards to temperature distribution from the heat loss.

Further screening tests demonstrated another critical finding, internal heat in the UniAir during operation. Internal heat has not been reported in previous tests done by Schaeffler, and thus further testing is required to understand the relationship between the oil temperature/viscosity and UniAir performance. The combination of both uncontrollable factors, heat loss and internal heat, have identified how to achieve the required temperatures of 0°C - 120°C set out in the scope of this thesis.

The test methodology has also described how the data will be captured consistently throughout the various different parameters and inputs to be tested in this thesis. Additionally, it has emphasized the rigs robustness and repeatability by statistically proving a 95% confidence in the preliminary results. This has reiterated the paucity of knowledge regarding the performance of the UniAir under all conditions.

Consequently, this has directed this thesis to gain a better understanding of the UniAir with the base 0W20 oil currently used in the production AJ200 engines. The new test methodology enables greater depth in its analysis over a wider range of temperatures to fully understand the performance and efficiency of the UniAir. Furthermore, further testing is conducted with different oils, which allows a thorough study, to help generate knowledge for additional understanding of how to improve the performance and attempting to facilitate the overall goal of reducing emissions.

# 8. RESULTS AND DISCUSSION

In this chapter, all the experimental results are captured as outlined in Chapter 7 and presented and examined in detail. To fully investigate and evaluate the results, they are presented in a systematic way to align with the aim and objectives for the thesis found in Section 1.3. To do this, the results are separated into four categories based on lift modes.

- Full lift
- No Lift
- EIVC
- LIVO

Within each category, the following will be investigated

1. Validation from literature, previous experimental and simulated results
2. Influence and optimal performance and efficiency of oil viscosity
3. Influence and optimal performance and efficiency of different lubricants

## 8.1 FULL LIFT RESULTS

To validate the valvetrain test rig, results are compared to the literature and previous experimental and simulated results. To do this, the results from 0W20 are analysed first as previous experimental and simulated studies used the same 0W20 in their tests.

### 8.1.1 VALIDATING 0W20 FULL LIFT RESULTS

This section analyses the valve lift and torque characteristics of the UniAir in Full Lift mode. To validate and satisfy the requirements of the overall project, comparisons are made with the literature, previous experimental and simulated tests (Chapters 4 and 5).

The boundary conditions outlined in Section 6.3 are implemented for the Full Lift tests using 0W20 oil and are presented in Table 25.

| | |
|---|---|
| Oil temperature measurement point (UniAir Avg) | 0°C, 35°C, 90°C, 120°C ±1°C |
| Oil Viscosity (cSt) | 313.3, 50.5, 9.8, 5.5 |
| Oil Pressure | 3 bar ± 0.1bar |
| Camshaft speeds | 500, 1000, 1500, 3000 rpm ±10rpm |
| Oil Type | Castrol SAE 0W-20 |

**TABLE 25 – BOUNDARY CONDITIONS OF 0W20 FULL LIFT TESTS**

The results for Full Lift tests using 0W20 oil are found in Table 26. This identifies a torque and valve lift value specific to the UniAir Avg oil temperature and camshaft speed. The baseline torque and valve lift (Section 7.5) have been illustrated in Table 26 (yellow boxes), which are used in later calculations.

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 313.305 | 50.541 | 9.783 | 5.541 |
| Measured Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Baseline torque value | 500 | 1.19 | 0.75 | 1.16 | 1.53 |
| | 1000 | 1.83 | 1.06 | 1.35 | 1.66 |
| | 1500 | 2.44 | 1.46 | 1.54 | 2.04 |
| | 3000 | 4.60 | 3.59 | 3.03 | 3.26 |
| Measured Valve Lift (mm) | | | | | |
| Baseline valve lift value | 500 | 10.42 | 10.29 | 9.67 | 8.68 |
| | 1000 | 10.47 | 10.40 | 10.03 | 8.68 |
| | 1500 | 10.51 | 10.51 | 10.13 | 10.05 |
| | 3000 | 10.78 | 10.72 | 10.52 | 10.27 |

**TABLE 26 – FULL LIFT TEST RESULTS OF 0W20 OIL, SHOWING TORQUE AND VALVE LIFT FOR VARYING VISCOSITY AND CAMSHAFT SPEEDS**

#### 8.1.1.1 Validating Full valve lift

This section analyses the valve lift characteristics of the UniAir while using Full Lift mode. Figure 86 illustrates the data from Table 26 showing the relationship between valve lift against camshaft speed while varying UniAir Avg oil temperatures.

It is observed in Figure 86 that valve lift increases with increasing camshaft speed across all UniAir Avg oil temperatures studied. Additionally, valve lift increases with decreasing oil temperature across all camshaft speeds studied. Koehler and Jackson (35) agree with this finding as they also found higher viscosity characteristics greatly improved volumetric efficiency. It is demonstrated for hydraulic pumps (Section 5.5.1), loss of the volumetric efficiency is produced by high temperature low viscosity lubricants bypassing pump clearances causing leakages.

Figure 17 in Section 2.4.1.1.1 illustrates the oil flow within the UniAir when generating Full Lift valve lift. To further understand how the theorized reduced valve lift of oil leakages observed in Figure 86, Figure 85 shows areas where oil leakage can occur during valve lift. During valve lift, the oil circuit is closed (black dashed line in Figure 85) and therefore has three areas where oil leakage can occur in the pumping element, valve stem and HLA (yellow dashed circles in Figure 85).

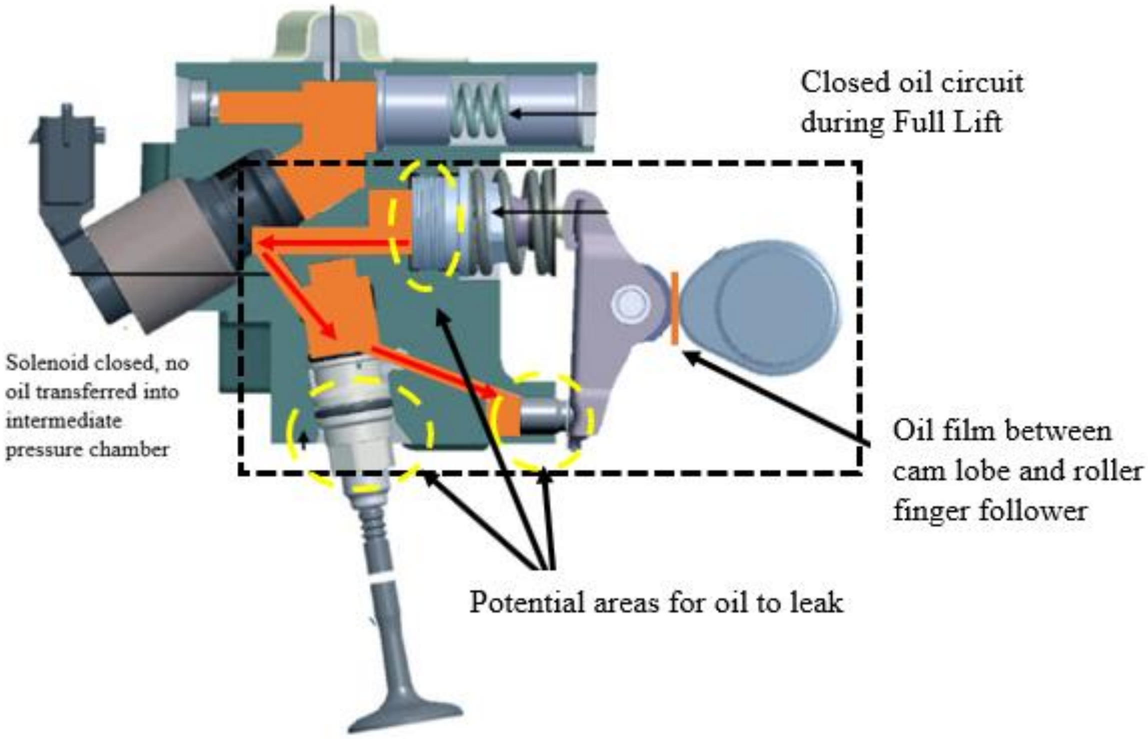


**FIGURE 85 - CROSS SECTION OF UNIAIR SHOWING OIL CIRCUIT AND POTENTIAL AREAS FOR OIL LEAKAGE WITH SOLENOID CLOSED**

Valve lift is generated by oil pressure from the pumping action actuated from the lobe on the camshaft. Figure 86 identifies at low camshaft speeds and high oil temperature, the valve lift deteriorates. This suggests at low camshaft speeds the oil pressure is not generated quickly enough to fully counteract the oil leakages suggested in Figure 85 for oil temperatures at 90°C and 120°C. Alternatively, for UniAir Avg oil temperatures at 0°C and 35°C where oil viscosities are higher, Figure 86 demonstrates how the results have high correlation, particularly at higher camshaft speeds. This is theorized due to the likelihood for higher viscosity oils, the leakage rate is lower, and the oil pressure generated is transferred directly to the valve causing higher valve lift.

The errors bars in Figure 86 identify the standard uncertainty of the results. Where the standard uncertainty of two data sets overlap, this illustrates where the results are not statistically different. This occurs at two points in Figure 86, at 1500rpm and 3000rpm between 0°C and 35°C valve lift curves. As Koehler and Jackson (35) state that higher viscosity characteristics greatly improved volumetric efficiency, it is assumed that at these overlapping points is the maximum possible valve lift of the system for the given speed.

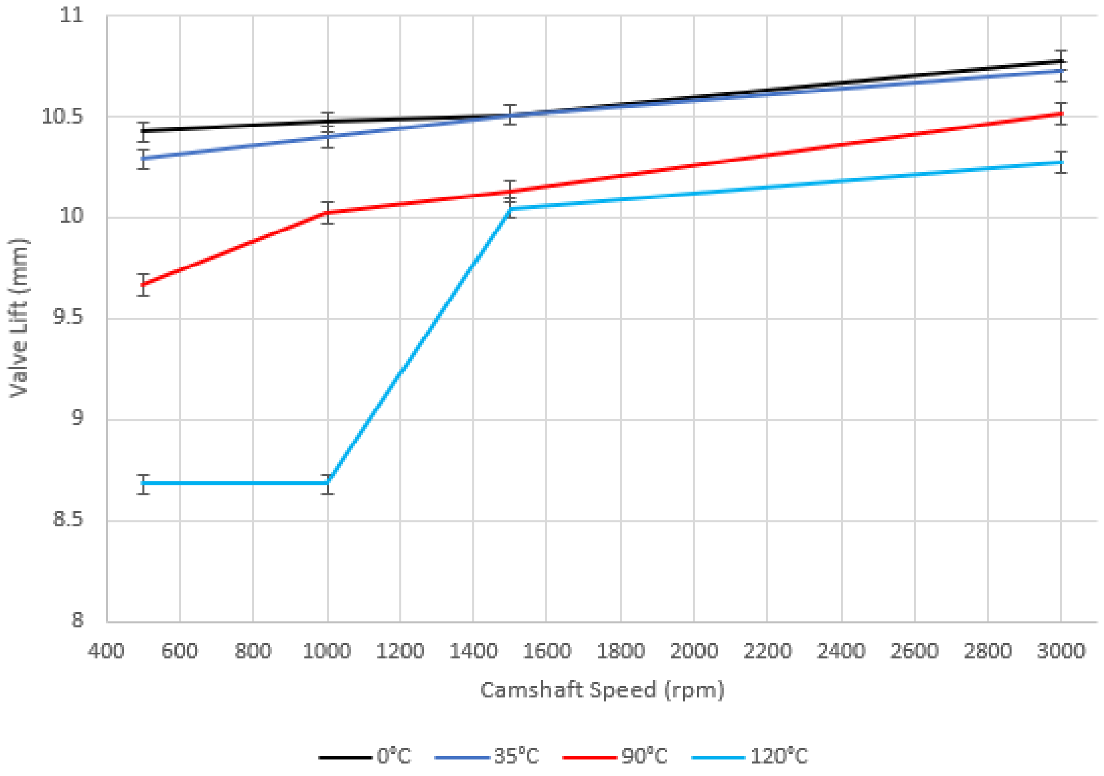


**FIGURE 86 - VALVE LIFT AGAINST CAMSHAFT SPEED TEST RESULTS FOR FULL LIFT MODE, USING 0W20 OIL**

The simulated results (Section 4.2.1), Figure 34 shows the simulated valve lift at 375rpm and 3100rpm camshaft speeds using 0W20 oil, while varying oil temperature. To visualise this, the valve profiles from the 0W20 Full Lift results are compared to the simulated results from JLR. Figure 87 illustrates the valve lift profiles of the Full Lift results at 500rpm, the closest camshaft speed measurement point in comparison to simualted results at 375 rpm.

Figure 86 identified at low camshaft speeds and high UniAir Avg oil temperatures, the valve lift deteriorates. Figure 87 highlights this further showing how varying oil temperatures change the valve lift profiles. In addition to the reduced valve lift height due to oil leakages at higher oil temperatures, the start of the valve lift is delayed. This further demonstrates for camshaft speeds at 500rpm, pressure is not generated fast enough to counteract the oil leakages for oil temperatures at 90°C and 120°C. As the camshaft rotates further onto the opening flank of the camshaft, the pressure builds up significantly faster due to the increased gradient of profile of the cam lobe. This additional pressure creates oil flow inside the UniAir to be higher than the leakage, therefore generating valve lift. Figure 34 (a), agrees with the findings at this camshaft speed, that lower viscosity oil causes lower valve lift, and higher viscosity oil causes higher valve lift. However, the simulated study does not take into consideration the leakages during the start of the valve lift. Figure 34 (a) suggests at all oil temperatures studied, the valve lift profiles start opening at the same point. This emphasizes the limitations of simulated studies.

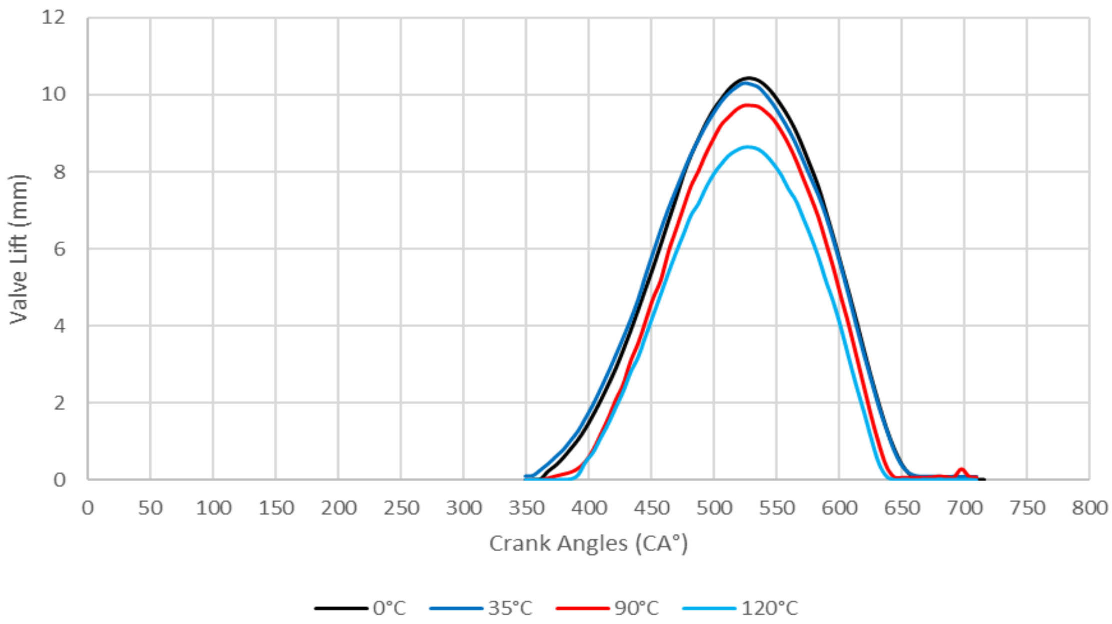


**FIGURE 87 - VALVE LIFT PROFILES OF FULL LIFT MODE AT 500RPM WITH VARYING OIL TEMPERATURE**

Figure 88 illustrates the valve lift profiles of the Full Lift results at 3000rpm, the closest camshaft speed measurement point in comparison to simualted results at 3100 rpm. Figure 88 illustrates higher camshaft speeds generate higher values of valve lift as observed in Figure 86. Furthermore, it agrees with the findings in Figure 34 (b), as higher camshaft speeds cause valve lift values to be more consistent across the operating temperature range. This is because the camshaft speed dictates how fast the cam lobe engages with the UniAir. The higher camshaft speeds generate pressure faster than low camshaft speeds. Consequently, the oil leakages have a smaller influence on the overall valve lift at higher camshaft speeds, as the oil has less time per revolution (i.e. time on base circle of the cam lobe) to leak out.

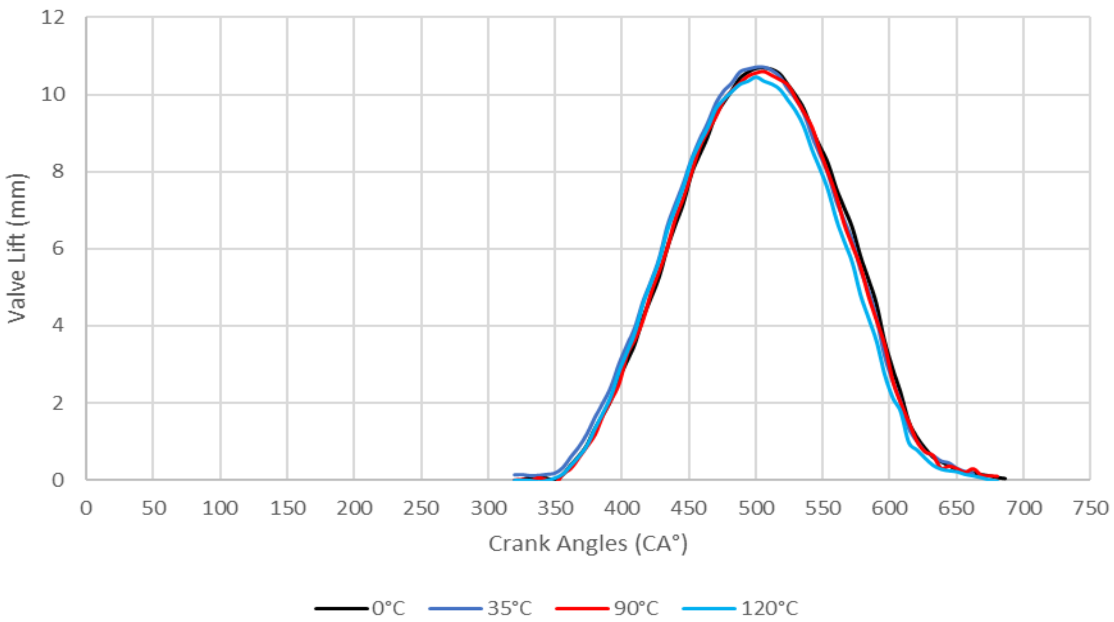


**FIGURE 88 - VALVE LIFT PROFILES OF FULL LIFT MODE AT 3000RPM WITH VARYING OIL TEMPERATURE**

The results agree with Koehler and Jackson (35) and have a high correlation with previous simulated results from JLR. Additionally, it has illustrated the benefits of experimental testing in comparison to simulated testing, as the results have identified another region in the valve lift profile where oil leakage occurs. Consequently, this has given a greater understanding of how oil viscosity influences the valve lift efficiency, but more importantly the results provide evidence in the objective of validating the rig with previous simulated tests.

#### 8.1.1.2 Validating Full Lift Torque

This section analyses the torque characteristics of the UniAir while using Full Lift mode. Figure 89 illustrates the data from Table 26, showing the torque characteristics of the UniAir, in Full lift mode, while varying UniAir Avg oil temperatures and camshaft speeds.

It was found in Figure 89 that increasing camshaft speed increased the torque across all viscosities studied. It is observed that the torque profiles are dependent on UniAir Avg oil temperature and change in camshaft speeds. Sections 3.3 and 4.1.1.1 highlight this is due to dynamic viscosity and changing lubrication regime. Dynamic viscosity is the absolute resistance to shear, therefore, with increasing viscosity increases the required energy to shear the oil. This is shown in Figure 89 at high camshaft speeds the higher viscosity oil requires more torque to shear. However, at low camshaft speeds the highest torque is produced from low viscosity oils at 120°C UniAir Avg oil temperature.

The Stribeck curve illustrates the behaviour observed in Figure 89 is partially due to the changing lubrication regime. The lubrication regime is dependent on the dynamic viscosity, moving part speed and load/pressure. Therefore, when the oil temperature is at 120°C, at low camshaft speeds, this high friction is theorized due to the likelihood of incomplete fluid film formation. The entraining velocity at lower speeds does not form sufficiently thick oil films to fully separate the mechanical interfaces. Furthermore, as camshaft speeds increase the torque profile at 120°C in relation to other temperatures decreases. This is because moving part speed enables the entraining velocity to create an oil film thick enough to change the lubrication regime to mixed.

This is further illustrated by the torque levels when the UniAir Avg oil temperature is at 35°C. At low camshaft speeds the torque is lowest at 35°C, suggesting the oil viscosity generates a mixed lubrication. However, with increasing the camshaft speed, the lubrication regime changes to hydrodynamic where the torque levels increase beyond torque values at 90°C and 120°C becoming second highest. This is due to the extra energy required to shear the higher viscosity oil at this temperature. This is further illustrated as the highest torque levels at 3000rpm camshaft speed, is the highest viscosity at 0°C. The results show that for the whole speed range of the engine application, the optimal engine oil operating temperature is between 35°C and 90°C but no higher for UniAir friction.

Previous experimental test results (Section 4.1.1.1) illustrates the valve lift in full lift mode of the non-production UniAir, with varying camshaft speed and oil temperatures. Although the oil temperature measurement point is different (as discussed in Chapter 7), it can be observed when comparing Figure 31 (JLR experimental torque curves in Full Lift mode) and Figure 89 there are some similarities. To gain a better understanding of this comparison, the results from Figure 31 for Full Lift torque is plot on the same graph as the torque results from Figure 89. Furthermore, the results from 0°C are not included to allow direct comparisons between tests.

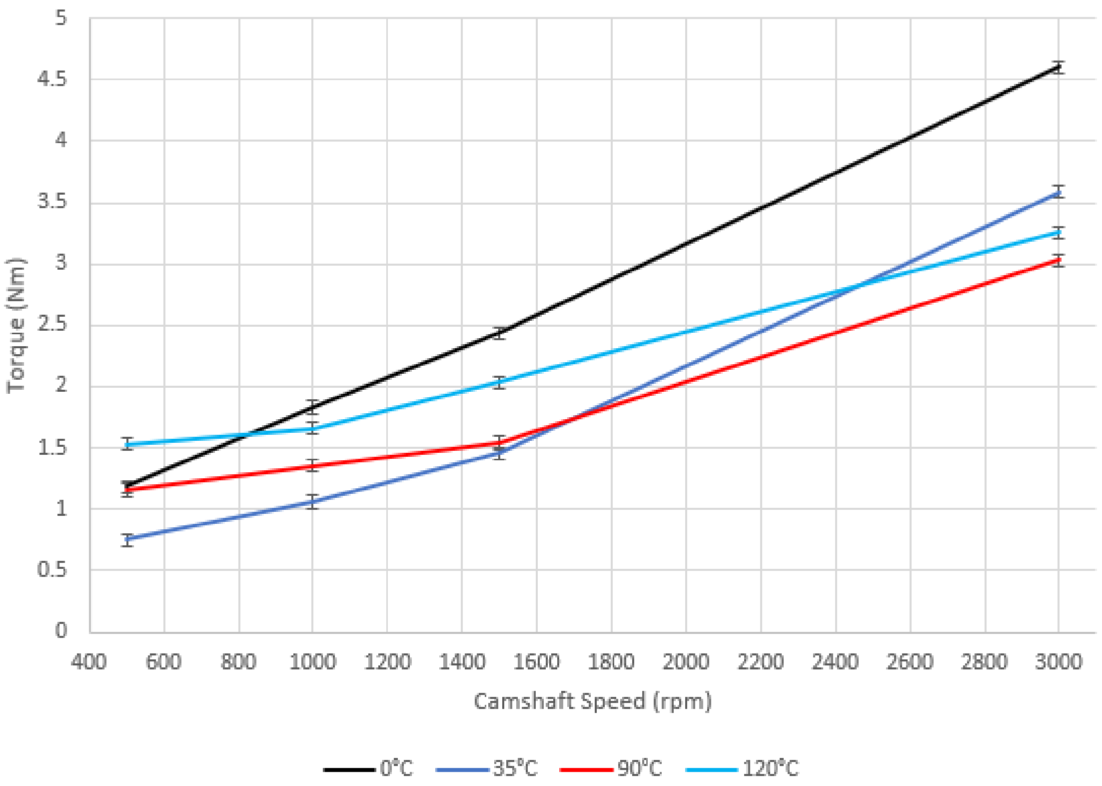


**FIGURE 89 - TORQUE AGAINST CAMSHAFT SPEED TEST RESULTS FOR FULL LIFT MODE, USING 0W20 OIL**

Figure 90 shows the torque characteristics of both the Full Lift results from Table 26 and the results from Schaeffler in Figure 31. When comparing the torque characteristics at low camshaft speeds, the lowest torque is at 35°C and highest at 120°C for each data set. Likewise, at high camshaft speeds, the highest torque is at 35°C while the lowest at 90°C for each data set. The relationship between torque, oil temperature and camshaft speed in Full lift mode are very similar. This would provide evidence that this rig is behaving in an expected manner and thus validating the rig.

However, when comparing the specific values of torque between each data set, highlights differences. At low camshaft speeds, temperatures at 35°C, 90°C, 120°C the Full Lift results are approximately all 0.5Nm lower than Schaeffler results. This difference could be to do with the oil pressure supplied to the UniAir. As the results in Table 26 are referenced to an oil pressure of 3 bar across all camshaft speeds studied. Schaeffler varied the oil pressure dependant on camshaft speed (Section 4.1, Table 3) to replicate the working conditions of the engine. However, the differences could be entirely down to the UniAir Schaeffler tested, being non-production, and therefore design/manufacturing changes on the production UniAir is the reason for changes in torque. Additionally, it could be due to the different testing methodologies implemented, as strong evidence shown in Chapter 7, highlights the different oil temperature measurement Schaeffler's system used and therefore torque measurement.

When observing the highest camshaft speeds tested by Schaeffler in Figure 90 at 2500rpm, the torque profiles at 35°C and 90°C show a high correlation between results from this work (Table 26). At these speeds, Schaeffler's oil pressure matches the oil pressures used in this thesis, therefore giving evidence the difference at lower speeds could be due to this difference in oil pressure. However, when comparing the torque profiles at 120°C the results do not correlate.

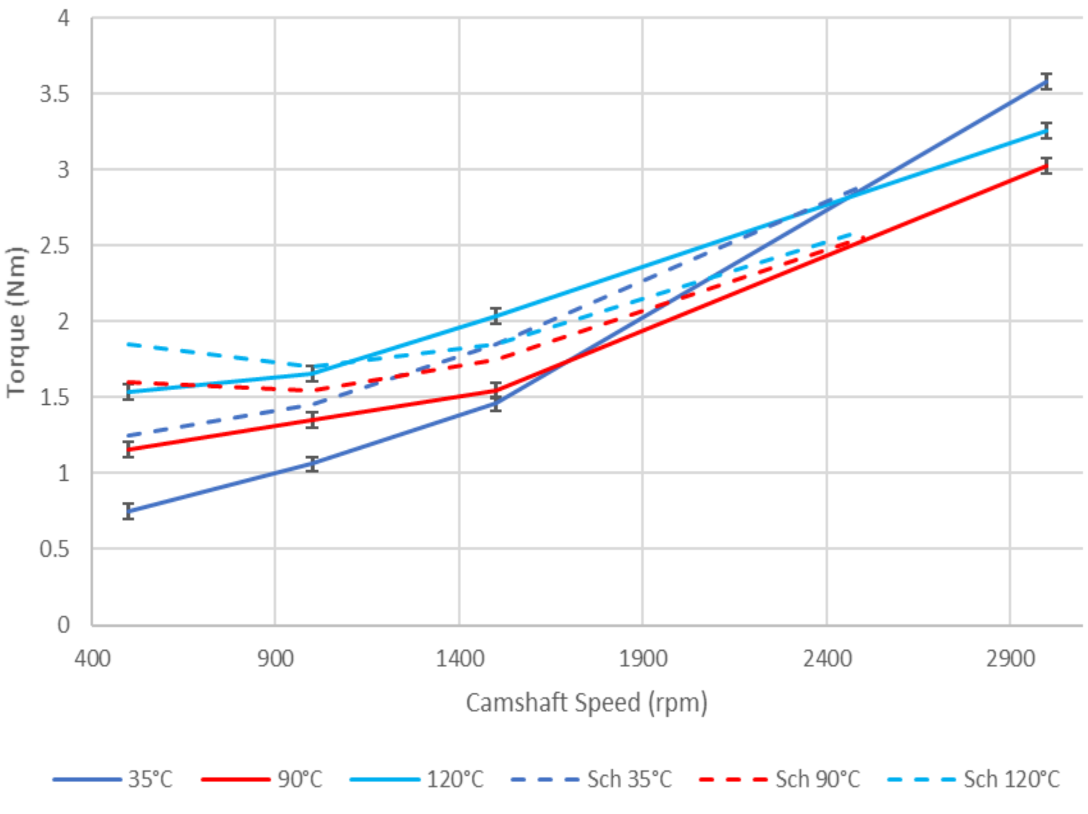


**FIGURE 90 - FULL LIFT TORQUE AGAINST CAMSHAFT SPEED COMPARING RESULTS FROM PREVIOUS EXPERIMENT RESULTS**

Chapter 7 highlights the most difficult oil temperature to achieve in this rig is at 120°C, especially at lower camshaft speeds due to less internal heating. Chapter 7 also highlights in Schaeffler's test rig, the actual temperature in the UniAir in comparison to their measurement point is unpredictably different. Therefore, Figure 90 could suggest the actual temperature for Schaeffler's 120°C torque profile is closer to 90°C, as the results are of similar magnitude. This gives additional evidence for the requirement of a new test methodology as the values of torque are different, and therefore this test methodology has highlighted new information with regards to understanding the optimal performance of the UniAir.

#### 8.1.1.3 Validating Full Lift mode characteristics

Chapters 5, 6 and 7 have demonstrated how to evaluate the influence of oil viscosity on the UniAir operation and efficiency by monitoring the torque and valve lift. However, to understand how to calculate the efficiency of the valvetrain, the behaviour of the system is analysed.

Kovach et al (Section 5.2) demonstrated the torque in traditional valvetrain camshafts. It can be seen in Figure 37 the torque curve decreases with increasing camshaft speed. This is explained to be because of the boundary lubrication at low camshaft speed, and hydrodynamic lubrication at high camshaft speed. This is also the condition for continuously variable valvetrains from Hyundai Motor Company (Section 5.3.2, Figure 39). This illustrates at lower camshaft speeds generates higher levels of torque, which decreases with increasing camshaft speed. However, when observing the torque characteristics of the UniAir in Figure 89, increasing camshaft speed increased the torque across all viscosities studied. This would suggest that the torque behaviour of the UniAir cannot be analysed in the same way as traditional valvetrains.

Alternatively, when considering the torque characteristics of an internal gear pump (Figure 44, Section 5.5.1), this shows a similar relationship to the UniAir torque characteristics. Likewise, with the internal gear pump, for low pressure applications, torque increases with increasing motor speed. Additionally, at low motor speeds the higher friction is dominated by low viscosity oils. Furthermore, when increasing the motor speed, the higher viscosity torque exceeds the lower viscosity torque. This behaviour is also being observed in Figure 89 and would strongly suggest that the UniAir's torque characteristics is dominated by the hydraulic pumping elements. Therefore, to quantify the influence of the oil viscosity on the UniAir, the hydraulic efficiency used in pumps is used to understand the performance and efficiency as demonstrated in Section 7.5.

The relative hydraulic efficiency is calculated using the baseline torque and valve lift values in Table 26, and equations 28-30. Table 27 shows the calculated relative mechanical, volumetric and overall efficiencies.

| Camshaft Speed (rpm) | Viscosity (cSt) | Relative Mechanical Efficiency | Relative Volumetric Efficiency | Relative Hydraulic Efficiency (%) |
|---|---|---|---|---|
| 500 | 313.3 | 0.634 | 1 | 63.4 |
| | 50.5 | 1 | 0.987 | 98.7 |
| | 9.8 | 0.651 | 0.927 | 60.4 |
| | 5.5 | 0.490 | 0.832 | 40.9 |
| 1000 | 313.3 | 0.578 | 1 | 57.9 |
| | 50.5 | 1 | 0.993 | 99.3 |
| | 9.8 | 0.783 | 0.957 | 75.0 |
| | 5.5 | 0.639 | 0.828 | 53.0 |
| 1500 | 313.3 | 0.598 | 1 | 60.0 |
| | 50.5 | 1 | 0.999 | 99.9 |
| | 9.8 | 0.945 | 0.963 | 91.1 |
| | 5.5 | 0.715 | 0.955 | 68.4 |
| 3000 | 313.3 | 0.657 | 1 | 65.8 |
| | 50.5 | 0.844 | 0.994 | 84.0 |
| | 9.8 | 1 | 0.975 | 97.6 |
| | 5.5 | 0.929 | 0.953 | 88.6 |

**TABLE 27 - TABLE OF EFFICIENCIES FOR FULL LIFT USING 0W20 OIL**

Calculating the relative hydraulic efficiency combines both parameters torque and valve lift which are used to understand the influence of oil viscosity on the UniAir operation and efficiency.

### 8.1.2 INFLUENCE OF OIL VISCOSITY ON UNIAIR PERFORMANCE AND EFFICIENCY IN FULL LIFT

The calculated relative hydraulic efficiency (Section 8.1.1, Table 27) of the UniAir while in Full Lift mode is used to understand the influence of oil viscosity on the performance and efficiency and optimal performance of the UniAir. Figure 91 illustrates the overall performance of the UniAir by combining both torque and valve lift to calculate the relative hydraulic efficiency. This creates a quantitative way of comparing the performance of the UniAir at different oil temperatures and camshaft speeds.

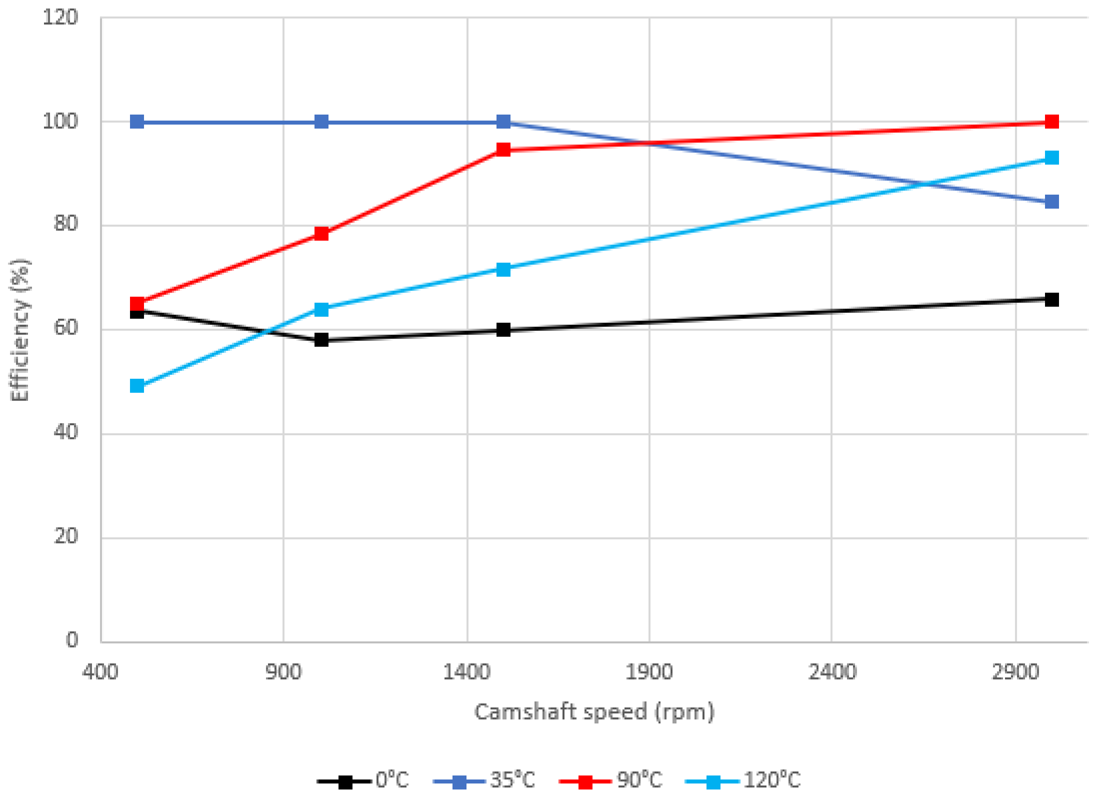


**FIGURE 91 - RELATIVE HYDRAULIC EFFICIENCY AGAINST CAMSHAFT SPEED**

Figure 91 demonstrates that the efficiency of the UniAir changes dependant on camshaft speed. At low camshaft speeds, the UniAir Avg temperature at 35°C is approximately 35% more efficient than all other UniAir Avg oil temperatures. As the camshaft speed increases, lubrication regime changes for the oils at higher temperatures. At high camshaft speeds the higher UniAir Avg oil temperatures become more efficient. However, the objective is to understand the influence of oil viscosity on the UniAir operation and efficiency. Therefore, investigating overall efficiency against viscosity determines this relationship. Figure 92 demonstrates the relationship between the relative hydraulic efficiency (Table 27) and viscosity (Table 26).

Plotting each data series as camshaft speed, enables the identification of how the oil viscosity influences the UniAir operation and efficiency. Figure 92 shows the relative hydraulic efficiency is dependent on oil viscosity and camshaft speed.

Equation 27 (Section 6.5.3) demonstrates power is directly proportional to the product of torque and angular speed. Figure 89 shows with increasing camshaft speed, torque increases. Consequently, higher camshaft speeds require more power and therefore more torque being applied to the camshaft to operate the UniAir. Equation 4 shows the Sommerfeld number is calculated with lubricant viscosity multiplied by speed over load. Therefore, the ratio of speed over load/torque will change for every camshaft speed. To further understand the influence of oil viscosity, each camshaft speed is investigated while observing the relative hydraulic efficiency against oil viscosity for the multiple oils presented in Section 6.1.2.

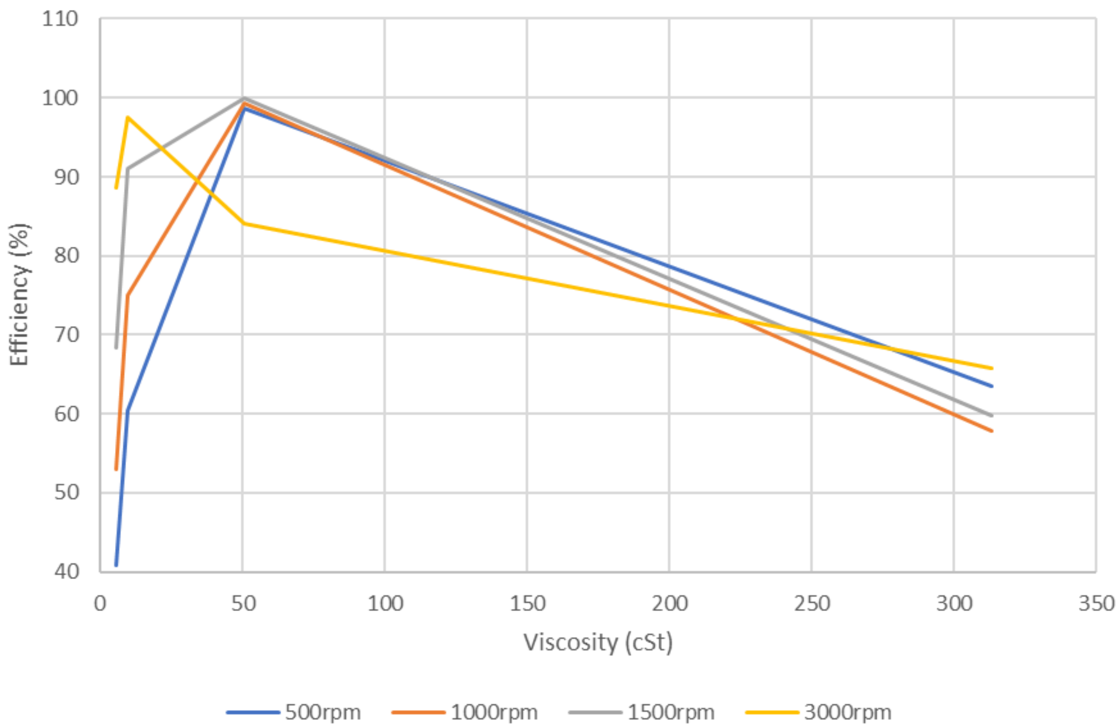


**FIGURE 92 - RELATIVE HYDRAULIC EFFICIENCY AGAINST VISCOSITY FOR 0W20 OIL AT DIFFERENT CAMSHAFT SPEEDS**

### 8.1.3 Influence of oil viscosity on UniAir efficiency in Full Lift with multiple oils

This section investigates the influence of oil viscosity of multiple oils to determine the optimal performance of the UniAir. Section 7.6 highlights the test plan for all oils studied, while Sections 8.1.1.1 and 8.1.1.2 indicate how the results of 0W20 oil are analysed. Results for 5W30 and Hyspin oil are captured and analysed in the exact same approach as 0W20 results. The relative hydraulic efficiency results for 5W30 and Hyspin oils, including torque and valve lift characteristics are found in the appendix. Therefore, the results present relative efficiency against viscosity for a specific camshaft speed for all oils studied.

#### 8.1.3.1 Influence of oil viscosity at 500rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 500rpm of three oils, 0W20, 5W30 and Hyspin. The properties of the oils are presented in 6.1.2, Figure 48. Figure 48 illustrates the differences in viscosity for a specific temperature for all three oils studied.

Figure 93 illustrates the relative hydraulic efficiency against oil viscosity at 500rpm of the three oils. It is observed that there is a high correlation between the results of all three oils. This suggests that the UniAir performance is heavily dictated by the oil viscosity and not the oil additives. All three oils have different additive packages, however the relative hydraulic efficiency at this speed shows that all three oils optimally perform at approximately 50cSt.

At 500rpm, it was observed from valve lift results in Section 8.1.1.1, at low viscosities the volumetric efficiency deteriorated dramatically. It is suggested that this is due the oil leakages, and at this camshaft speed it is more prominent due to the speed at which pressure builds up in the system. Furthermore, the torque results in Section 8.1.1.2 demonstrates at 500rpm for low viscosities, the torque is higher due to incomplete oil films as the entraining velocity is not fast enough for these viscosities. This results in a low relative hydraulic efficiency as shown in Figure 93. At the viscosities of approximately 10cSt, where the oil temperature is 90°C, it can be seen it is approximately 40% lower than the optimum range at approximately 50cSt or 35°C UniAir Avg oil temperature.

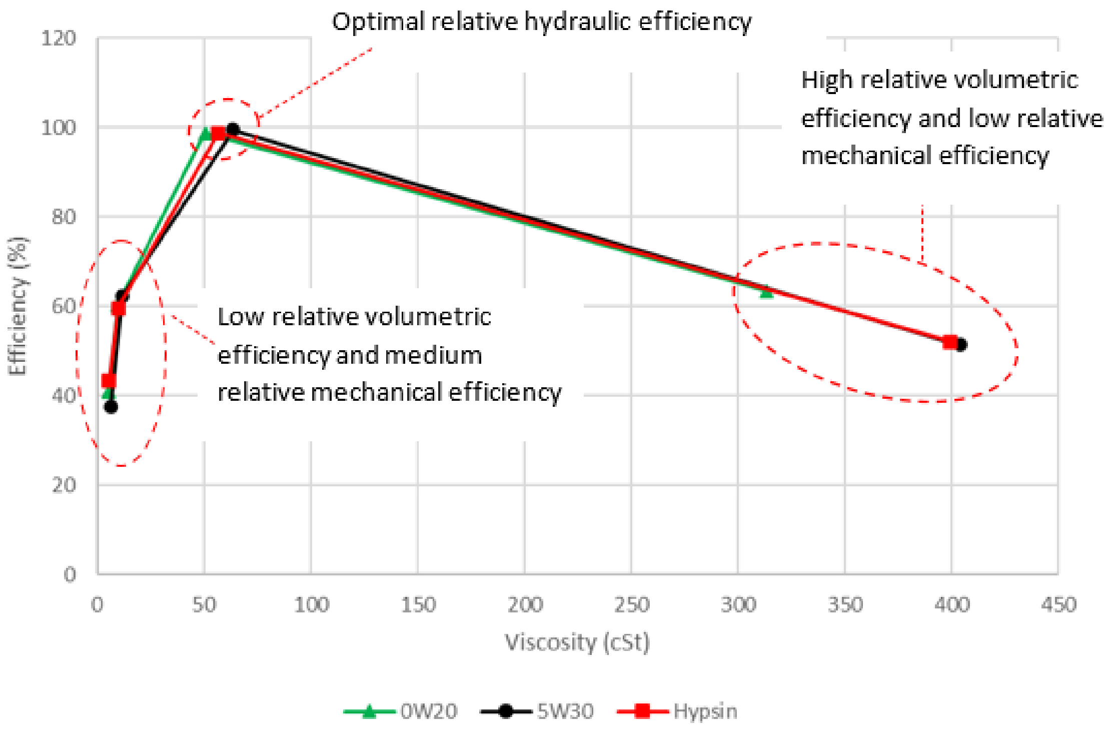


**FIGURE 93 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 500RPM**

Alternatively, the valve lift results (Section 8.1.1.1) demonstrate at high viscosities the volumetric efficiency is superior. Therefore, Figure 93 suggests that for high viscosities the relative hydraulic efficiency is dominated by low mechanical efficiencies. The torque results (Section 8.1.1.2) suggests that as viscosity increases, so does the energy required to shear the oil. This negates the high volumetric efficiency as the mechanical efficiency is relatively poor. For viscosities at 0°C UniAir Avg oil temperatures, it can be seen from Figure 93 that the relative hydraulic efficiency is also approximately 40% lower than the optimal viscosity at 50cSt or 35°C. This highlights how the relative hydraulic efficiency is optimal at approximately 50cSt. It is the compromise of lower volumetric efficiency than at higher viscosities, but the mechanical efficiency is optimum. This is theorized due to the viscosity being thick enough to generate mixed or hydrodynamic lubrication regimes for mechanical interfaces, but also not being too thick so the energy required to shear the oil is low.

The results have high correlation demonstrating the influence of oil viscosity while defining the viscosity for optimal performance and efficiency at 500rpm in Full Lift mode.

### 8.1.3.2 INFLUENCE OF OIL VISCOSITY AT 1000RPM ON UNIAIR PERFORMANCE AND EFFICIENCY

This section investigates the influence of oil viscosity at 1000rpm of three oils, 0W20, 5W30 and Hyspin. Figure 94 illustrates the relative hydraulic efficiency against oil viscosity at 1000rpm from the results calculated in Table 27.

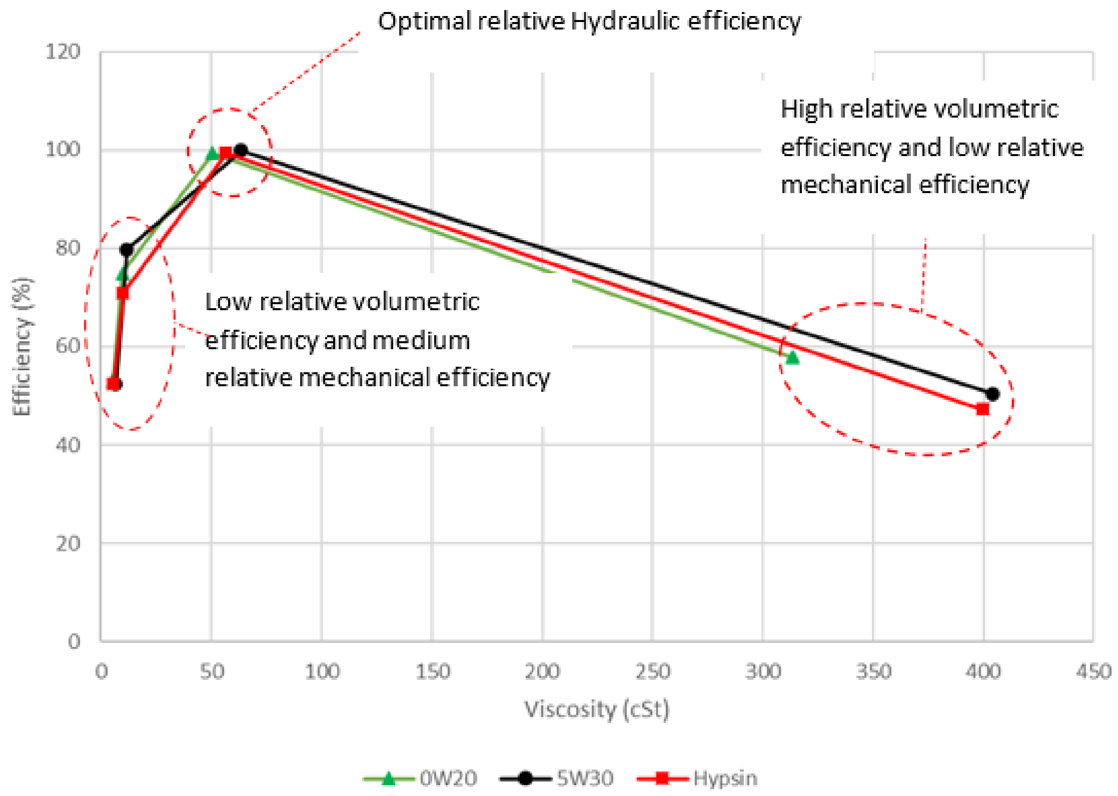


**FIGURE 94 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 1000RPM**

It is observed that the optimal efficiency for all three oils studied at 1000rpm is at approximately 50cSt. As observed at 500rpm (Section 8.1.3.1), with increasing viscosity from the optimum region, it is shown that the efficiency is approximately 40% lower. However, at lower viscosities at approximately 10cSt, the efficiency has increased from approximately 60% at 500rpm to 75% at 1000rpm. This demonstrates how the increase in camshaft speed changes the lubrication regime for mechanical interfaces, but also how the shear strength of viscosity becomes more dominant for higher camshaft speeds. Additionally, as camshaft speed increases, the volumetric efficiency increases for all viscosities as demonstrated in the valve lift results (Section 8.1.1.1) shows the leakages become less prominent.

As the results have high correlation, this provides more evidence and satisfies the objectives that at 1000rpm in Full Lift mode, the influence of oil viscosity is demonstrated, while defining the viscosity for optimal performance and efficiency for the viscosities studied.

### 8.1.3.3 Influence of oil viscosity at 1500rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 1500rpm of three oils, 0W20, 5W30 and Hyspin. Figure 95 illustrates the relative hydraulic efficiency against oil viscosity at 1500rpm. It is demonstrated that 0W20 and Hyspin oils optimal performance is at approximately 50cSt. Figure 93 and Figure 94 have demonstrated that with increasing camshaft speed, the performance of the oil viscosities at approximately 10cSt or 90°C oil temperature is also increasing. Figure 95 illustrates that for 0W20 and Hyspin oils, at lower viscosities of approximately 10cSt, the efficiency has increased from approximately 60% at 500rpm and 75% at 1000rpm, to 90% at 1500rpm. This further demonstrates how the increase in camshaft speed changes the lubrication regime for mechanical interfaces, but also how the shear strength of viscosity becomes more dominant for higher camshaft speeds.

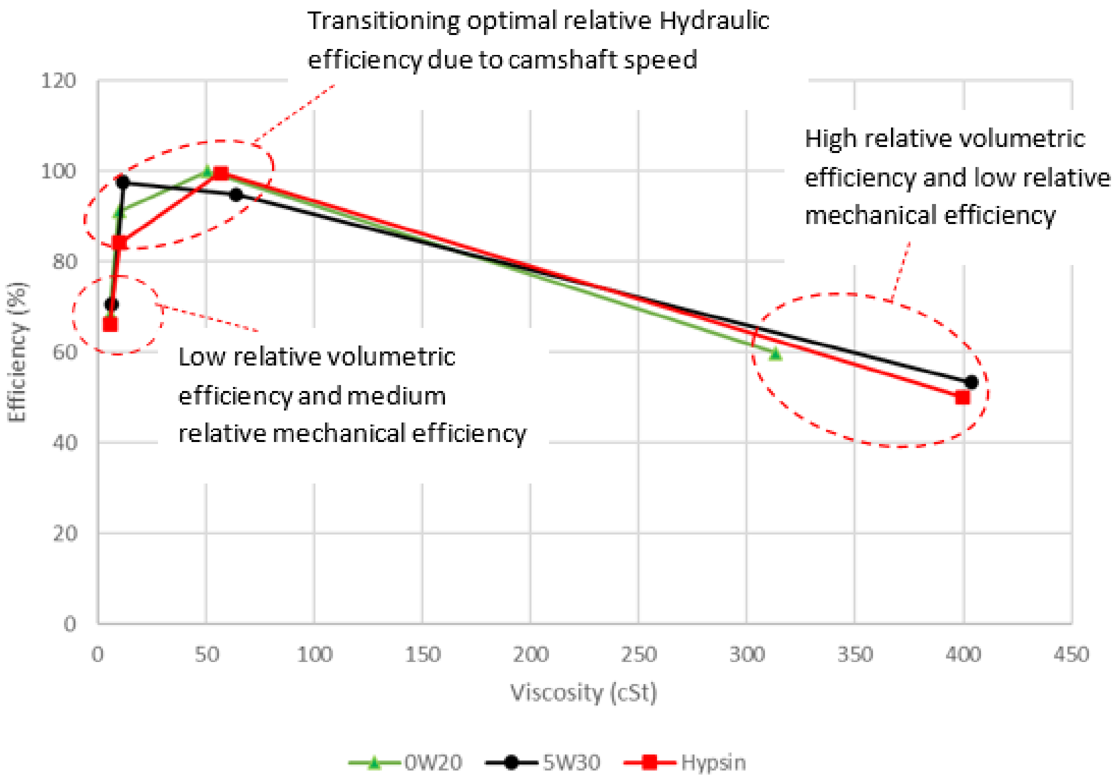


**FIGURE 95 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 1500RPM**

However, for the 5W30 oil it can be observed that the optimal performance is at approximately 10cSt. Figure 48 illustrates the viscosities of each oil for a specific temperature. This shows that the viscosity at 35°C is approximately 14cSt higher for 5W30 oils than 0W20 and Hypsin oils. To understand this relationship further, the relative mechanical, volumetric and overall efficiency of the 5W30 oil are investigated. Figure 96 illustrates the relative mechanical, volumetric and overall efficiency against viscosity for the 5W30 oil at 1500rpm.

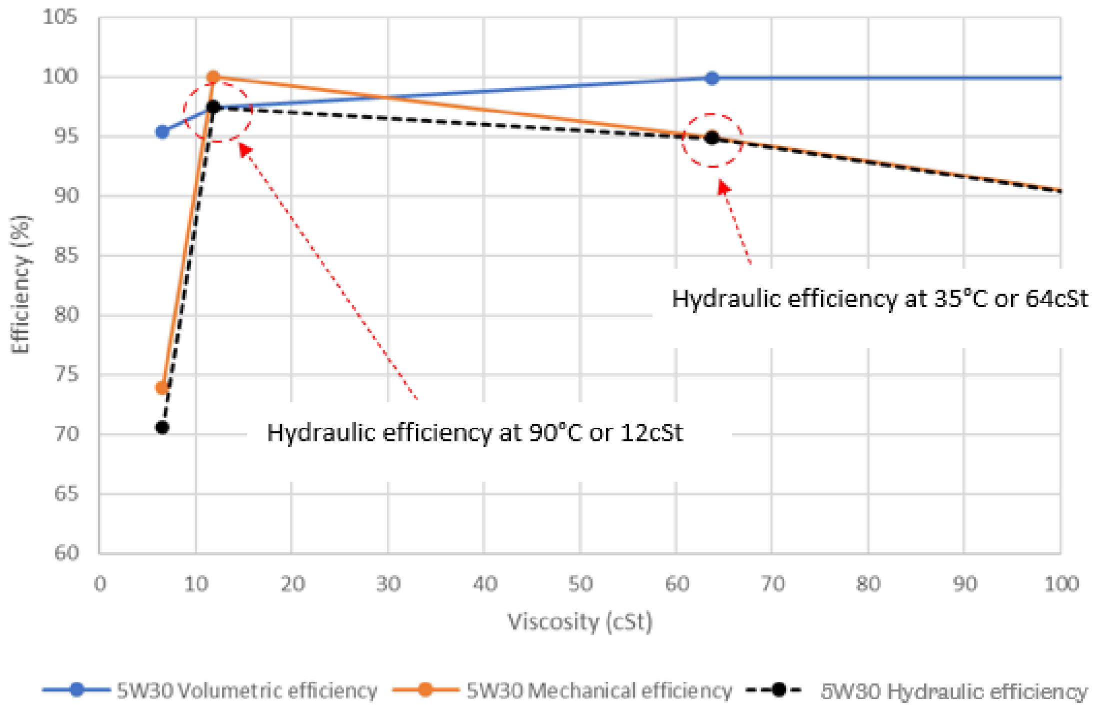


**FIGURE 96 – RELATIVE MECHANICAL, VOLUMETRIC, AND OVERALL EFFICIENCY OF 5W30 OIL AT 1500RPM**

The axis of efficiency and viscosity is reduced to highlight the relationship at the viscosities of interest, for oil temperatures at 35°C and 90°C. Figure 96 demonstrates the relationship of overall efficiency as the product of mechanical and volumetric efficiency. At viscosities at 64cSt, it shows the efficiency for volumetric efficiency is higher than mechanical efficiency. Alternatively, at viscosities at 12cSt, Figure 96 shows the mechanical efficiency is higher than the volumetric efficiency. The overall efficiency at both these points are approximately 2.5% different. This illustrates the oil viscosities influence on the transition of efficiency for increasing camshaft speed. Figure 96 further demonstrates that higher viscosities produce higher volumetric efficiency, while also showing the viscosities influence on mechanical efficiency at 1500rpm.

#### 8.1.3.4 Influence of oil viscosity at 3000rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 3000rpm of three oils, 0W20, 5W30 and Hyspin. Figure 97 illustrates the relative hydraulic efficiency against oil viscosity at 3000rpm. It is demonstrated that all oils studied have an optimal performance at approximately 10cSt.

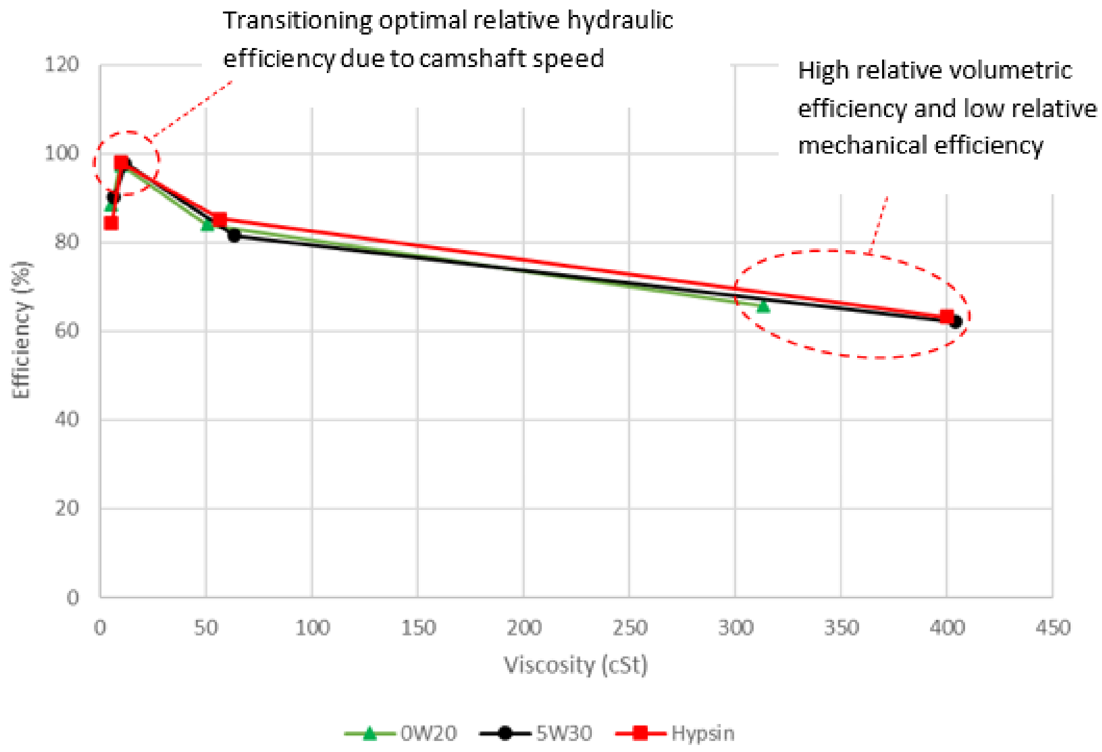


**FIGURE 97 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 3000RPM**

At 3000rpm camshaft speed, all the highest relative overall efficiencies have transitioned from 50cSt to 10cSt as observed at 1500rpm for 5W30 oil (Figure 96). At this speed, the entraining velocity is fast enough to generate an oil film thick enough to separate the surfaces with an oil viscosity of approximately 10cSt. Furthermore, this lower viscosity requires less energy to shear the volume of oil inside the UniAir, resulting in higher mechanical efficiencies at higher camshaft speeds. Additionally, at 3000rpm it is observed for the valve lift profiles in Figure 88, the influence of oil viscosity on volumetric efficiency is smaller due to the speed at which the pressure builds up in the UniAir. This results in smaller reductions in volumetric efficiency, subsequently creating higher levels of relative hydraulic efficiency at 10cSt for this camshaft speed.

### 8.1.4 Summary of Full Lift results

The Full Lift torque and valve lift results have been presented and analysed (Section 8.1.1) allowing for comparisons to previous experimental (Section 4.1) and simulated studies (Section 4.2) to validate the rigs performance. It has reiterated how the novel design and instrumentation of the rig has evaluated the operation and efficiency of the UniAir. The operation of the UniAir was evaluated by measuring the valve lift and torque while varying camshaft speeds and oil viscosities.

Evidence presented in Section 8.1.1 demonstrates how valve lift is directly related to both oil viscosity and camshaft speed. It was found that valve lift deteriorates with decreasing viscosity across all camshaft speeds investigated, which is theorized due to oil leakages bypassing clearances in the UniAir.

Consequently, at low camshaft speeds (500rpm) and low viscosities (5.5cSt) this caused a delayed valve lift profile and reduced overall height by up to 16%. Alternatively, it was found that valve lift height increases with increasing camshaft speed. At high camshaft speeds (3000rpm) and low viscosities (5.5cSt) it is shown that valve lift height is reduced by up to 5%. This is proposed to be due to the higher rate at which the oil pressure builds up inside the UniAir counteracting the rate at which the oil leaks and thus generating higher valve lift heights.

Section 8.1.1.2 shows torque is also dependent on both oil viscosity and camshaft speed. It was found that increasing camshaft speed increased torque across all viscosities studied. This behaviour is also experienced in hydraulic pumps (28), which suggested the UniAir torque is dominated by the hydraulic components inside the UniAir. It is observed that torque values at fixed camshaft speeds are different due to oil viscosities. This is due to dynamic viscosity and changing lubrication regime.

For high viscosities (313cSt) it was found more energy is required to shear the oil inside the UniAir, however this provided an oil film thickness to generate mixed or hydrodynamic lubrication between surfaces. It is observed at high viscosities (313cSt) and low camshaft speed (500rpm) the torque values are 40% higher than at optimal viscosities (50cSt) at this camshaft speed. Alternatively, at high viscosities (313cSt) and high camshaft speeds (3000rpm) it is identified torque values are 34% higher than optimal viscosities (10cSt).

For low viscosities (5cSt) it was found the required energy to shear the oil inside the UniAir is lower. However, the oil film thickness between surfaces is too thin, causing boundary lubrication at low camshaft speeds (500rpm) generating torque values 50% higher than optimal viscosities (50cSt). Alternatively, at high camshaft speeds (3000rpm) and low viscosities (5.5cSt) it was found torque values are 7% higher than optimal viscosities (10cSt) at this camshaft speed.

The results have demonstrated the approach used to identify the most suitable way to understand the influence of oil viscosity on the efficiency of the UniAir, by adopting the current method for hydraulic efficiency used on traditional pumps. It was found the influence of oil viscosity on the UniAir efficiency changed dependent on camshaft speed. At low camshaft speeds, the lowest relative hydraulic efficiency was found to be at 41% for oil viscosities at 5.5cSt while optimal relative hydraulic efficiency was found at oil viscosities of approximately 50cSt.

Alternatively, at high camshaft speeds the lowest relative hydraulic efficiency was found to be at 65% for oil viscosities at 313cSt while optimal relative hydraulic efficiency was found at oil viscosities of approximately 10cSt.

Viscosity values of 10 cSt and 50 cSt are conventional Kv100 and Kv40 values for a 0W-30 oil, typically used on diesel engines currently in the EU which do not use hydraulic valvetrains. 0W-20 oils typical of petrol engines are conventionally around Kv100 = 7.5cSt and Kv40=35-40 cSt, and lowering the viscosity to a 0W-16 grade would direct further from this optimal viscosity range, which is shown true, regardless of the oil composition. This is an important overall conclusion as a move to lower viscosities (which is the current trend across all OEMs) will not benefit hydraulic valvetrains if considered in isolation. Alternatively, this means that with a move to lower viscosity grades (which tend to benefit the overall engine efficiency), there are gains to be had by optimising the hydraulic valvetrain which are currently most efficient in a slightly higher viscosity range.

Evidence from the Test Methodology (Chapter 7) and the Full Lift results (Section 8.1.1) have demonstrated a novel way to understand the UniAir's performance, therefore creating novel results from previously studied temperatures. Furthermore, the results have identified the operational performance and efficiency for oil temperatures at 0°C which have not been previously studied.

## 8.2 No Lift results

This section analyses the torque characteristics of the UniAir in No Lift mode. To validate and satisfy the requirements of the overall project, comparisons are made with the literature, previous experimental and simulated tests (Chapters 4 and 5).

No Lift is described as the solenoid being open during the Lift phase of the camshaft (Section 2.4.1). With the solenoid open, the oil is transferred into the intermediate pressure chamber instead of building up pressure and displacing the valve. Consequently, there are no valve lift measurements and the torque characteristics do not show the spring forces as the valves remain closed. Therefore, the torque required is from the camshaft, mechanical interfaces between camshaft and UniAir, and the oil displaced inside the UniAir.

The boundary conditions outlined in Section 6.3 are implemented for the No Lift tests using 0W20 oil and are presented in Table 28.

| | |
|---|---|
| Oil temperature measurement points (UniAir Avg) | 0°C, 35°C, 90°C, 120°C ±1°C |
| Oil Viscosity (cSt) | 313.305, 50.541, 9.783, 5.541 |
| Oil Pressure | 3 bar ± 0.1bar |
| Camshaft speeds | 500, 1000, 1500, 3000 rpm ±10rpm |
| Oil Type | Castrol SAE 0W-20 A12013PBABOT936 |

**TABLE 28 - BOUNDARY CONDITIONS FOR TESTS USING 0W20 OIL IN NO LIFT MODE**

The results for No Lift tests using 0W20 oil are found in Table 28. This identifies a torque value specific to the UniAir Avg oil temperature and camshaft speed. The baseline torque (Section 7.5) have been illustrated in Table 28 (yellow boxes), which are used in later calculations.

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 313.305 | 50.541 | 9.783 | 5.541 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 0.902 | 0.260 | 0.166 | 0.369 |
| | 1000 | 1.385 | 0.701 | 0.623 | 0.806 |
| | 1500 | 1.870 | 1.086 | 0.929 | 1.26 |
| | 3000 | 3.6 | 2.667 | 2.266 | 2.487 |

**TABLE 29 - TEST RESULTS OF 0W20 OIL, SHOWING TORQUE FOR VARYING VISCOSITY AND CAMSHAFT SPEEDS**

### 8.2.1 VALIDATING NO LIFT TORQUE

This section analyses the torque characteristics of the UniAir while using No Lift mode. Figure 98 illustrates the data from Table 28, showing the torque characteristics of the UniAir, in No lift mode, while varying UniAir Avg oil temperatures and camshaft speeds.

The Stribeck curve (Figure 24) illustrates the Sommerfeld number is calculated with lubricant viscosity, speed and load/pressure. As stated in section 8.2, the torque characteristics do not show the spring forces as the valves remain closed. Therefore, the load/pressure is reduced, meaning the transition to different lubrication regimes is more dependent on oil viscosity and camshaft speed.

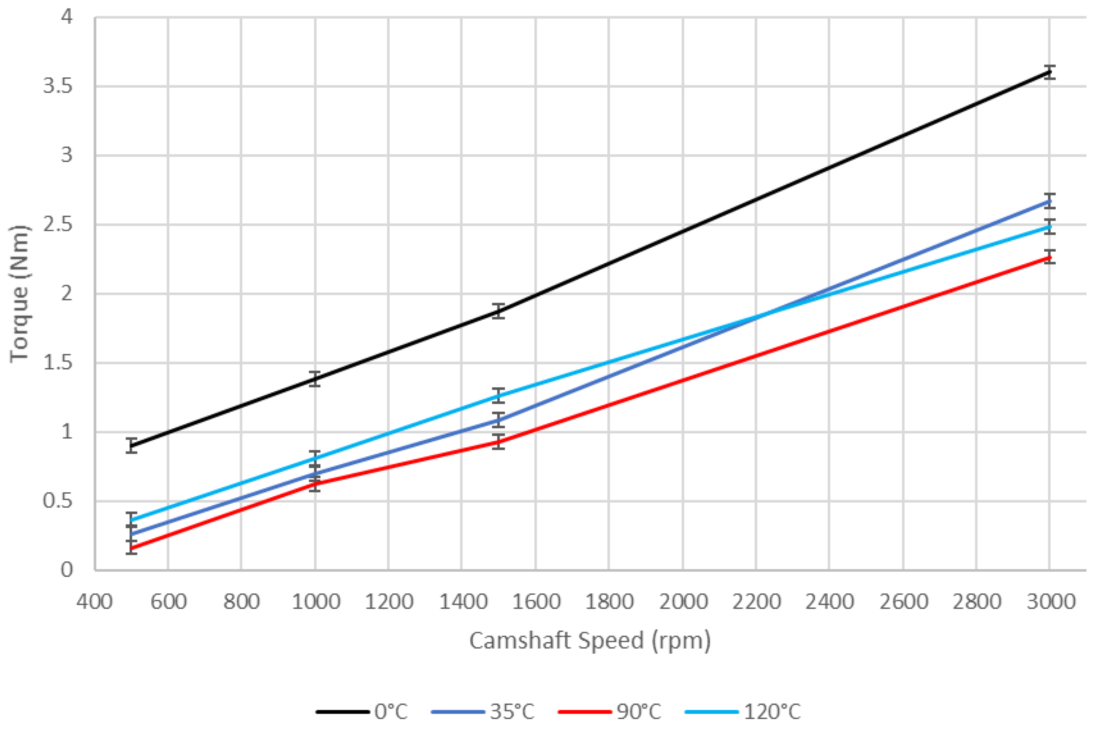


**FIGURE 98 - TORQUE CHARACTERISTICS OF NO LIFT MODE USING 0W20 OIL WHILE VARYING OIL TEMPERATURE**

At UniAir Avg oil temperatures of 0°C, the lubrication regime between all mechanical interfaces is most likely hydrodynamic due to high viscosity. However, this high viscosity oil is being continuously circulated inside the UniAir. Therefore, additional energy is required to shear the volume of oil inside the UniAir, resulting in the highest torque values across all camshaft speeds investigated. The torque characteristics are therefore assumed to be mainly comprised of two components, mechanical interfaces (Sommerfeld number) and shearing the volume of oil inside the UniAir (dynamic viscosity).

Alternatively, at 120°C and low camshaft speeds, Figure 98 illustrates the torque values are second highest. Likewise, with results found for Full Lift (Section 8.1.1.2), it is theorized that the torque values at 120°C and low camshaft speeds are because of incomplete fluid film formation as the oil viscosity is not strong enough to fully separate the mechanical interfaces. Evidence for this is also shown in Figure 98, for increasing camshaft speeds at 3000rpm, results show for the torque profile at 120°C the lubrication regime has changed from boundary lubrication to mixed lubrication, resulting in the torque profile at 35°C to become the second highest torque value due to the higher viscosity oil requiring more energy to shear the oil.

As observed in Full Lift (Section 8.1.1.2), the torque characteristics of an internal gear pump (Figure 44, Section 5.5.1), shows a similar relationship to the UniAir torque characteristics for No Lift in Figure 98. The internal gear pump, for low pressure applications, torque increases with increasing motor speed for all temperatures studied. Furthermore, at low motor speeds, the higher internal gear pump friction is dominated by high temperature oils from the temperatures studied. This relationship is also observed in Figure 98 for the No Lift torque results between equivalent temperatures at 35°C, 90°C, 120°C. Additionally, the torque characteristics of an internal gear pump (Figure 44) also demonstrates when increasing the motor speed, the lower temperature torque profiles exceeds the higher temperature torque profile. This provides evidence that the UniAir No Lift torque characteristics are dominated by the hydraulic components inside the UniAir.

Comparing the results to previous experimental work done by Schaeffler (Section 4.1.1.1), Figure 31 shows the overall relationship for No Lift torque against camshaft speed. To gain a better understanding of this comparison, the results from Figure 31 for No Lift torque is plot on the same graph as the torque results from Table 28.

Figure 99 illustrates the No Lift torque characteristics against camshaft speed for the results found in Table 29 and Schaeffler results from Figure 31 for No Lift torque. Figure 99 shows Schaeffler results also identified torque increases with camshaft speed across all temperatures studied, therefore suggesting the non-production UniAir No Lift torque characteristics are governed by the hydraulic components.

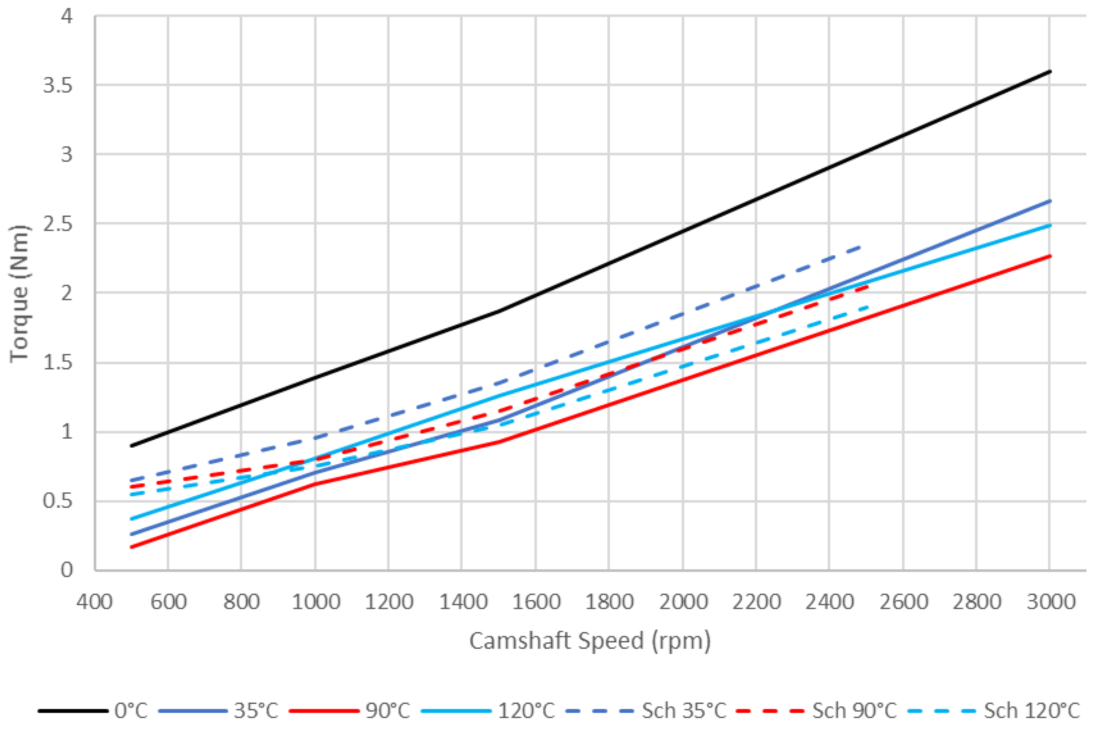


**FIGURE 99 - TORQUE CHARACTERISTICS OF NO LIFT AGAINST CAMSHAFT SPEED.**

However, unlike the internal gear pump torque characteristics, Schaeffler highlight when increasing the motor speed, the lower temperature torque profiles remain higher across all camshaft speeds investigated. This disagrees with the literature for internal gear pumps and the results found in this thesis. However, when only observing the torque profiles from this thesis in Figure 99 at 0°C, 35°C and 90°C, the lower temperature torque profiles remain higher across all camshaft speeds investigated. This directly agrees with the Schaeffler findings and further suggests the differences in oil temperatures Schaeffler are using to represent the oil temperatures inside the UniAir. Evidence shown in Chapter 7 suggests the actual temperatures being experience inside the UniAir for Schaeffler (especially 120°C), are lower.

However, at low camshaft speeds, Figure 99 illustrates that temperatures at 35°C, 90°C, 120°C for No Lift results are approximately all 0.5Nm lower than Schaeffler results. This relationship is experienced in Full Lift results (Section 8.1.1.2), which suggested the difference is due to the oil pressure supplied to the UniAir. The results in Table 28 are related to an oil pressure of 3 bar across all camshaft speeds studied. Schaeffler varied the oil pressure dependant on camshaft speed (Section 4.1, Table 3) to replicate the working conditions of the engine. However, the torque characteristics of an internal gear pump (Figure 44, section 5.5.1) show that increasing oil pressure increases torque. Therefore, the differences in values between Schaeffler and this thesis at low camshaft speeds in Figure 99 cannot be due to oil pressure differences. This would suggest the differences are due to the differences in design/manufacturing/materials for non-production and production UniAir.

Alternatively, differences in results could also be suggested due to the different testing methodologies implemented, as strong evidence shown in Chapter 7, highlights potential issues (for this study) with Schaeffler's oil temperature measurement system and therefore torque measurement. Due to the findings in Chapter 7, highlighting the potential problems with the temperature measurement position in Schaeffler's tests, Figure 99 appears to provide more evidence of this inappropriate position. The optimisation tests conducted in Section 7.3.1 suggested that Schaeffler's actual oil temperature in the UniAir could be unpredictably different. It is very probable that the actual temperatures inside the UniAir at 120°C would be significantly lower.  Due to "cooling water" tempering fluids, heat loss between measurement point and UniAir, and especially in No Lift mode due to less internal heating.

Figure 99 shows Schaeffler's results at 120°C demonstrates a relationship that is not observed in this thesis results but also the literature. This suggests that the oil temperatures Schaeffler are using to represent the oil temperature inside the UniAir are incorrect. Therefore, Figure 99 highlights the novel relationship of the No Lift UniAir torque characteristics against camshaft speeds, with changing oil temperatures. This reiterates the validity of the rigs performance and results, and subsequently shows that the torque results at 0°C and 120°C are novel.

### 8.2.2 INFLUENCE OF OIL VISCOSITY ON UNIAIR PERFORMANCE AND EFFICIENCY IN NO LIFT

The test procedure for hydraulic efficiency in Section 7.5 highlighted the most suitable way to understand the influence of oil viscosity in Full Lift mode of the UniAir, by adopting the current method of hydraulic efficiency used on traditional pumps. However, as No Lift mode creates no valve lift, therefore has no volumetric efficiency and overall hydraulic efficiency cannot be determined. However, the torque characteristics in Figure 98 also match the torque characteristics of an internal gear pump (Figure 44, section 5.5.1). This suggests the hydraulic components of the UniAir are dominating the overall torque value regardless of no valve lift. Each revolution of the camshaft still engages with the pumping elements of the UniAir, and although little pressure is generated the body of oil inside the UniAir is still being transferred throughout the UniAir. Subsequently, the relative mechanical efficiency in Table 30 is calculated using the baseline torque values in Table 29 (outlined in Section 7.5) and equations 28 and 30.

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency (%) |
|---|---|---|
| 500 | 313.3 | 18.4 |
| | 50.5 | 63.7 |
| | 9.8 | 100 |
| | 5.5 | 44.99 |
| 1000 | 313.3 | 45 |
| | 50.5 | 88.9 |
| | 9.8 | 100 |
| | 5.5 | 77.33 |
| 1500 | 313.3 | 49.7 |
| | 50.5 | 85.5 |
| | 9.8 | 100 |
| | 5.5 | 73.7 |
| 3000 | 313.3 | 63 |
| | 50.5 | 85 |
| | 9.8 | 100 |
| | 5.5 | 91.1 |

**TABLE 30 - MECHANICAL EFFICIENCY OF NO LIFT RESULTS USING 0W20 OIL**

Calculating the relative mechanical efficiency is used to understand the influence of oil viscosity on the UniAir operation and efficiency. To do this, the relationship between relative mechanical efficiency against viscosity is illustrated in Figure 100.

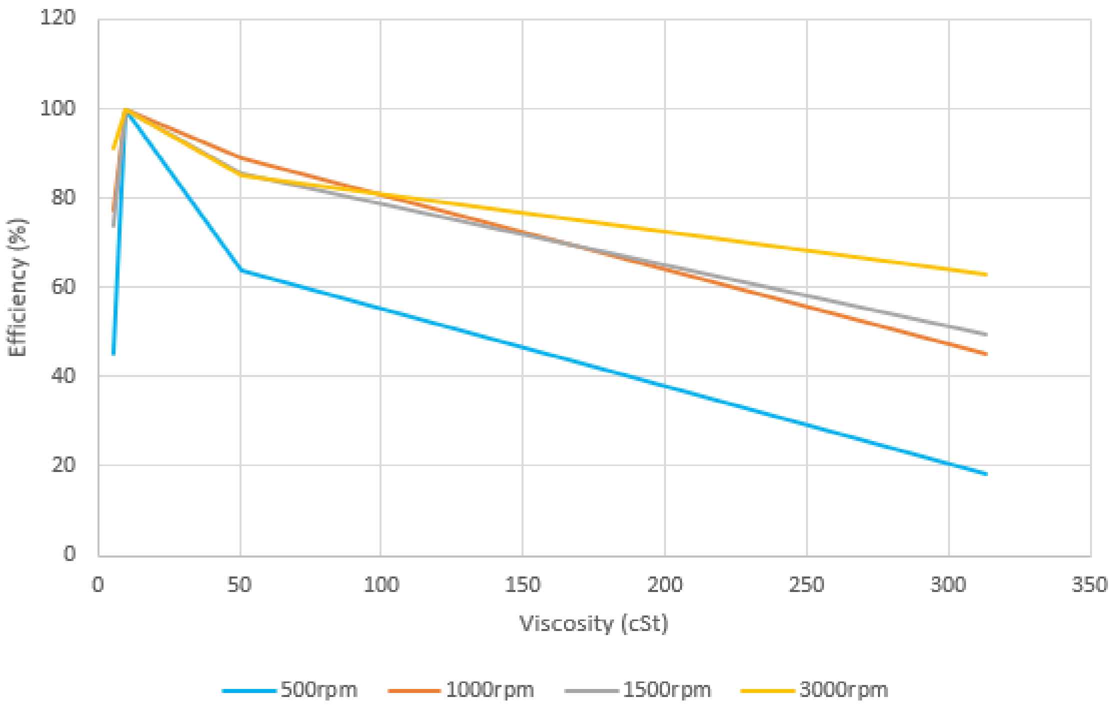


**FIGURE 100 - RELATIVE MECHANICAL EFFICIENCY AGAINST VISCOSITY FOR 0W20 OIL IN NO LIFT MODE**

Figure 100 demonstrates that the efficiency is dictated by both oil viscosity and camshaft speed. Evidence presented in Full Lift results (Section 8.1.2) demonstrates power is directly proportional to the product of torque and angular speed. Therefore, to further understand the influence of oil viscosity, each camshaft speed is investigated while observing the relative mechanical efficiency against oil viscosity of the multiple oils outlined in Section 7.6.

### 8.2.3 Influence of oil viscosity on UniAir efficiency in No Lift with multiple oils

This section investigates the influence of oil viscosity of multiple oils to determine the optimal performance of the UniAir in No Lift mode. Section 7.6 highlights the test plan for all oils studied, while section 8.2.2 indicates how the results of 0W20 oil are analysed. Results for 5W30 and Hyspin oil are captured and analysed in the exact same approach as 0W20 results. The relative mechanical efficiencies for 5W30 and Hyspin oils, including torque characteristics are found in the appendix. Therefore, the results present relative mechanical efficiency against viscosity for a specific camshaft speed for all oils studied.

#### 8.2.3.1 Influence of oil viscosity at 500rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 500rpm of three oils, 0W20, 5W30 and Hyspin. The properties of the oils are presented in 6.1.2, Figure 48. Figure 48 illustrates the differences in viscosity for a specific temperature for all three oils studied.

Figure 101 illustrates the relative mechanical efficiency against oil viscosity at 500rpm of the three oils. It is observed that there is a near perfect correlation between the results of 0W20 and 5W30 oils. However, the Hyspin oil results do not correlate perfectly for all viscosities studied. All three oils have different additive packages, however the relative mechanical efficiency at this speed shows that all three oils optimally perform at approximately 10cSt. However, both 0W20 and 5W30 oils are both production level engine oils and are assumed to have similar additive packages. This suggests that the UniAir performance is still heavily dictated by the oil viscosity, however the oil additives also have an influence on the mechanical efficiency.

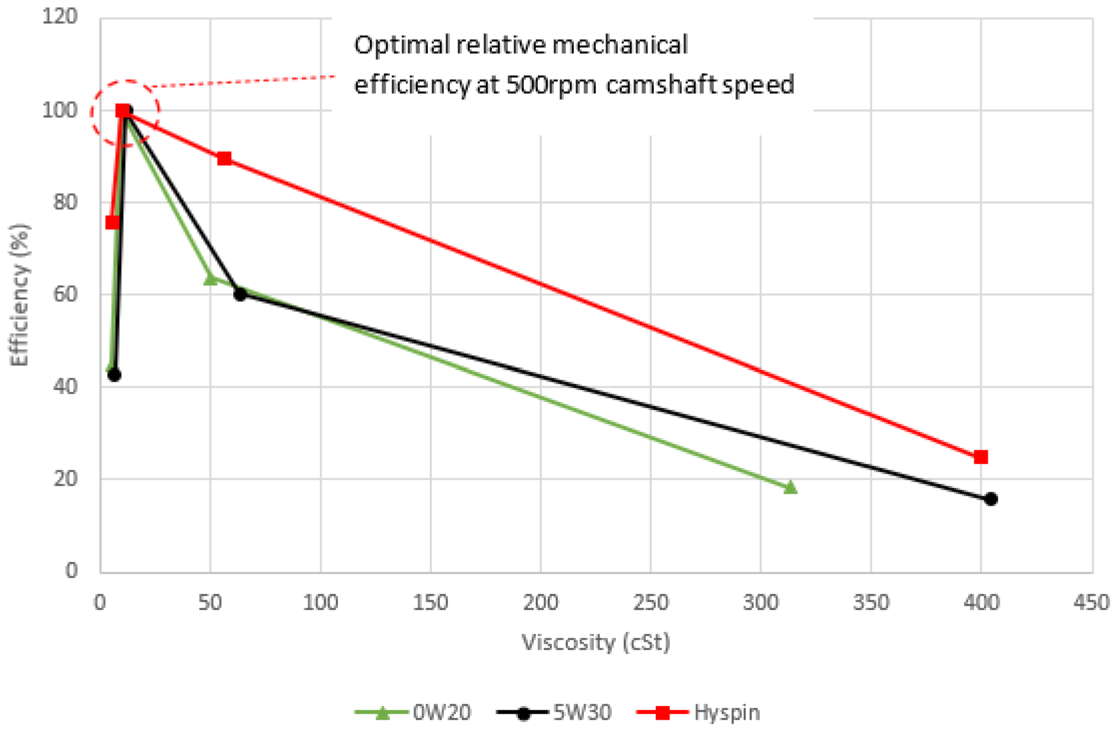


**FIGURE 101 – RELATIVE MECHANICAL EFFICIENCY AGAINST VISCOSITY AT 500RPM, HIGHLIGHTING INFLUENCE OF OIL VISCOSITY ON UNIAIR PERFORMANCE**

The influence of oil viscosity at 500rpm for Full Lift results (Section 8.1.3.1) identified that for all the oils studied, the UniAir performed optimally at approximately 50cSt. However, in comparison to Figure 101, this shows the optimal efficiency has transitioned to approximately 10cSt. Although direct comparisons cannot be made as No Lift mode has no volumetric efficiency, this difference is most likely due to the change in load.

No Lift torque characteristics do not show the spring forces as the valves remain closed. This reduction in load/torque enables the entraining velocity at this speed to generate an oil film thick enough at 10cSt to separate the mechanical interfaces. Evidence of this is shown in Figure 101, as with increasing viscosity from 10cSt the relative mechanical efficiency decreases across all oils. Theorizing the lubrication regime is mixed or hydrodynamic at 10cSt, increasing the viscosity generates thicker oil films and higher required torque to shear the oil.

Furthermore, when decreasing viscosity from the optimal efficiency at 10cSt, Figure 101 illustrates the relative mechanical efficiency also decreases. This suggests the reduced loads in No Lift mode, the viscosity at approximately 5cSt is not strong enough for the entraining velocities at this camshaft speed, to separate the mechanical interfaces.

The results have high correlation demonstrating the influence of oil viscosity while defining the viscosity for optimal performance and efficiency at 500rpm in No Lift mode.

### 8.2.3.2 Influence of oil viscosity at 1000rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 1000rpm of three oils, 0W20, 5W30 and Hyspin in No Lift mode. Figure 102 highlights the relationship between relative mechanical efficiency against oil viscosity at 1000rpm. Similarly to Figure 101 (500rpm relative mechanical efficiency against viscosity), Figure 102 illustrates there is a high correlation between the results of 0W20 and 5W30 oils. However, the Hyspin oil results do not correlate for all viscosities studied. Nevertheless, the relative mechanical efficiency at this speed shows that all three oils optimally perform at approximately 10cSt.

Figure 102 shows at lower viscosities at approximately 5cSt, the relative mechanical efficiency of 0W20 and 5W30 is 75%. Equation 6 highlights how the dynamic viscosity calculated, with a fixed camshaft speed and distance, the shear strength of the oil is dictated by the dynamic viscosity. Therefore, for viscosities that are lower than the viscosity at the optimal efficiency, is due to the lubrication regime. This suggests that the lubrication regime at viscosities of 5cSt are in the boundary or mixed region, subsequently generating lower relative mechanical efficiencies.

Alternatively, higher viscosities at 50cSt, shows the relative mechanical efficiency is 85%. This reduction in efficiency with higher viscosity is a combination of the lubrication regime and shear strength of the oil. As the relative mechanical efficiency is lower with a higher viscosity at 50cSt, this suggests the lubrication regime is hydrodynamic, creating a thicker oil film and therefore more oil to shear. Furthermore, with a higher viscosity requires more energy to shear the volume of oil inside the UniAir, subsequently reducing the relative mechanical efficiency.

The results have high correlation demonstrating the influence of oil viscosity while defining the viscosity for optimal performance and efficiency at 1000rpm in No Lift mode.

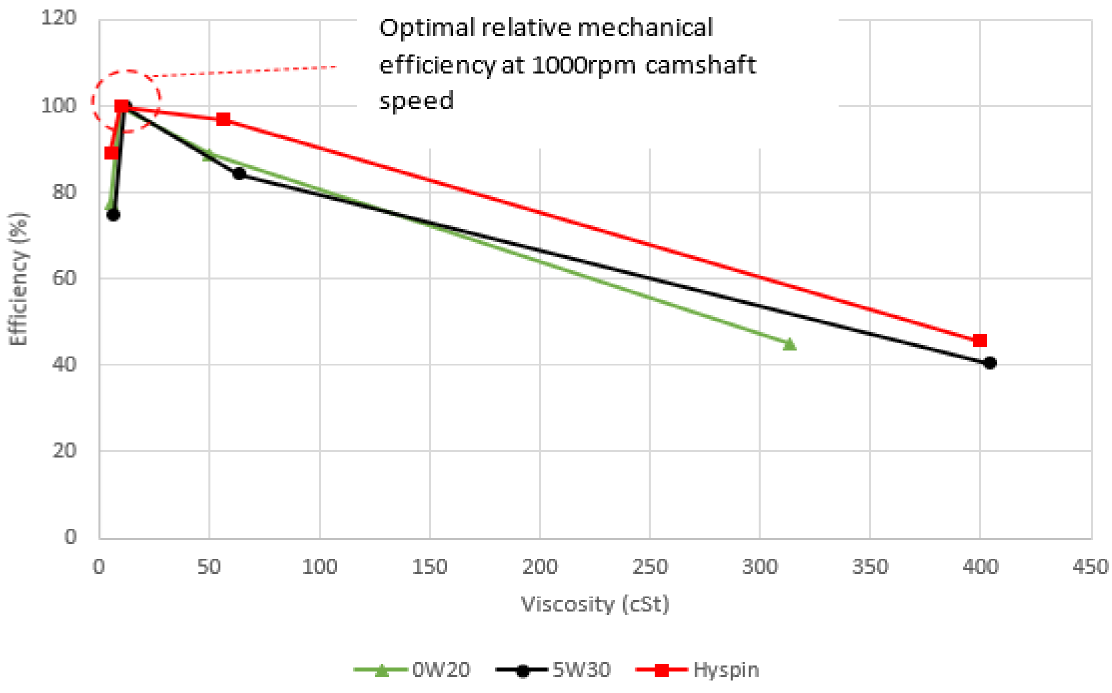


**FIGURE 102 - RELATIVE MECHANICAL EFFICIENCY AGAINST VISCOSITY AT 1000RPM, HIGHLIGHTING INFLUENCE OF OIL VISCOSITY ON UNIAIR PERFORMANCE**

### 8.2.3.3 Influence of oil viscosity at 1500rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 1500rpm of three oils, 0W20, 5W30 and Hyspin in No Lift mode. Figure 103 illustrates the relationship of relative mechanical efficiency against viscosity at 1500rpm for all three oils studied. Figure 103 identifies the optimal performance of the UniAir in No Lift mode at 1500rpm is at approximately 10cSt for all oils studied. The relationship observed in Figure 103 almost has a perfect correlation with the results at 1000rpm in Figure 102. This suggests the Sommerfeld number at 1500rpm is similar to 1000rpm as the efficiencies are so closely correlated across all viscosities and oils studied.

The results have high correlation demonstrating the influence of oil viscosity while defining the viscosity for optimal performance and efficiency at 1500rpm in No Lift mode.

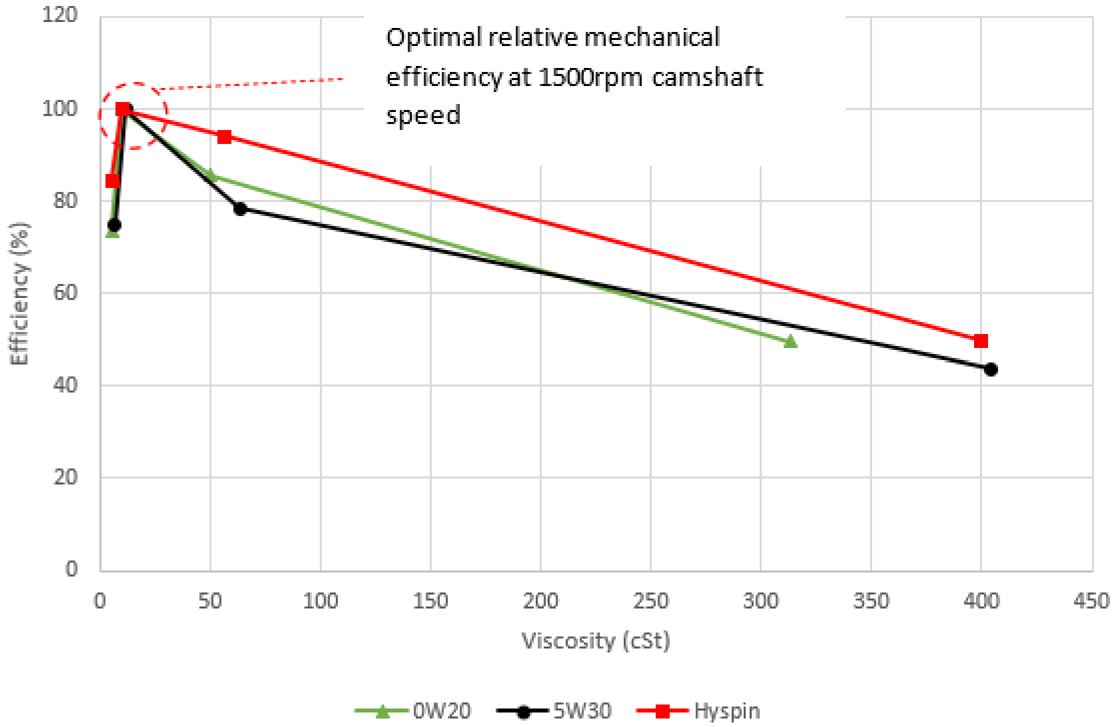


**FIGURE 103 - RELATIVE MECHANICAL EFFICIENCY AGAINST VISCOSITY AT 1500RPM, HIGHLIGHTING INFLUENCE OF OIL VISCOSITY ON UNIAIR PERFORMANCE**

### 8.2.3.4 Influence of oil viscosity at 3000rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 3000rpm of three oils, 0W20, 5W30 and Hyspin in No Lift mode. Figure 104 demonstrates the relationship between relative mechanical efficiency against viscosity at 3000rpm. It is observed the optimal viscosity in No Lift mode for all three oils is at approximately 10cSt. As experienced with lower camshaft speeds, viscosities at approximately 5cSt are assumed to have lower relative mechanical efficiency due to incomplete fluid film thickness between mechanical interfaces. Alternatively, viscosities at approximately 50cSt and higher have lower relative mechanical efficiencies due to increased shear strength of the oil.

However, at 3000rpm the Hyspin results are much more highly correlated to the engine oils. This suggests the oil additives are much less influential at higher camshaft speeds. The results are highly correlation between oils and provides further evidence and satisfies the objectives for understanding the influence of oil viscosity, while defining the viscosity for optimal performance.

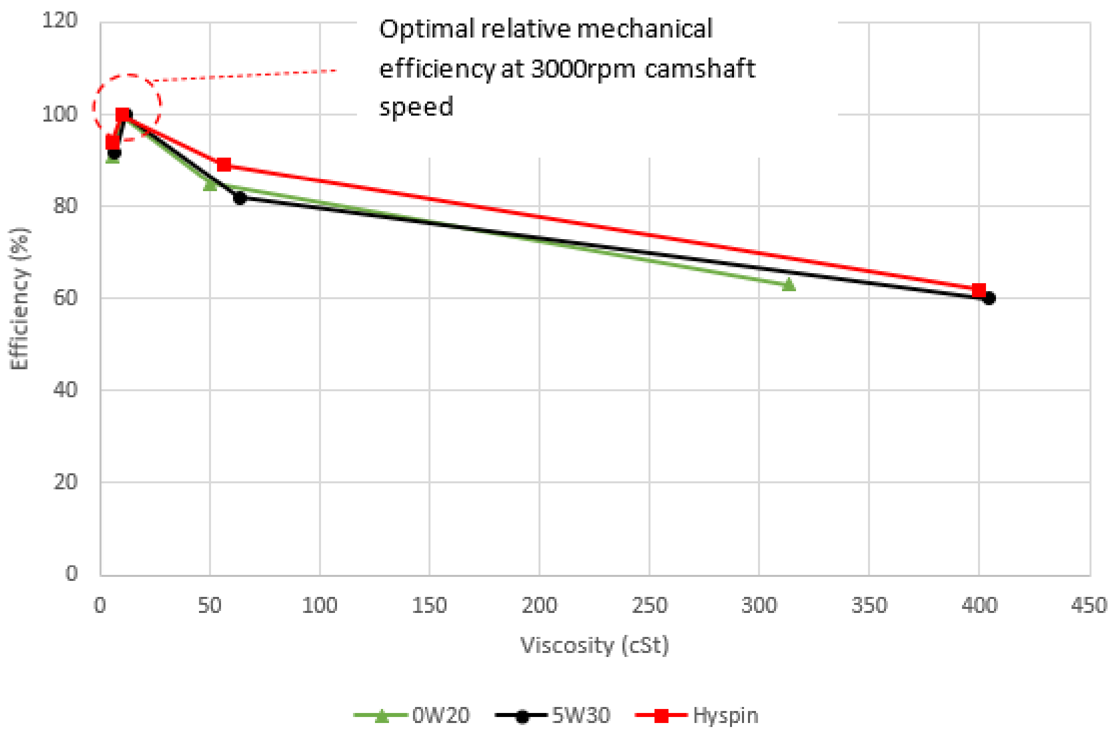


**FIGURE 104 - RELATIVE MECHANICAL EFFICIENCY AGAINST VISCOSITY AT 3000RPM, HIGHLIGHTING INFLUENCE OF OIL VISCOSITY ON UNIAIR PERFORMANCE**

### 8.2.4 SUMMARY OF NO LIFT RESULTS

The No Lift torque results have been presented and analysed (Section 8.2.1) allowing for comparisons to previous experimental studies (Section 4.1) to validate the rigs performance. It has reiterated how the novel design and instrumentation of the rig has evaluated the operation and efficiency of the UniAir. The operation and efficiency of the UniAir in No Lift mode was evaluated by measuring the torque while varying camshaft speed and oil viscosities.

As found in Full Lift results (Section 8.1), No Lift results (Section 8.2.1) highlight the torque is dependent on both oil viscosity and camshaft speed. It was found that increasing camshaft speed increased torque across all viscosities studied. This behaviour is also experienced in hydraulic pumps (28), which suggested the No Lift UniAir torque is also dominated by the hydraulic components inside the UniAir. It is observed that torque values at fixed camshaft speeds are different due to oil viscosities. This is due to dynamic viscosity and changing lubrication regime.

For high viscosities (313cSt) it was found more energy is required to shear the oil inside the UniAir, however this provided an oil film thickness to generate mixed or hydrodynamic lubrication between surfaces. It is observed at high viscosities (313cSt) and low camshaft speed (500rpm) the torque values are 82% higher than at optimal viscosities (10cSt) at this camshaft speed. Alternatively, at high viscosities (313cSt) and high camshaft speeds (3000rpm) it is identified torque values are 37% higher than optimal viscosities (10cSt).

Likewise with Full Lift torque results (Section 8.1.1.2), No Lift torque results found low viscosities (5cSt) the required energy to shear the oil inside the UniAir is lower. However, the oil film thickness between surfaces is too thin, causing boundary lubrication at low camshaft speeds (500rpm) generating torque values 55% higher than optimal viscosities (10cSt). Alternatively, at high camshaft speeds (3000rpm) and low viscosities (5.5cSt) it was found torque values are 8% higher than optimal viscosities (10cSt) at this camshaft speed.

Section 8.2.2 details the approach used to identify the most suitable way to understand the influence of oil viscosity on the efficiency of the UniAir, by adopting the current method used on traditional pumps. Section 8.2.3 demonstrates the adopted hydraulic pump efficiency of the UniAir. This provides evidence for the UniAir performance and efficiency to be analysed as a traditional pump. It was found the influence of oil viscosity on the UniAir mechanical efficiency changed dependent on camshaft speed.

At low camshaft speeds (500rpm), the lowest mechanical efficiency was found to be at 18% for oil viscosities at 313cSt while optimal mechanical efficiency was found at oil viscosities of approximately 10cSt. Alternatively, at high camshaft speeds the lowest mechanical efficiency was found to be at 63% for oil viscosities at 313cSt while optimal relative hydraulic efficiency was found at oil viscosities of approximately 10cSt.

As found in Full Lift results (Section 8.1), lowering the viscosity to a 0W-16 grade would add no benefit to the optimal viscosity range, which is shown true, regardless of the oil composition. This is an important overall conclusion as a move to lower viscosities (which is the current trend across all OEMs) will not benefit hydraulic valvetrains if considered in isolation. Alternatively, this means that with a move to lower viscosity grades (which tend to benefit the overall engine efficiency), there are gains to be had by optimising the hydraulic valvetrain which are currently most efficient in a slightly higher viscosity range.

Evidence from the Test Methodology (Chapter 7) and the No Lift results (Section 8.2.1) have demonstrated novel way in determining the UniAir's performance, therefore creating novel results from previously studied temperatures. Furthermore, the results have identified the operational performance and efficiency for oil temperatures at 0°C which have not been previously studied.

## 8.3 EIVC 540° LIFT RESULTS

This section investigates the torque and valve lift characteristics of the UniAir in EIVC mode to validate the valvetrain test rig, the results are compared to the literature and previous experimental and simulated results.

The mechanisms of EIVC are described in Section 2.4.1.1.3. It is explained to generate EIVC lift, the ECU produces a current to open the solenoid at a specific crank angle. Section 14.4 identifies the variation and repeatability of the UniAir when using EIVC valve lift for a specific crank angle at 540°. Furthermore, the simulated results (Section 4.2) show a crank angle of 520° used. The test plan (Section 7.6) emphasised the large number of tests conducted in this thesis, therefore only one crank angle for the EIVC tests can be used. Consequently, this dictated the crank angle of 540° to use for validation as it replicates the variation and repeatability in Section 14.4 and is close to the 520° crank angle used in the simulated results (Section 4.2) for comparisons of the valve lift profiles.

To validate the valvetrain test rig, results are compared to previous experimental (Section 4.1.1.2) and simulated results (Section 4.2.1). To do this, the results from 0W20 are analysed first as previous experimental and simulated studies used 0W20 in their tests.

The boundary conditions outlined in Section 6.3 are implemented for the EIVC Lift tests using 0W20 oil and are presented in Table 31.

| | |
|---|---|
| Oil temperature measurement points (UniAir Avg) | 0°C, 35°C, 90°C, 120°C ±1°C |
| Oil Viscosity (cSt) | 313.3, 50.5, 9.8, 5.5 |
| Oil Pressure | 3 bar ± 0.1bar |
| Camshaft speeds | 500, 1000, 1500, 3000 rpm ±10rpm |
| Oil Type | Castrol SAE 0W-20 A12013PBABOT936 |
| Closing Crank Angle (CA°) | 540° |

**TABLE 31 - BOUNDARY CONDITIONS OF EIVC TESTS USING 0W20 OIL**

The results for EIVC Lift tests using 0W20 oil are found in Table 32. This identifies a torque and valve lift value specific to the UniAir Avg oil temperature and camshaft speed. The baseline torque and valve lift (Section 7.5) have been illustrated in Table 32 (yellow boxes), which are used in later calculations.

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 313.3 | 50.5 | 9.8 | 5.5 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Baseline torque value | 500 | 4.53 | 3.73 | 3.52 | 3.81 |
| | 1000 | 4.72 | 4.08 | 3.97 | 4.17 |
| | 1500 | 4.30 | 3.81 | 3.60 | 3.85 |
| | 3000 | 4.21 | 3.19 | 2.85 | 3.05 |
| Valve Lift (mm) | | | | | |
| Baseline valve lift value | 500 | 10.38 | 10.35 | 9.76 | 9.28 |
| | 1000 | 9.30 | 9.83 | 9.66 | 9.49 |
| | 1500 | 6.78 | 7.85 | 8.15 | 8.18 |
| | 3000 | 2.22 | 2.18 | 2.50 | 2.46 |

**TABLE 32 – EIVC TEST RESULTS OF 0W20 OIL, SHOWING TORQUE AND VALVE LIFT FOR VARYING VISCOSITY AND CAMSHAFT SPEEDS**

### 8.3.1.1 VALIDATING EIVC VALVE LIFT

This section analyses the valve lift characteristics of the UniAir while using EIVC lift mode. Figure 105 illustrates the data from Table 32 showing the relationship between valve lift against camshaft speed while varying oil temperature.

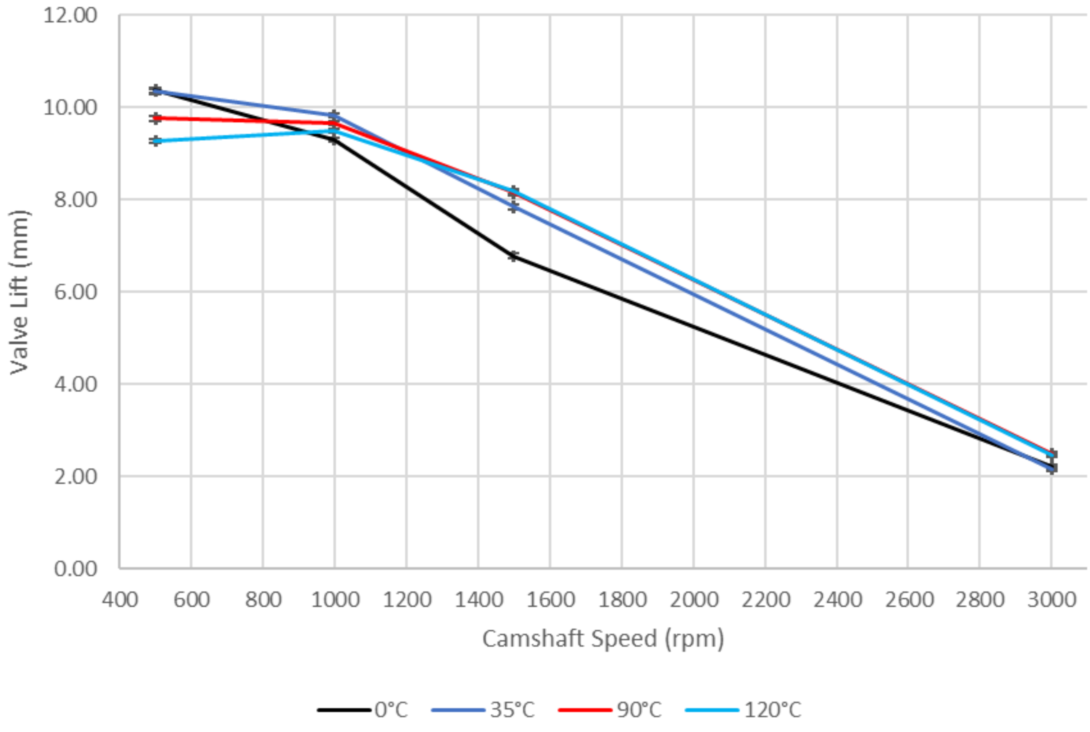


**FIGURE 105 - VALVE LIT AGAINST CAMSHAFT SPEED FOR EIVC LIFT MODE, HIGHLIGHTING VALVE LIFT CHARACTERISTICS WITH DIFFERENT OIL TEMPERATURES**

Figure 105 demonstrates at low camshaft speeds of 500rpm, valve lift height decreases with increasing oil temperature. This relationship was also identified in Full Lift results (Section 8.1.1) and explained that loss of the volumetric efficiency is produced by high temperature low viscosity lubricants bypassing pump clearances causing leakages. Koehler and Jackson (35) agree with this finding as they also found higher viscosity characteristics greatly improved volumetric efficiency (35). However, for camshaft speeds at 500rpm, oil temperatures at 0°C and 35°C where oil viscosities are higher, Figure 105 demonstrates how the results have high correlation. The errors bars in Figure 105 identify the standard uncertainty of the results. Where the standard uncertainty of two data sets overlap, this illustrates where the results are not statistically different. To help understand this relationship, the valve profiles from the 0W20 EIVC 540° Lift results in Table 32 are considered.

Figure 106 illustrates the valve lift profiles of the EIVC 540° valve lift results at 500rpm. Figure 106 demonstrates a Full Lift valve lift profile captured at 500rpm and 35°C (black dashed line labled 'FL 35°' on Figure 106). This reference helps understand how EIVC lift is achieved and also the definition of the closing angle of 540CA°.

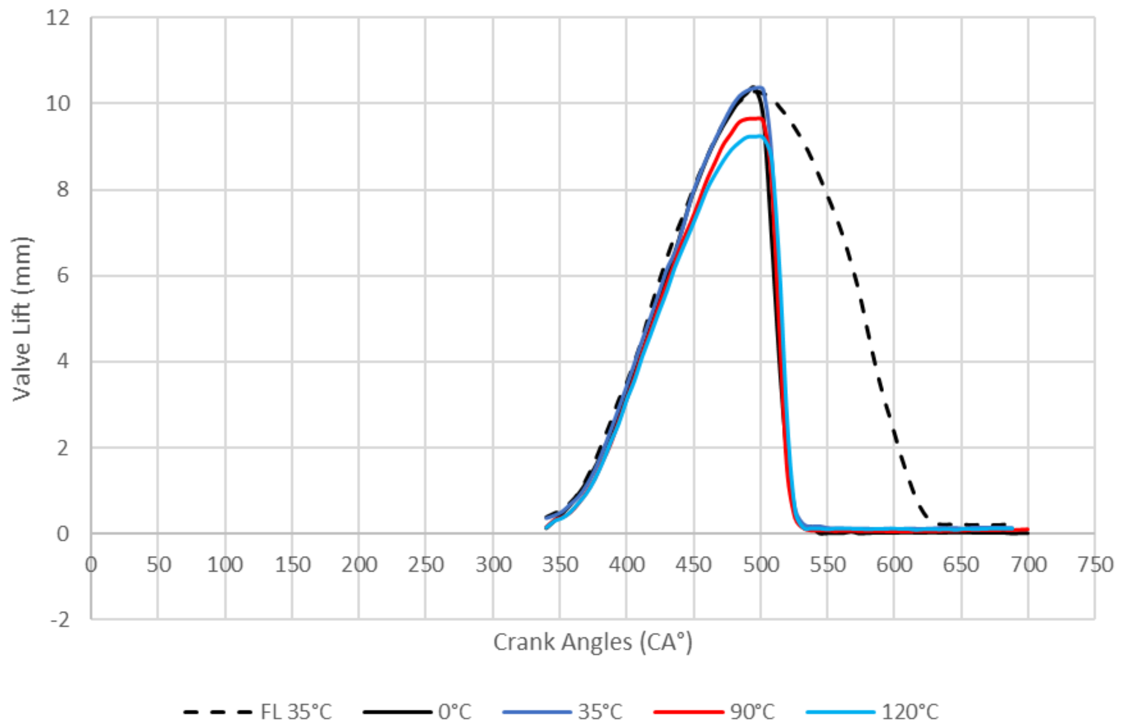


**FIGURE 106 - VALVE LIFT PROFILES OF EIVC LIFT MODE AT 500RPM USING 540CA° AND VARYING OIL TEMPERATURES**

Figure 106 identifies the closing angle is set at 540CA° and the valve lift profile ends at 540CA°. However, when considering the simulated results in section 4.2.1, Figure 35 shows the simulated EIVC 520° valve lift at 375rpm, 1500rpm and 3100rpm camshaft speeds using 0W20 oil, while varying oil temperature (Figure 35 (a)). This demonstrates the closest camshaft speed measurement point in comparison to simualted results at 375rpm. As observed in Figure 106, the simulated EIVC valve lift has a closing angle set to 520CA° and the valve lift profile ends at 520CA°. This demonstrates this rig's validity by indicating the control of the valve lift is the same as experienced in the simulated results. However, Figure 35 (a) suggests the opening crank angle is approximately 310°, in comparison to

Figure 106 which higlights the opening crank angle is approximately 340°. This could suggest the limitations of simulated testing, or the simulated test is purposely set to have a different opening angle. To further understand this relationship, the valve lift profiles with increasing camshaft speed are considered.

Figure 105 demonstrates as the camshaft speed increases, valve lift deteriorates dramatically across all oil temperatures. At 1500rpm, the valve lift at 0°C can be seen to decrease at a faster rate than all other temperatures studied. This disagrees with the literature, as it is found volumetric efficiency increases with increasing viscosity. To illustrate this relationship, the valve profiles from the 0W20 EIVC 540° Lift results at 1500rpm are investigated in Figure 107.

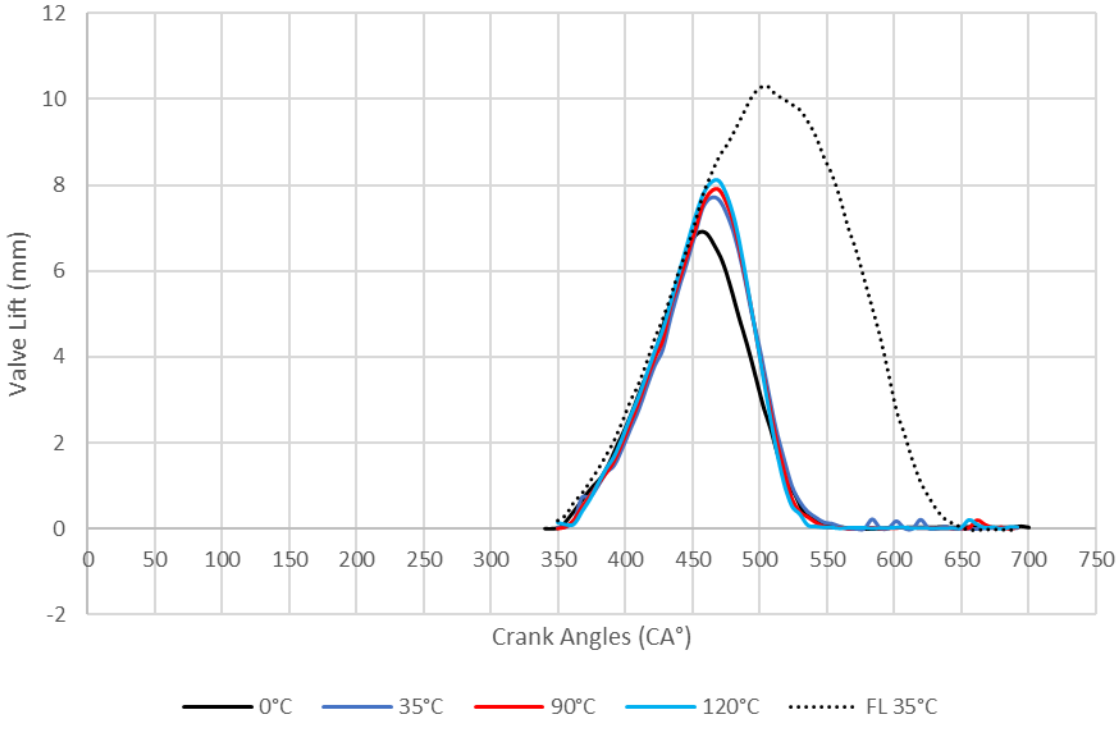


**FIGURE 107 - VALVE LIFT PROFILES OF EIVC LIFT MODE AT 1500RPM USING 540CA° AND VARYING OIL TEMPERATURE**

As observed in the valve lift profile at 500rpm in Figure 106, Figure 107 demonstrates a Full Lift valve lift profile captured at 1500rpm and 35°C (black dotted line labled 'FL 35°' on Figure 107). This reference helps illustrate how EIVC lift is achieved and also the definition of the closing crank angle of 540°. The closing crank angle is set at 540° and the valve lift profile ends at 540°.

Figure 107 identifies a limitation in the design of the UniAir. Section 5.5.3 highlights the influence of oil aeration on hydraulic components. Dissolved air has no considerable influence on the physical parameters of an oil. However, undissolved gas fractions (dispersed gas and surface foam) have a direct influence on the physical properties of an oil and a strong influence on the compressibility of a fluid. It is explained two main parameters effect oil aeration, oil temperature and pressure. Figure 47 illustrates the saturation limit of a liquid which enables air to be dissolved or undissolved. This indicates that with reducing pressure allows more air to be undissolved in the system. The nature of EIVC lift releases pressure every revolution to achieve the closing crank angle, in this case 540°. Evidence in Figure 107 suggests this creates dynamic effects and flow restrictions in the oil circuit causing a drop below saturation pressure limit, increasing the amount of undissolved air in the UniAir. With increasing camshaft speed, this cycle of generating and releasing pressure happens more frequently. Therefore, the increased camshaft speed at 1500rpm in Figure 107 suggests how undissolved air is affecting the compressability of the oil and reducing the valve lift height away from the intended height. Furthermore, increasing the oil temperature allows more air to be dissolved in the oil. Therefore, Figure 107 provides additional evidence of oil aeration, as higher oil temperature valve lift profiles generate higher valve lift height. This suggests the colder oil temperatures will allow more undissolved air in the oil and reducing overall valve lift.

Figure 35 (b) shows the simulated EIVC 520° valve lift at 1500rpm. As the closing crank angles are 20° different between the results and simulated tests, it was expected that the simulated valve lift at 1500rpm would have lower valve lift. However, as observed at 500rpm, the opening crank angle is approximately 310° for 1500rpm simulated results in comparison to 340° observed in the results. This completely changes the valve lift charactersitcs at this speed and direct comparisons cannot be made. However, when comparing Figure 35 (a) and Figure 35 (b), it can be seen with increasing camshaft speed, Figure 35 (b) at 1500rpm shows the valve lift profile reducing at an earlier crank angle than Figure 35 (a), which suggests the simulated study has taken into consideration the oil aeration effects on the performance of the UniAir.

Figure 105 demonstrates as the camshaft speed increases further to 3000rpm, valve lift deteriorates again across all oil temperatures studied. Figure 105 identifies at 3000rpm, valve lift height improves with lower viscosity. To illustrate this relationship, the valve profiles from the 0W20 EIVC 540° Lift results at 3000rpm are illustrated in Figure 108.

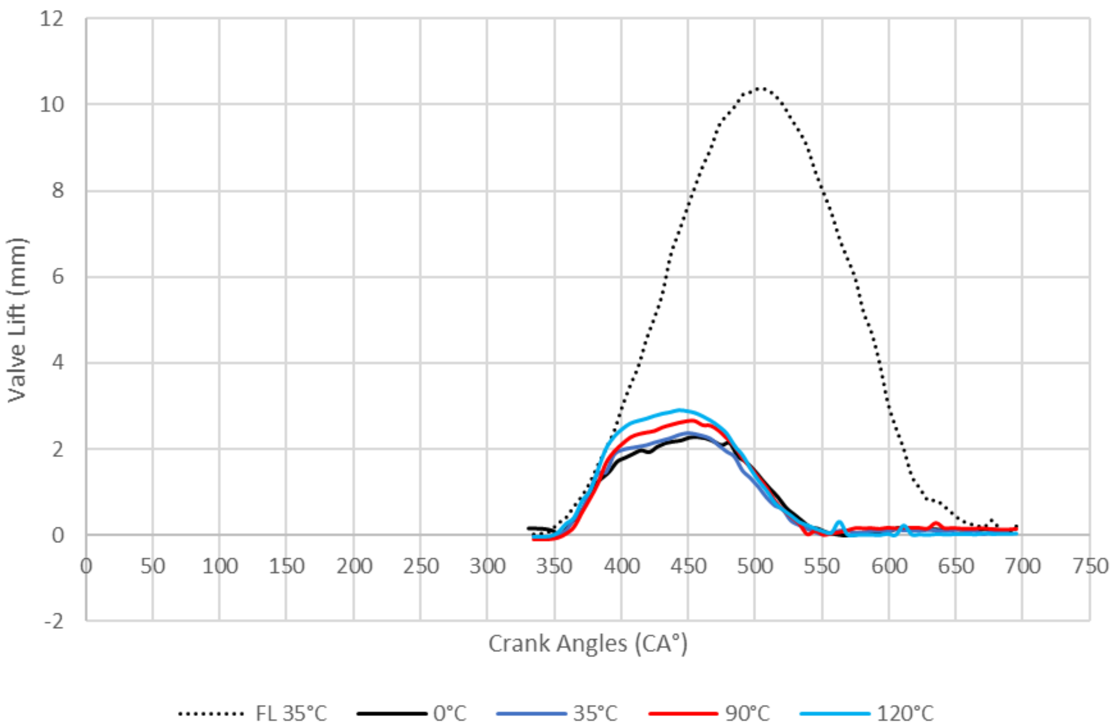


**FIGURE 108 -VALVE LIFT PROFILES OF EIVC LIFT MODE AT 3000RPM USING 540CA° AND VARYING OIL TEMPERATURE**

Figure 108 demonstrates a Full Lift valve lift profile captured at 3000rpm and 35°C (black dotted line labled 'FL 35°' on Figure 108). This reference helps illustrate how EIVC lift is achieved and also the definition of the closing crank angle of 540°. The closing crank angle is set at 540° and the valve lift profile ends at 540°.

Figure 108 further illustrates the reduced valve lift across all oil temperatures studied. Koch et al explain oil aeration increases with increasing engine speed as this increases the oil flow, which subsequently has a shorter time for air bubbles to escape from the oil while being pumped. Figure 105 directly agrees with this as the valve lift height reduces with increasing camshaft speed. Furthermore, higher oil temperatures produce higher valve lift, suggesting more air has been dissolved in the oil thus producing better compressability.

Figure 35 (c) shows the simulated EIVC 520° valve lift at 3100rpm. This directly agrees with the findings in Figure 108, as the valve lift profile achieves the set closing crank angle, however the valve lift overall height reducing with increasing camshaft speed. Furthermore, Figure 35 (c) illustrates that higher oil temperatures produce higher valve lift height. However, as mentioned previously, the closing crank angles are 20° different between the results and simulated tests, it was expected that the simulated valve lift at 3100rpm would have lower valve lift. However, Figure 35 (c) demonstrates the opening crank angle is approximately 310° for 3100rpm simulated results in comparison to 340° observed in the results. This completely changes the valve lift charactersitcs at this speed and direct comparisons cannot be made.

The results agree with the literature (Section 5.5.3) and have a high correlation with previous simulated results (Section 4.2). Additionally, it has illustrated the benefits of experimental testing in comparison to simulated testing, as the results have identified the effects of oil aeration in the valve lift profile while also considering oil leakage. Consequently, this has given a greater understanding of how oil viscosity influences the valve lift efficiency, but more importantly the results provide evidence in the objective of validating the rig with previous simulated tests.

#### 8.3.1.2 VALIDATING EIVC LIFT TORQUE

This section analyses the torque characteristics of the UniAir while using EIVC lift mode. Figure 109 illustrates the data from Table 32 showing the relationship between torque against camshaft speed while varying UniAir Avg oil temperature in EIVC lift mode. Unlike Full Lift and No Lift, the torque characteristics of EIVC do not increase with increasing camshaft speed. This is theorized due to the oil aeration (Section 8.3.1.1), which identified the valve lift performance of the UniAir is severely compromised with increasing camshaft speed.

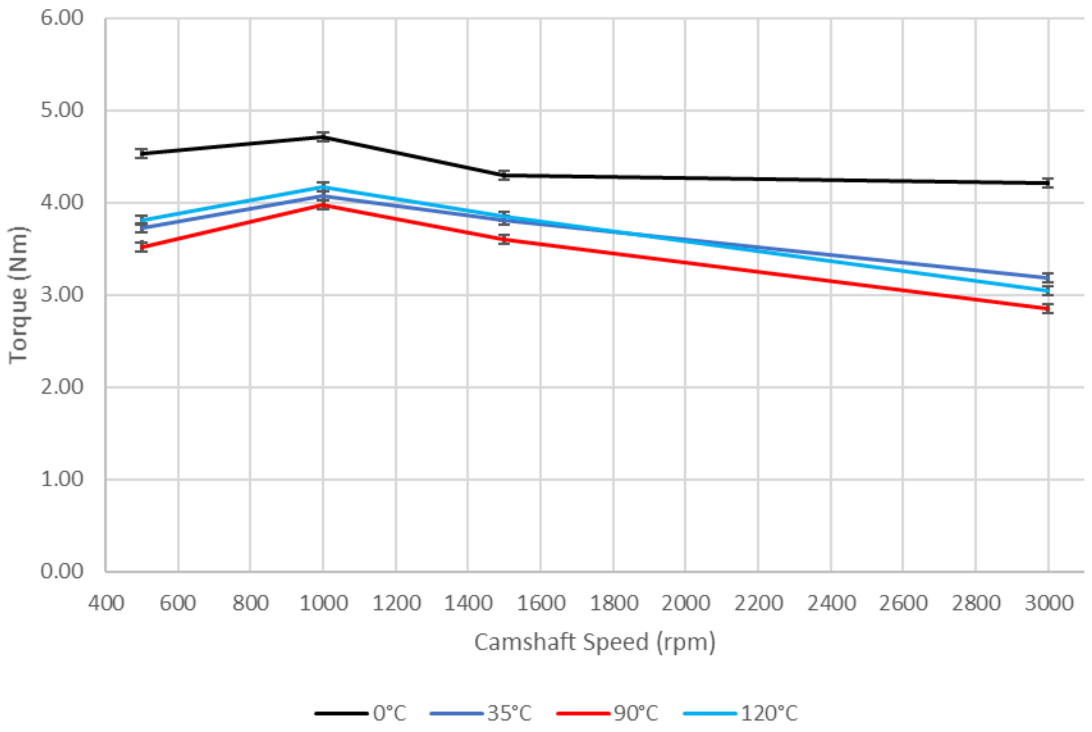


**FIGURE 109 - TORQUE AGAINST CAMSHAFT SPEED TEST RESULTS FOR EIVC LIFT MODE, USING 0W20 OIL**

Consequently, the reduction in valve lift then reduces the required torque to operate the UniAir. Nonetheless, Figure 109 demonstrates as expected high oil viscosities at 0°C generate the highest levels or torque at all camshaft speeds due to the increasing viscosity which increases the required energy to shear the oil. For oil temperatures at 35°C, 90°C and 120°C it can be seen the error bars overlap at various points suggesting the results are not significantly different. Therefore, it is assumed the physical limitations of the design of the UniAir have the highest influence on the torque characteristics and not oil viscosity.

Schaeffler's previous experimental test results in EVIC lift mode (Section 4.1.1.2, Figure 32) of the non-production UniAir, show varying camshaft speed and oil temperatures using 0W20 oil. For validation, the results from Table 32 are compared to the EIVC results (Section 4.1.1.2). To gain a better understanding of this comparison, the results from Schaeffler's previous experimental test results (Section 4.1.1.2, Figure 32) for EIVC torque is plot on the same graph (Figure 110) as the torque results from Figure 109. Furthermore, the results from 0°C are not included to allow direct comparisons between tests.

Figure 110 illustrates the comparison of EIVC results and previous experimental results (Section 4.1.1.2, Figure 32). It is observed the results have a high correlation across most camshaft speeds. However, at 500rpm camshaft speeds there is a difference in results as Schaeffler results are approximately all 0.5Nm higher than EIVC results from Table 32. This relationship is also observed in Full Lift and No Lift results, specifically at low camshaft speeds. This difference could be to do with the oil pressure supplied to the UniAir. As the results in Table 32 are referenced to an oil pressure of 3 bar (Table 31 ) across all camshaft speeds studied. Schaeffler varied the oil pressure dependant on camshaft speed (Section 4.1, Table 3) to replicate the working conditions of the engine.

Alternatively, the differences are theorised to be the UniAir Schaeffler tested, being non-production, and therefore design/manufacturing changes on the production UniAir is the reason for changes in torque. Additionally, it could be due to the different testing methodologies implemented, as strong evidence shown in Chapter 7, highlights issues (for this study) with Schaeffler's oil temperature measurement system and therefore torque measurement. Nonetheless, with increasing camshaft speeds there is a high correlation with the results which demonstrates additional validation on the rig against previous experimental results.

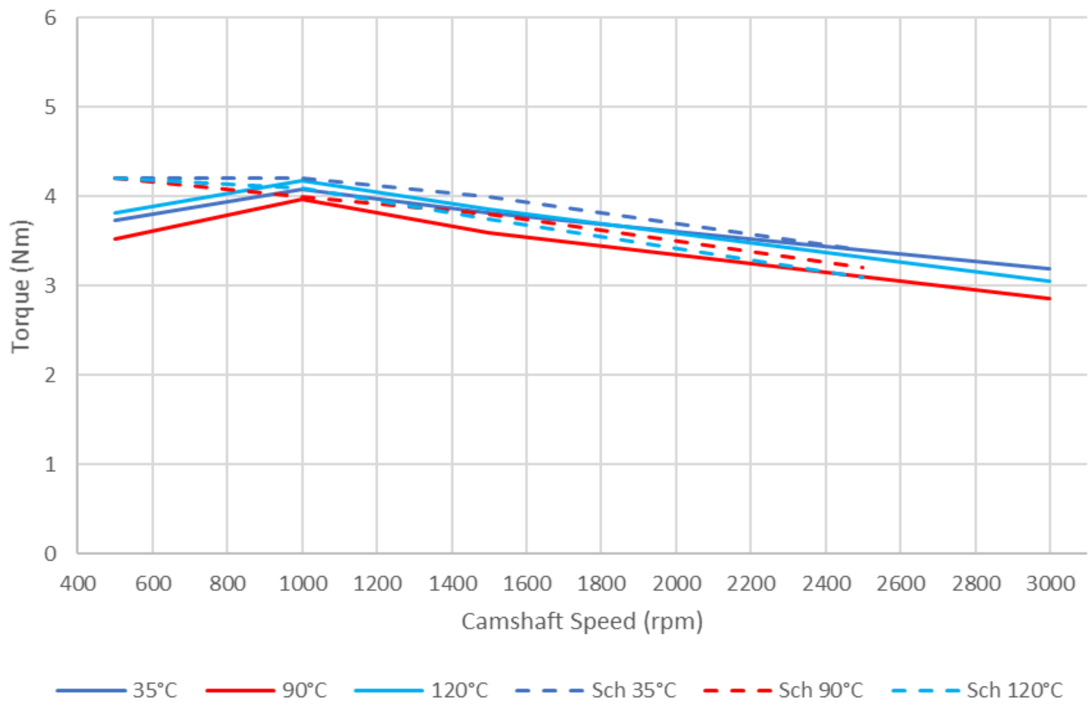


**FIGURE 110 - EIVC TORQUE AGAINST CAMSHAFT SPEED COMPARING RESULTS FROM PREVIOUS EXPERIMENTAL RESULTS**

### 8.3.1.3 Summary of EIVC Results

Section 8.1.1.3 identified how the Full Lift characteristics are analysed with an adopted method used for pumps to understand the influence of oil viscosity on the performance of the UniAir. However, the EIVC valve lift (Section 8.3.1.1) and torque results (Section 8.3.1.2) identified the negative influences of oil aeration on valve lift and subsequently torque.

The EIVC valve lift results (Section 8.3.1.1) were validated by comparing with previous simulated results (Section 4.2.1). It was found at low camshaft speeds of 500rpm, the valve lift profile performed as intended by producing maximum valve lift (for a given oil viscosity) and closing early at the set closing angle of 540°. Likewise, with Full Lift valve lift results at 500rpm (Section 8.1.1.3), it was found that valve lift deteriorates with decreasing viscosity, which is theorized due to oil leakages bypassing clearances in the UniAir. Therefore, it is hypothesised at this camshaft speed there is not enough oil aeration to influence valve lift. This directly agrees with the previous simulated results (Section 4.2.1), which demonstrates the same behavior. EIVC valve lift results (Section 8.3.1.1) and simulated results (Section 4.2.1) found with increasing camshaft speed the valve lift deteriorates which demonstrates the rig is working as expected and validates the results.

Section 8.3.1.2 shows EIVC torque results which are validated using previous experimental results (Section 4.1.1.2) from Schaeffler. It was found the results have a good correlation across all speeds apart from 500rpm. As experienced in Full Lift (Section 8.1.1.2) and No Lift (Section 8.2.1) results this difference is summarized to be due to the UniAir and testing methodology implemented, with evidence showing the testing methodology used in this work provides a better representation of the torque captured for a given oil temperature.

It was found EIVC creates the highest levels of torque across all lift modes because the energy used to create partial lift is entirely lost when the solenoid opens, releasing the oil pressure back into the intermediate pressure chamber and losing the valve spring force when past the peak cam lobe. This consistent pressure change then shifts the saturation point of the oil and consequently the amount of undissolved air in the system. The effects of oil aeration constantly change the systems characteristics and performance with increasing camshaft speed. Therefore, this makes it increasingly difficult to satisfy the objectives by understanding the influence of oil viscosity on the performance of the UniAir. This has highlighted fundamental design issues in the UniAir at certain camshaft speeds for using EIVC lift mode with a fixed closing crank angle of 540°.

Consequently, the method used to understand the hydraulic efficiency for Full Lift results (Section 8.1.1.3) cannot be used for EIVC because of the physical restrictions of the UniAir design. Therefore, to fully understand the operational performance and hydraulic efficiency of the UniAir when using EIVC lift mode, more closing crank angles need to be tested.

## 8.4 LIVO 420° Lift results

This section investigates the torque and valve lift characteristics of the UniAir in LIVO lift mode. The mechanisms for LIVO lift mode (Section 4.1.1.3), show to generate LIVO lift, the ECU produces a current to open the solenoid at a specific crank angle (CA°). Section 14.4, identifies the variation and repeatability of the UniAir when using LIVO valve lift for a specific crank angle at 420°. The test plan (Section 7.6) emphasised the large number of tests conducted in this thesis, therefore only one crank angle for the LIVO tests can be used. Consequently, this dictated the crank angle of 420° to use for validation as it replicates the variation and repeatability in Section 14.4 for comparisons of the valve lift profiles.

To validate the valvetrain test rig, results are compared to the literature and previous experimental results. To do this, the results from 0W20 are analysed first as previous experimental studies used 0W20 in their tests.

The boundary conditions outlined in Section 6.3 are implemented for the LIVO Lift tests using 0W20 oil and are presented in Table 33.

| | |
|---|---|
| Oil temperature measurement points (UniAir Avg) (°C) | 0, 35, 90, 120 ±1 |
| Oil Viscosity (cSt) | 313.3, 50.5, 9.8, 5.5 |
| Oil Pressure (bar) | 3 bar ± 0.1 |
| Camshaft speeds (rpm) | 500, 1000, 1500 ±10 |
| Oil Type | Castrol SAE 0W-20 A12013PBABOT936 |
| Opening Crank Angle (CA°) | 420° |

**TABLE 33 - BOUNDARY CONDITIONS FOR LIVO TESTS USING 0W20 OIL**

The results for LIVO tests using 0W20 oil are found in Table 34. This identifies a torque and valve lift value specific to the UniAir Avg oil temperature and camshaft speed. The baseline torque and valve lift (Section 7.5) have been illustrated in Table 34 (yellow boxes), which are used in later calculations.

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 313.3 | 50.5 | 9.8 | 5.5 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 1.25 | 0.68 | 0.82 | 1.22 |
| | 1000 | 1.96 | 1.20 | 1.22 | 1.53 |
| | 1500 | 2.68 | 1.87 | 1.63 | 1.90 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 6.56 | 6.56 | 6.49 | 6.33 |
| | 1000 | 6.88 | 6.87 | 6.77 | 6.70 |
| | 1500 | 7.07 | 7.07 | 7.01 | 6.97 |

**TABLE 34 - LIVO TEST RESULTS OF 0W20 OIL, SHOWING TORQUE AND VALVE LIFT FOR VARYING VISCOSITY AND CAMSHAFT SPEEDS**

#### 8.4.1.1 Validating LIVO valve lift

For validation of the valve lift performance while using LIVO, the variation and repeatability data published by Schaeffler (Section 14.4) is used, as there is no previous simulated results. This section analyses the valve lift characteristics of the UniAir while using an opening crank angle of 420° as it replicates the variation and repeatability in Section 14.4 for LIVO mode. Figure 111 illustrates the data from Table 34 showing the relationship between valve lift against camshaft speed while varying UniAir Avg oil temperature. It is observed the difference in valve lift across all oil temperature studied is very small.

Similar to the results in Full Lift (Section 8.1.1.1), it can be observed in Figure 111 higher oil temperatures produce lower valve lift across all camshaft speeds studied. This is comparable to hydraulic pumps, where the loss of the volumetric efficiency is produced by high temperature low viscosity lubricants bypassing pump clearances causing leakages. Koehler and Jackson agree (35) with this finding as they also found higher viscosity characteristics greatly improved volumetric efficiency.

At 0°C and 35°C where oil viscosities are higher, Figure 111 demonstrates how the results have high correlation at all camshaft speeds. The error bars in Figure 111 identify the standard uncertainty of the results. Where the standard uncertainty of two data sets overlap, this illustrates where the results are not statistically different. There is no statistical difference in valve lift results across all data points in Figure 111 for oil temperatures at 0°C and 35°C. This suggests that the maximum possible lift is achieved with the oil viscosities at 0°C and 35°C. Alternatively, the only data point to be statistically different to the rest of the data set is at 500rpm and 120°C oil temperature. To help understand this further, the valve profiles from the 0W20 LIVO lift results at 500rpm are considered.

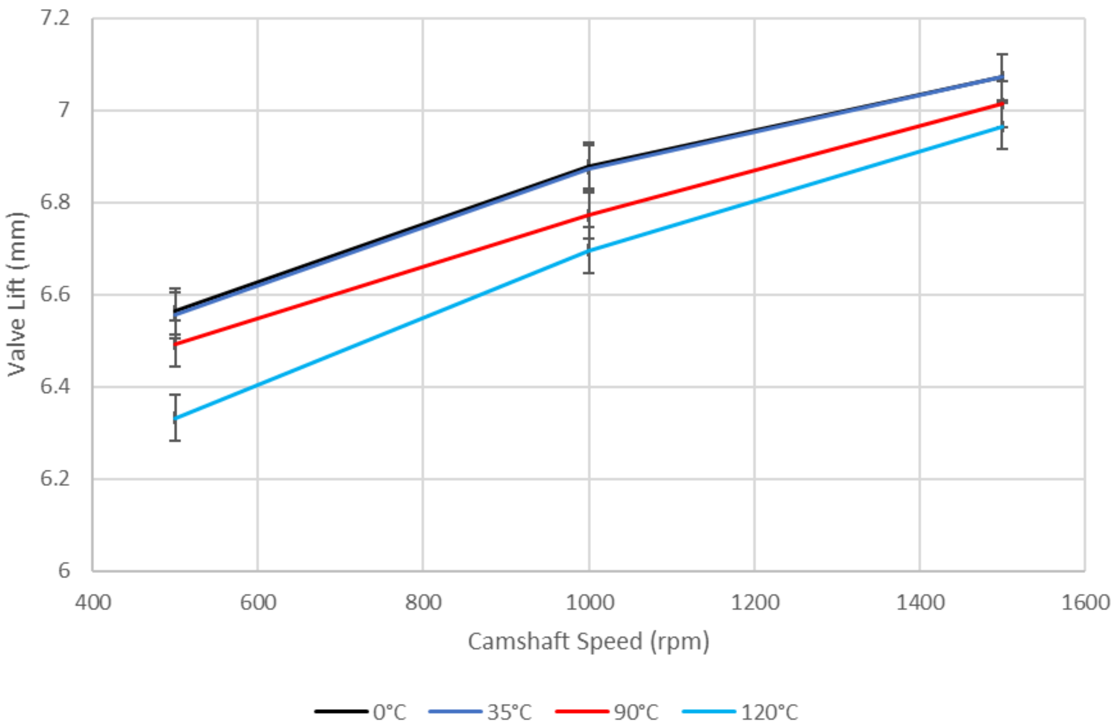


**FIGURE 111 - VALVE LIFT AGAINST CAMSHAFT SPEED TEST RESULTS FOR LIVO LIFT MODE, USING 0W20 OIL.**

Figure 112 demonstrates the LIVO valve lift profiles with valve lift against crank angle. Figure 112 shows a Full Lift valve lift profile captured at 500rpm and 35°C (black dotted line on Figure 112) used as a reference to help illustrate how LIVO lift is achieved and also the definition of the opening crank angle of 420°. With an opening crank angle of 420° Figure 112 demonstrates the LIVO lift valve lift starts opening at 420° which further demonstrates this rigs validity by indicating the control of the valve lift is correct.

As presented in the Full Lift valve lift results (Section 8.1.1), similarly, Figure 111 shows how with increasing camshaft speed, the valve lift height of all oil temperatures studied also increase. Furthermore, it agrees with the Full Lift results (Section 8.1.1), as higher camshaft speeds cause valve lift values to be more precise. This is because the camshaft speed dictates how fast the cam lobe engages with the UniAir. The higher camshaft speeds generate pressure faster than low camshaft speeds. Consequently, the oil leakages have a smaller influence on the overall valve lift at higher speeds, as the oil has less time to leak out.

To investigate this relationship further, the valve profiles from the 0W20 LIVO lift results at 1500rpm are considered. Figure 113 illustrates the valve lift profiles at 1500rpm using LIVO lift mode. A Full Lift valve lift profile captured at 1500rpm and 35°C (black dotted line on Figure 113) is used as a reference. Figure 113 demonstrates with an opening crank angle of 420° the LIVO lift valve lift starts opening at 420°.

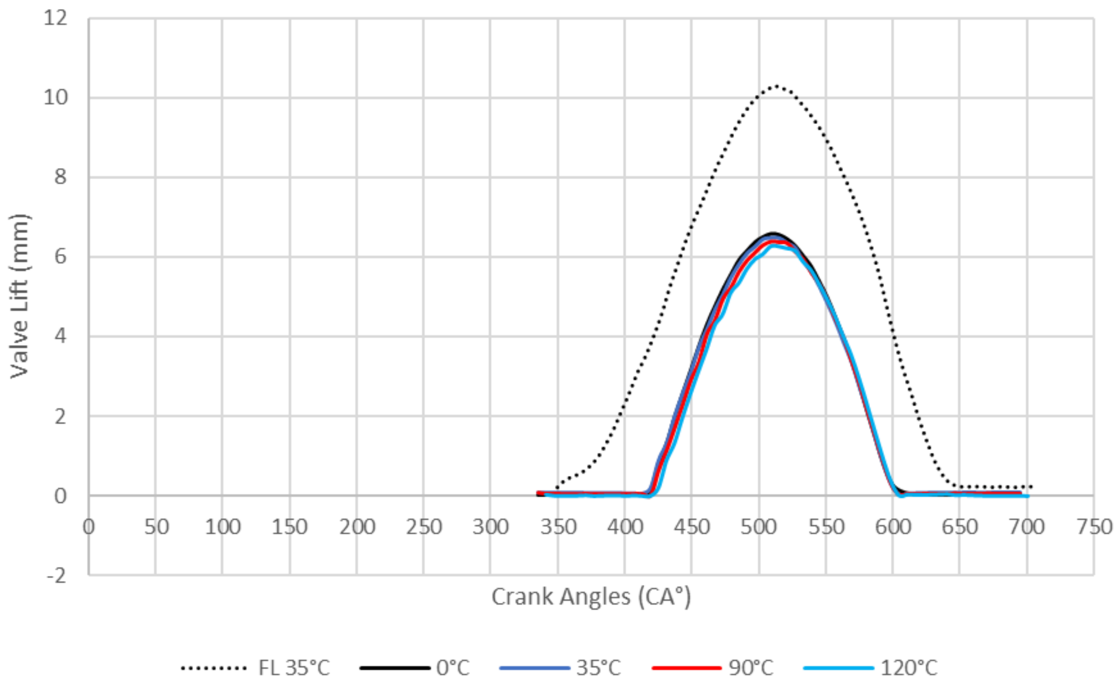


**FIGURE 112 - VALVE LIFT PROFILES OF LIVO LIFT MODE AT 500RPM USING 420° OPENING CRANK ANGLE WITH VARYING OIL TEMPERATURES**

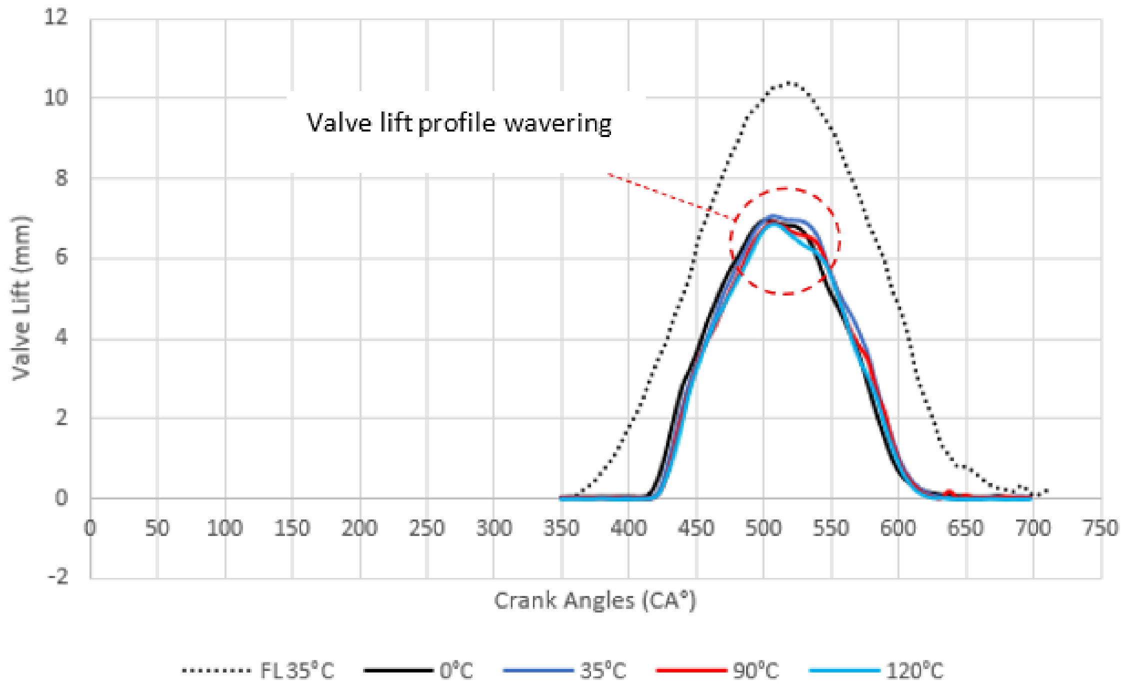


**FIGURE 113 - VALVE LIFT PROFILES OF LIVO LIFT MODE AT 1500RPM USING 420° OPENING CRANK ANGLE WITH VARYING OIL TEMPERATURES**

Comparing the valve lift profiles at 1500rpm to 500rpm, Figure 113 illustrates the valve lift profiles wavering. Figure 33 (Section 4.1.1.3), indicates the LIVO torque results from Schaeffler. It was shown at speeds of 1000rpm and 1500rpm there are disadvantages using LIVO compared to Full Lift due to hydraulic losses from “bottle neck solenoid valve”. Although this is in reference to torque, this could help understand the valve lift waving observed in the valve lift profile in Figure 113. Alternatively, due to the mechanisms of LIVO lift mode, it is explained the camshaft speed dictates how fast the cam lobe engages with the UniAir. The higher camshaft speeds generate pressure faster than low camshaft speeds. Therefore, the valve lift waving could be due to oil aeration. As experienced in EIVC, the instantaneous pressure change via the solenoid generates high levels of oil aeration, which increased with camshaft speed. It is shown in Figure 112 at 500rpm, there is no significant influence from oil aeration (illustrated with smooth curves on valve lift profile) due to the camshaft speed, however at 1500rpm in Figure 113 the effects of oil aeration can potentially be seen due to valve lift wavering.

The results show a loss in volumetric efficiency with increasing temperature which agrees with the literature (Section 5.5.1), however there is no previous experimental or simulated valve lift for comparisons. Alternatively, the valve lift is validated by comparing the the control of the valve lift with the set parameters for the opening crank angle of 420°.

#### 8.4.1.2 Validating LIVO Torque

This section analyses the torque characteristics of the UniAir while using LIVO lift mode. Figure 114 illustrates the torque data from Table 34, showing the torque characteristics of the UniAir in LIVO mode, while varying UniAir Avg oil temperature and camshaft speed.

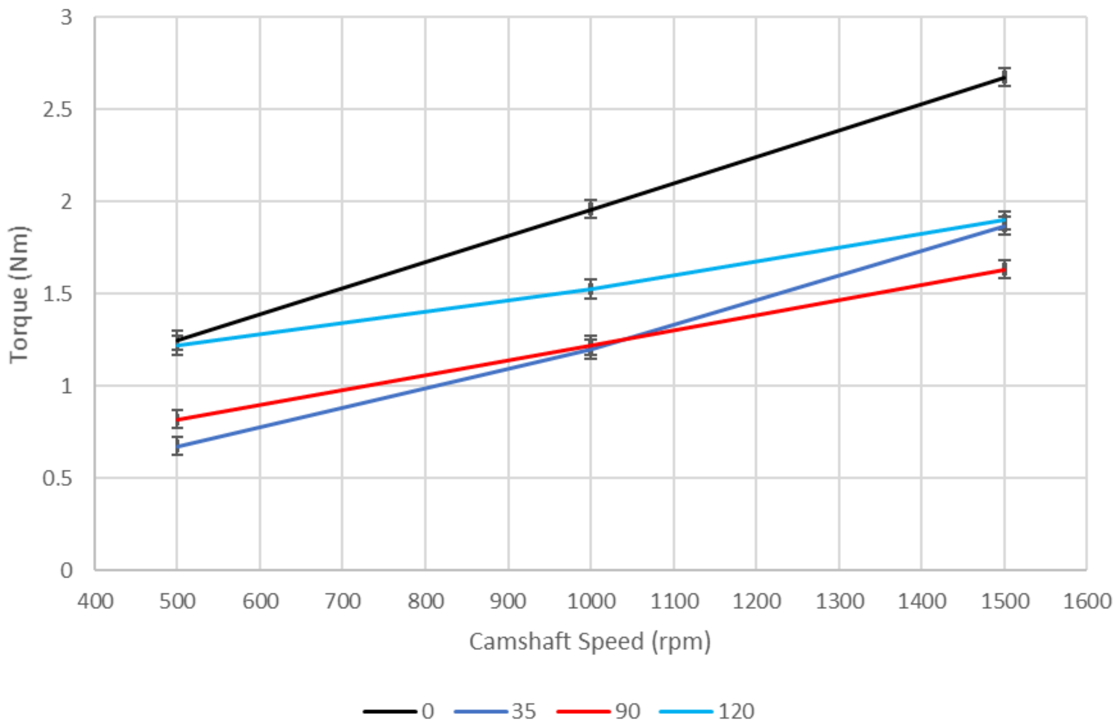


**FIGURE 114 - TORQUE AGAINST CAMSHAFT SPEED FOR LIVO MODE, USING 0W20 OIL**

Figure 114 shows increasing camshaft speed increases torque at all oil temperatures studied. Similarly, the relationship observed for Full Lift (Section 8.1.1.2, Figure 89), the relationship observed in Figure 114 is assumed to be due to the changing lubrication regime. The second highest torque characteristic is observed to be at the oil temperature of 120°C. As found for Full Lift (Section 8.1.1.2, Figure 89), the lubrication regime is dependent on the dynamic viscosity, moving part speed and load/pressure. Therefore, when the oil temperature is at 120°C, at low camshaft speeds, this high friction is theorized due to the likelihood of incomplete fluid film formation. The entraining velocity at lower speeds does not form sufficiently thick oil films to fully separate the mechanical interfaces.

Alternatively, Figure 114 demonstrates the highest torque values across all camshaft speeds investigated is at oil temperatures at 0°C. This provides evidence of the lubrication regime being hydrodynamic at this oil viscosity, which causes higher levels of shear force to overcome dynamic viscosity.

Previous experimental test results by Schaeffler (Section 4.1.1.3, Figure 33) show torque results using LIVO mode of the non-production UniAir, while varying camshaft speed and oil temperatures. For validation, the results from Figure 33 are plot on the same graph as the torque results from Figure 114. Furthermore, the results from 0°C are not included to allow direct comparisons between tests. Figure 115 shows the torque characteristics against camshaft speed of both the LIVO results from Table 34 and the results from Schaeffler in Figure 33.

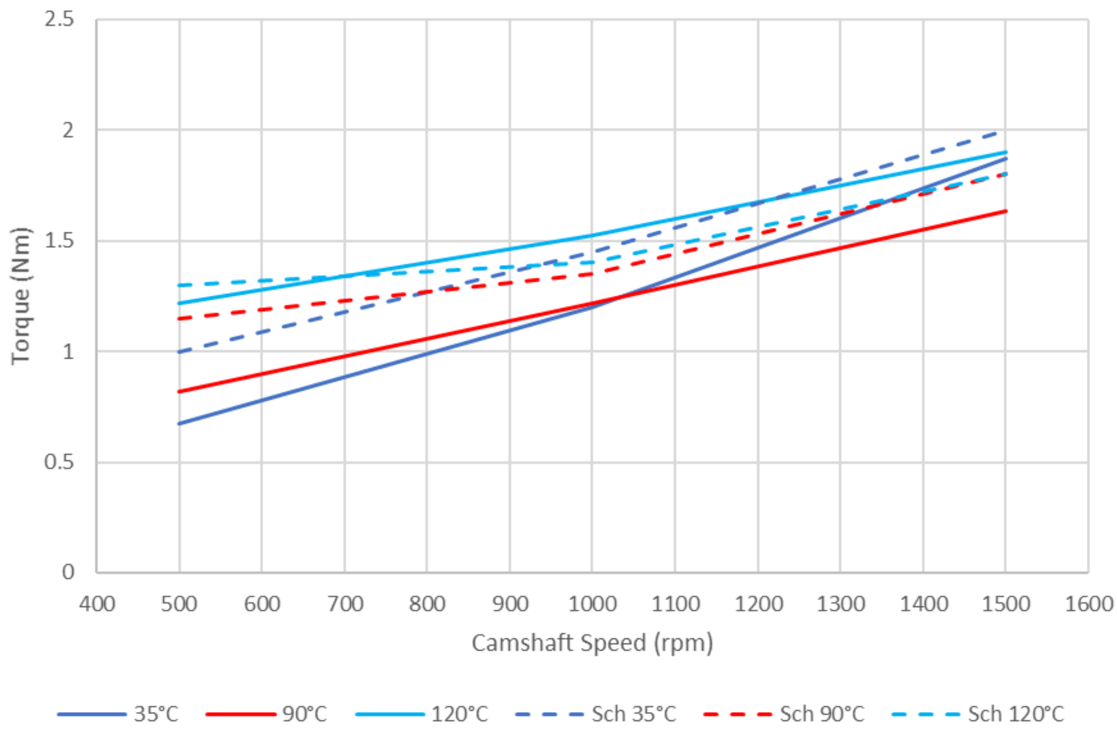


**FIGURE 115 - LIVO TORQUE AGAINST CAMSHAFT SPEED COMPARING RESULTS FROM PREVIOUS EXPERIMENTAL RESULTS**

Comparing the torque characteristics in Figure 115 at low camshaft speeds, the lowest torque is at 35°C and highest at 120°C for both data sets. However, as found in all previous lift modes, the Schaeffler values are approximately 0.5Nm higher at oil temperatures 35°C and 90°C. It is suspected again that the differences are partially due to the UniAir Schaeffler tested, being non-production, and therefore design/manufacturing changes on the production UniAir is the reason for changes in torque.

Additionally, the differences are again theorized to be due to the different testing methodologies implemented, as strong evidence shown in Chapter 7, highlights issues (for this study) with Schaeffler's oil temperature measurement system and therefore torque measurement. Further evidence for this is found in Figure 115, as the torque values for LIVO results at oil temperatures 35°C and 90°C have an almost perfectly correlated relationship at different magnitudes to Schaeffler. Alternatively, torque values at 120°C have no correlation and could suggest the actual temperature for Schaeffler's 120°C torque profile was closer to 90°C. This gives additional evidence for the requirement of a new test methodology as the values of torque are different, and therefore this test methodology has highlighted new information with regards to understanding the optimal performance of the UniAir.

#### 8.4.1.3 Validating LIVO characteristics

Section 8.1.1.3 demonstrated how to analyse the performance of the UniAir while using Full Lift mode. Likewise, LIVO torque characteristics indicate torque increases with increasing camshaft speed and would again strongly suggest that the UniAir's torque characteristics was dominated by the hydraulic pumping elements. Therefore, to quantify the influence of the oil viscosity on the UniAir, the hydraulic efficiency used in pumps was used to understand the performance and efficiency.

Section 7.5 demonstrated how to calculate the relative volumetric and mechanical efficiencies for the UniAir when considering hydraulic efficiency. Therefore using LIVO valve lift and torque results (Table 34), the adopted relative mechanical, volumetric and hydraulic efficiencies are calculated and presented in Table 35.

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Hydraulic Efficiency (%) |
|---|---|---|---|---|
| 500 | 313.3 | 0.54 | 1.00 | 54.12 |
| | 50.5 | 1.00 | 1.00 | 99.86 |
| | 9.8 | 0.82 | 0.99 | 81.46 |
| | 5.5 | 0.55 | 0.96 | 53.44 |
| 1000 | 313.3 | 0.61 | 1.00 | 61.22 |
| | 50.5 | 1.00 | 1.00 | 99.91 |
| | 9.8 | 0.98 | 0.98 | 96.82 |
| | 5.5 | 0.79 | 0.97 | 76.51 |
| 1500 | 313.3 | 0.61 | 1.00 | 61.06 |
| | 50.5 | 0.87 | 1.00 | 87.34 |
| | 9.8 | 1.00 | 0.99 | 99.17 |
| | 5.5 | 0.86 | 0.98 | 84.66 |

**TABLE 35 - TABLE OF EFFICIENCIES FOR LIVO USING 0W20 OIL**

Calculating the relative hydraulic efficiency combines both parameters torque and valve lift which are used to understand the influence of oil viscosity on the UniAir operation and efficiency.

### 8.4.2 Influence of oil viscosity on UniAir performance and efficiency using LIVO

The calculated relative hydraulic efficiency (Section 8.4.1.3, Table 35) of the UniAir while in LIVO Lift mode is used to understand the influence of oil viscosity on the performance and efficiency and optimal performance of the UniAir. Figure 116 illustrates the overall performance of the UniAir by combining both torque and valve lift to calculate the relative hydraulic efficiency. This creates a quantitative way of comparing the performance of the UniAir at different oil temperatures and camshaft speeds.

Figure 116 demonstrates that the hydraulic efficiency of the UniAir changes dependant on camshaft speed. For low camshaft speeds, the UniAir at 35°C is at least 20% more efficient than all other oil temperatures. As the camshaft speed increases, the lubrication regime changes for the higher temperature oils. At high camshaft speeds the higher temperature oils becomes more efficient. Therefore, to understand the influence of oil viscosity on the UniAir operation and efficiency, hydraulic efficiency against viscosity is presented in Figure 117.

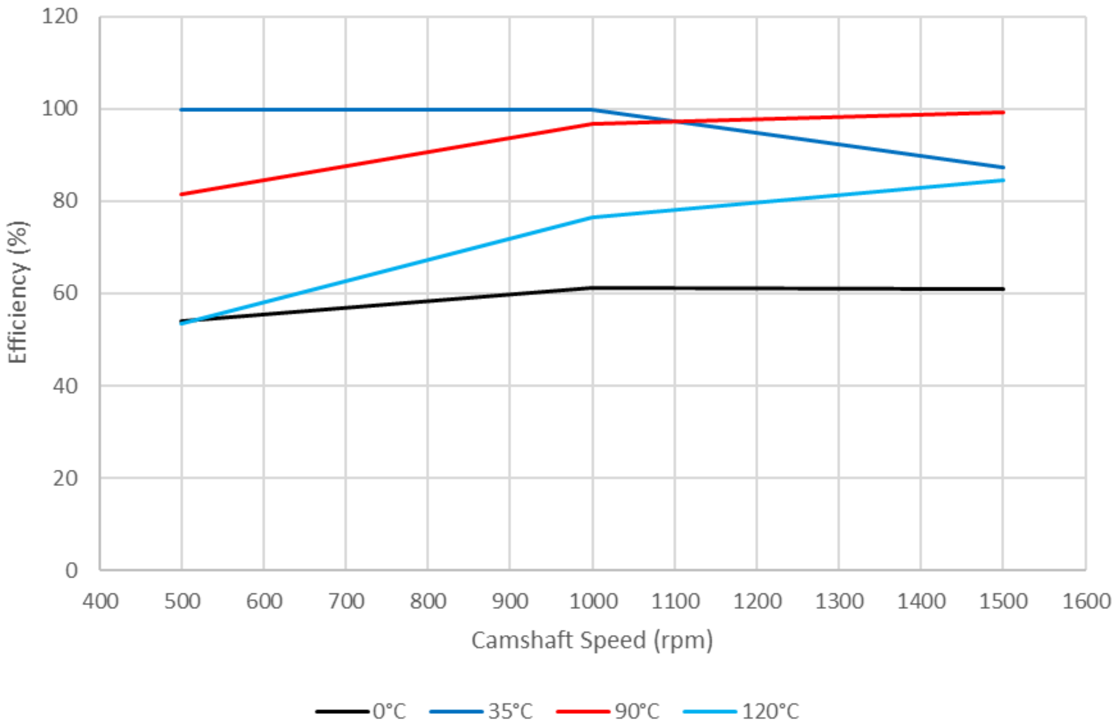


**FIGURE 116 - RELATIVE HYDRAULIC EFFICIENCY AGAINST CAMSHAFT SPEED**

Figure 117 demonstrates the relationship between the hydraulic efficiency and viscosity. Plotting each data series as camshaft speed, enables the identification of how the oil viscosity influences the UniAir operation and efficiency dependant on camshaft speed. Figure 117 shows the relative hydraulic efficiency is highly dependent on oil viscosity and camshaft speed. Therefore, to further understand the influence of oil viscosity, each camshaft speed was investigated while investigating the relative hydraulic efficiency against oil viscosity of multiple oils (Section 6.1.2).

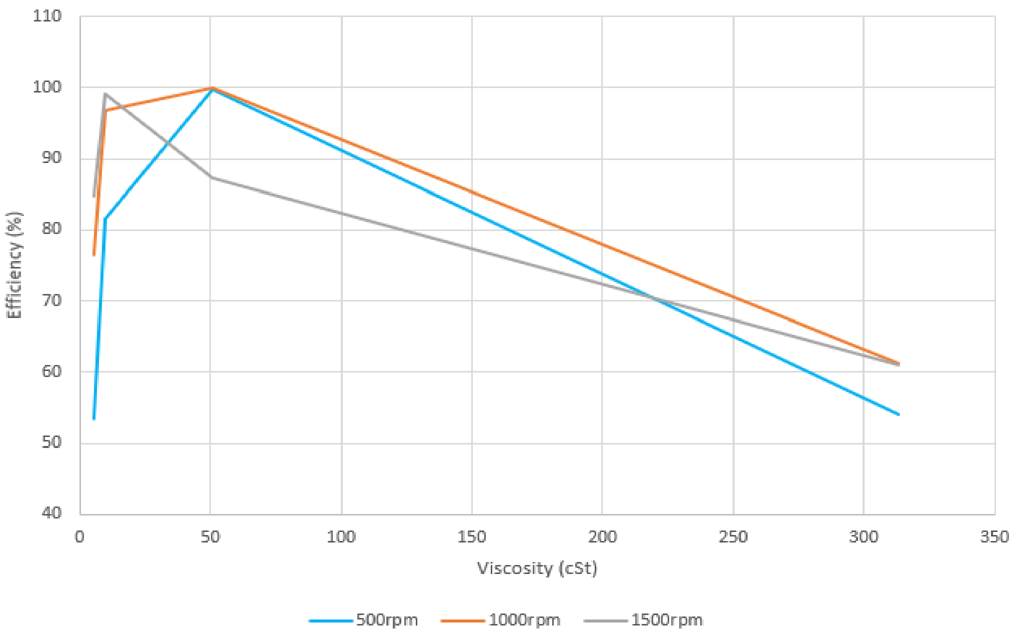


**FIGURE 117 - RELATIVE HYDRAULIC EFFICIENCY AGAINST VISCOSITY FOR 0W20 OIL AT DIFFERENT CAMSHAFT SPEEDS**

### 8.4.3 Influence of oil viscosity on UniAir efficiency in LIVO lift mode

This section investigates the influence of oil viscosity of multiple oils to determine the optimal performance of the UniAir. Results for 5W30 and Hyspin oil are captured and analysed in the exact same approach as 0W20 results (Section 8.4.1.3). Therefore, the results show efficiency against viscosity for a specific camshaft speed for all oils studied.

#### 8.4.3.1 Influence of oil viscosity at 500rpm on UniAir performance and efficiency

Figure 48 highlights the differences in viscosity for a specific temperature. Figure 118 illustrates the relative hydraulic efficiency against oil viscosity at 500rpm of the three oils. It is shown that there is a high correlation between the results of all three oils. This suggests that the UniAir performance is heavily dictated by the oil viscosity and not the oil additives. All three oils have different additive packages, however the relative hydraulic efficiency at this speed shows that all three oils optimally perform at approximately 50cSt.

For higher oil viscosities at oil temperatures of 0°C, the efficiency is approximately 40% lower than the optimal region at 50cSt. Although volumetric efficiency is at its highest for higher oil viscosities, the mechanical efficiency is the worst. The mechanical efficiency is reduced at high viscosities due to the shear strength of the oil at these viscosities (Section 8.4.1.2). The combination of high dynamic viscosity in the system, as well as hydrodynamic lubrication regimes outweighs the positives of high volumetric efficiency.

Alternatively, Figure 118 demonstrates that at low viscosities of 5cSt is equally as bad as high viscosities. However, for low viscosities the reduced relative hydraulic efficiency is due to oil leakages reducing volumetric efficiency, and incomplete fluid films reducing mechanical efficiency. This highlights how the relative hydraulic efficiency is optimal at approximately 50cSt. It is the compromise of lower volumetric efficiency than at higher viscosities, but the mechanical efficiency is optimum. This is due to the viscosity being thick enough to generate mixed or hydrodynamic lubrication regimes for mechanical interfaces, but also not being too thick so the energy required to shear the oil is low.

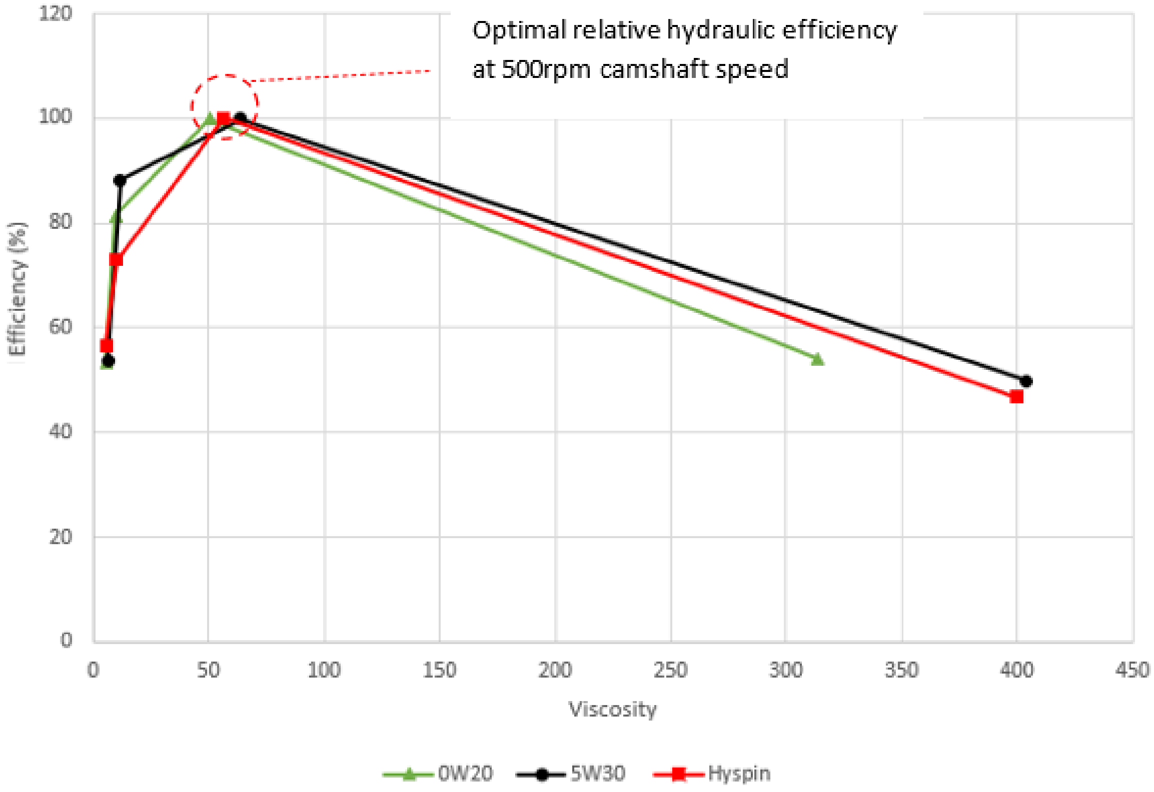


**FIGURE 118 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 500RPM**

### 8.4.3.2 Influence of oil viscosity at 1000rpm on UniAir performance and efficiency

This section investigates the influence of oil viscosity at 1000rpm of three oils, 0W20, 5W30 and Hyspin. Figure 119 highlights the relative hydraulic efficiency against oil viscosity at 1000rpm. It was found that the optimal relative hydraulic efficiency for camshaft speeds of 1000rpm is approximately 50cSt for all oils. However, it can be seen that the relative hydraulic efficiency for both engine oils 0W20 and 5W30 at 10cSt also demonstrates high overall efficiency. This shows the transitioning of the optimal region due to changing lubrication regime with increasing camshaft speed.

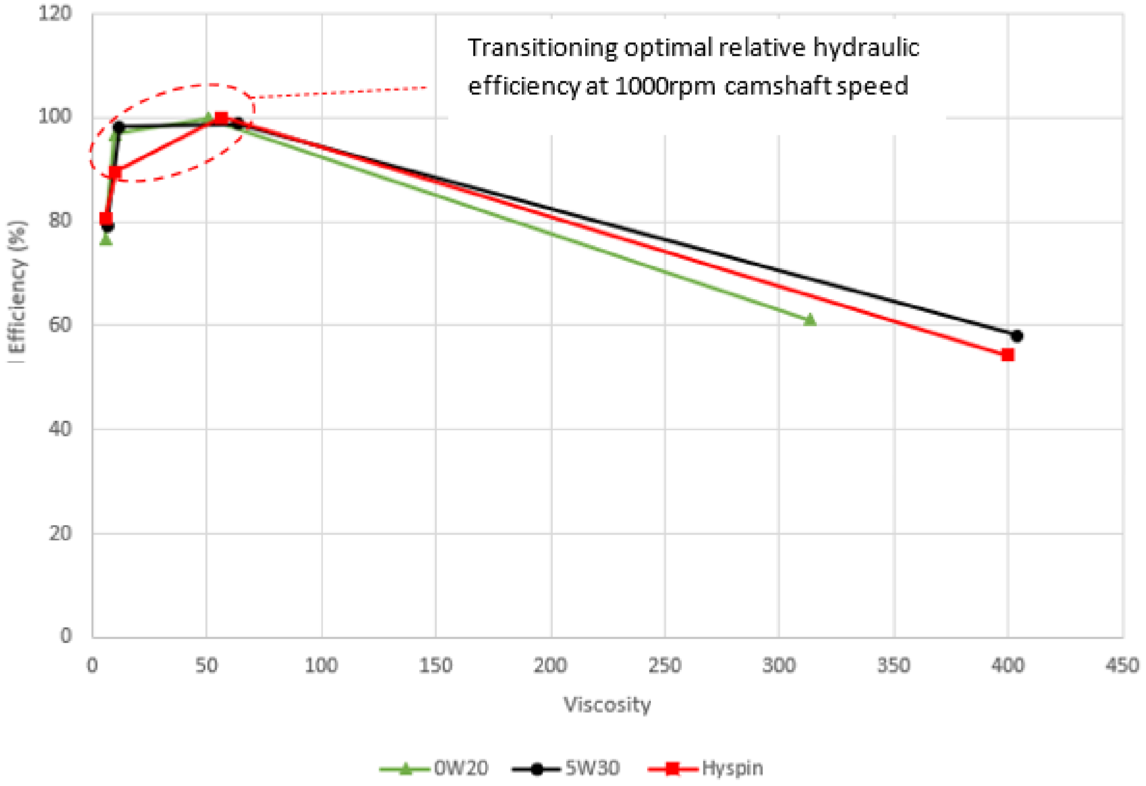


**FIGURE 119 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 1000RPM**

Alternatively, for the Hyspin oil, it can be observed that the relative hydraulic efficiency at 10cSt is lower than the engine oils. As the viscosities are very similar the difference in relative hydraulic efficiency is assumed to be due to the additives. To fully understand and compare between the performances of oils is beyond the scope of this research, however Figure 119 does demonstrate there are applications where the influence of oil additives are experienced in the UniAir.

#### 8.4.3.3 Influence of oil viscosity at 1500rpm on UniAir performance and efficiency

Figure 120 illustrates the relative hydraulic efficiency against oil viscosity at 1500rpm using three oils, 0W20, 5W30 and Hyspin. With the increase in camshaft speed, it can be observed in Figure 120 the optimal viscosity for highest relative hydraulic efficiency is at approximately 10cSt for all oils studied. The relative volumetric efficiency results in Table 35 show a high efficiency across all temperatures studied. Therefore, the relative overall efficiencies are more influenced by the relative mechanical efficiencies. This further demonstrates the change in efficiency is due to the lubrication regime due to changing viscosity and camshaft speeds. At this speed, the entraining velocity is fast enough to generate an oil film thick enough to separate the surfaces with an oil viscosity of approximately 10cSt. Consequently, at this viscosity requires less energy to shear the volume of oil inside the UniAir, resulting in higher mechanical efficiencies at higher camshaft speeds.

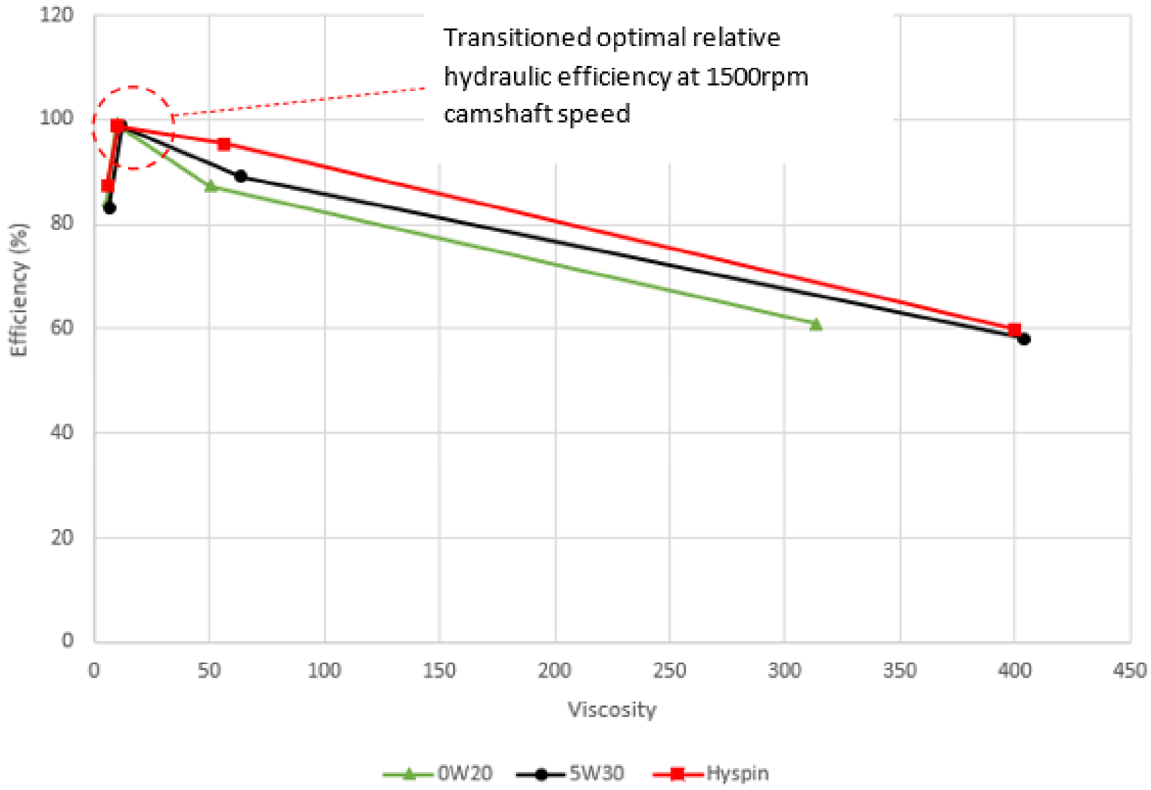


**FIGURE 120 - RELATIVE HYDRAULIC EFFICIENCY AGAINST OIL VISCOSITY AT 1500RPM**

### 8.4.4 Summary of LIVO results

The torque and valve lift results of the UniAir while using LIVO lift mode have been presented and analysed (Sections 8.4.1.1 and 8.4.1.2) allowing for comparisons to previous experimental (Section 4.1) to validate the rigs performance. The operation of the UniAir while using LIVO lift mode was evaluated by measuring the valve lift and torque while varying camshaft speeds and oil viscosities

As found in the Full Lift results (Section 8.1.1), LIVO lift results (Section 8.4.1.1) demonstrates how valve lift is directly related to both oil viscosity and camshaft speed. It was found that valve lift deteriorates with decreasing viscosity across all camshaft speeds investigated, which is theorized due to oil leakages bypassing clearances in the UniAir. However, LIVO results found the deterioration of valve lift due to oil temperature was less than those experienced in Full Lift, especially at 120°C oil temperature. This is suggested due to the way LIVO lift mode actuates the valves. With a late opening crank angle of 420°, this closes the solenoid at a higher position on the cam lobe. At this point, the gradient on the cam lobe is higher, which generates higher pressures in the oil circuit within the UniAir at a faster rate. This high amount of oil pressure creates enough pressure to generate valve lift with oil leakages at lower viscosities. However, as there are no previous experimental or simulated results for valve lift while using LIVO lift mode, validation was determined by the valve lift profiles starting at the designated crank angle of 420°.

At low camshaft speeds (500rpm) and low viscosities (5.5cSt) it was found valve lift height was reduced by 4% from the optimal viscosity at 313cSt. Alternatively, at high camshaft speeds (1500rpm) and low viscosities (5.5cSt) it was found valve lift height was reduced by 1% from the optimal viscosity at 313cSt. This is proposed to be due to the higher rate at which the oil pressure builds up inside the UniAir counteracting the rate at which the oil leaks and thus generating higher valve lift heights.

As found in Full Lift Results (Section 8.1.1.2), LIVO lift results (Section 8.4.1.2) shows torque is also dependent on both oil viscosity and camshaft speed. It was found that increasing camshaft speed increased torque across all viscosities studied. This behaviour is also experienced in hydraulic pumps (28), which suggested the UniAir torque is dominated by the hydraulic components inside the UniAir. It is observed that torque values at fixed camshaft speeds are different due to oil viscosities, due to dynamic viscosity and changing lubrication regime.

For high viscosities (313cSt) it was found more energy is required to shear the oil inside the UniAir, however this provided an oil film thickness to generate mixed or hydrodynamic lubrication between surfaces. It is observed at high viscosities (313cSt) and low camshaft speed (500rpm) the torque values are 45% higher than at optimal viscosities (50cSt). Alternatively, at high viscosities (313cSt) and high camshaft speeds (1500rpm) it is identified torque values are 40% higher than optimal viscosities (10cSt).

For low viscosities (5cSt) it was found the required energy to shear the oil inside the UniAir is lower. However, the oil film thickness between surfaces is too thin, causing boundary lubrication at low camshaft speeds (500rpm) generating torque values 44% higher than optimal viscosities (50cSt). Alternatively, at high camshaft speeds (1500rpm) and low viscosities (5.5cSt) it was found torque values are 14% higher than optimal viscosities (10cSt).

Validation of LIVO lift torque results are confirmed as comparisons are made with previous experimental results, which showed similar trends at different magnitudes. The differences are again theorized to be due to the different testing methodologies implemented, as strong evidence shown in Chapter 7, highlights issues (for this study) with Schaeffler's oil temperature measurement system and therefore torque measurement. This gives additional evidence for the requirement of a new test methodology as the values of torque are different, and therefore this test methodology has highlighted new information with regards to understanding the optimal performance of the UniAir.

The results have demonstrated the approach used to identify the most suitable way to understand the influence of oil viscosity on the efficiency of the UniAir, by adopting the current method for hydraulic efficiency used on traditional pumps. It was found the influence of oil viscosity on the UniAir efficiency changed dependent on camshaft speed. At low camshaft speeds (500rpm), the lowest relative hydraulic efficiency was found to be at 53% for oil viscosities at 5.5cSt while optimal relative hydraulic efficiency was found at oil viscosities of approximately 50cSt.

Alternatively, at high camshaft speeds (1500rpm) the lowest relative hydraulic efficiency was found to be at 61% for oil viscosities at 313cSt while optimal relative hydraulic efficiency was found at oil viscosities of approximately 10cSt.

As found in Full Lift results (Section 8.1), lowering the viscosity to a 0W-16 grade would add no benefit to the optimal viscosity range, which is shown true, regardless of the oil composition. This is an important overall conclusion as a move to lower viscosities (which is the current trend across all OEMs) will not benefit hydraulic valvetrains if considered in isolation. Alternatively, this means that with a move to lower viscosity grades (which tend to benefit the overall engine efficiency), there are gains to be had by optimising the hydraulic valvetrain which are currently most efficient in a slightly higher viscosity range.

# 9. DISCUSSION

Chapter 8 identified the UniAir characteristics changed dependent on the lift mode used. Consequently, the results cannot be compared directly and therefore were split into four sections dependent on the lift mode, Full Lift, No Lift, EIVC and LIVO.

Chapter 8 has validated the rig by comparing to previous experimental and simulated results, while identifying the influence of oil viscosity and defining the optimal performance of the UniAir operation. For lift modes Full Lift, EIVC and LIVO it was found that valve lift deteriorates with decreasing viscosity across all camshaft speeds investigated, which is theorized due to oil leakages bypassing clearances in the UniAir. However, it was found that valve lift height increases with increasing camshaft speed, which is proposed to be due to the higher rate at which the oil pressure builds up inside the UniAir counteracting the rate at which the oil leaks and thus generating higher valve lift heights.

It was found higher viscosities (300-400cSt) have optimal levels of volumetric efficiencies while having lower mechanical efficiencies. The mechanical efficiency which was calculated using torque, was also dependent on both oil viscosity and camshaft speed. It was found that increasing camshaft speed increased torque across all viscosities and lift modes studied. This behaviour is also experienced in hydraulic pumps (28), which suggested the UniAir torque is dominated by the hydraulic components inside the UniAir. It is observed that torque values at fixed camshaft speeds are different due to dynamic oil viscosity and changing lubrication regime.

For high viscosities (300-400cSt) it was found higher energy is required to shear the oil inside the UniAir, however provided an oil film thickness to generate mixed or hydrodynamic lubrication between surfaces. Alternatively, for low viscosities (~5cSt) it was found the required energy to shear the oil inside the UniAir is lower. However, the oil film thickness between surfaces is too thin, causing boundary lubrication at lower camshaft speeds.

The influence of oil viscosity on the UniAir efficiency changed dependent on camshaft speed. Fixing the camshaft speed to understand the influence of oil viscosity on the performance of the UniAir, it was found that there was up to 60% difference in relative hydraulic efficiency. Depending on the lift mode and camshaft speed the optimal oil viscosity for relative hydraulic efficiency was found to be between 50cSt and 10cSt.

It is identified that the optimal position for volumetric and mechanical efficiency are at different oil viscosities and camshaft speeds, therefore the optimal relative hydraulic efficiency is a compromise between the volumetric and mechanical efficiency.

# 10. CONCLUSIONS

This thesis details the design, development and validation of a motored valvetrain test rig that enables the characterisation of the performance of the UniAir system as utilised by JLR. The rig forms part of a test methodology also developed during this work, that can determine the influence of the viscosity of different lubricants to understand the performance, frictional operation and efficiency of a continuously variable valvetrain (CVVT). A testing programme was carried out, measuring the valve lift profiles and torque required to operate the continuously variable valve lift (CVVL) system, while varying oil formulation and temperatures from 0°C to 120°C, and environmental temperatures varying from 0°C to 120°C using climate chambers. The results were then analysed and used to inform recommendations and future research direction for JLR.

Chapter 6 highlights a detailed novel design methodology and rig development demonstrating the understanding and implementation of all the components on the rig. The main variables used to monitor the influence of the oil viscosity on the UniAir performance and efficiency are valve displacement (lift), torque, camshaft speed, oil temperature and pressure.

Chapter 7 demonstrates a robust and optimal test procedure, a systematic method to determine the relationships between inputs affecting the process and the outputs of that process. This systematic method was split into three main categories:

- Screening
- Optimisation
- Robustness

Screening tests (Section 7.2) discovered significant novel uncontrollable factors and the responses of the system. Two uncontrollable factors were found during the screening tests, internal heat generated, and heat lost in the oil circuit. Optimisation tests (Section 7.3) were used to understand the limitations of the rig clarifying the initial scope of the research (Section 1.1) could be achieved, in particular oil temperatures. Robustness tests (Section 7.4) were used to establish the method can produce stable, reliable, and repeatable values. Section 7.4 demonstrated in detail how the main variables were measured and captured to generate at least 95% confidence in all the results with maximum tolerances for valve lift ±0.01mm and torque values of 0.03N.m. The tolerances generated are lower than previous experimental results and specified values from Schaeffler (Section 14.4) demonstrating the validity and robustness of the rig and results.

The UniAir is a unique component which demanded a novel method in analysing its operational performance and efficiency. Section 7.5 demonstrates the adopted current method of hydraulic efficiency used on traditional pumps, to understand the hydraulic efficiency of the UniAir. This resulted in creating a novel method for analysing hydro-electronic valve trains so the operational performance and efficiency can be understood.

# 11. CONTRIBUTION TO KNOWLEDGE

Evidence in this thesis facilitates the following conclusions:

- Novel rig design. Developed and designed a valvetrain test rig that can measure the frictional operation, oil condition, speed/position and valve lift while integrating an environmental chamber for an unmodified AJ200 Jaguar engine cylinder head.
- Novel temperatures. Both air and oil temperatures have not been previously studied at 0°C on a valvetrain test rig, specifically on the UniAir.
- Novel experimental methodology. Evidence has been provided emphasising the rigs robustness, which enables higher levels of accuracy and precision in the results in comparison to previous experimental and simulated test results.
- Novel method developed for analysing the efficiency of electrohydraulic valvetrains.
- Novel results. Due to the novel experimental methodology and analysis, all results were previously unknown. This includes valve lift and torque measurements for multiple lift modes while varying the oil formulation. In addition to this, UniAir efficiency is determined while highlighting the optimal regions of performance for the variables studied.

# 12. FURTHER WORK

The adopted procedure for hydraulic efficiency demonstrates a method to understand the UniAir performance and efficiency. However, the relative hydraulic efficiency that is used, is a product of mechanical and volumetric efficiency. This assumes that the change in mechanical and volumetric efficiencies are equal when analysing the UniAir performance. Further work is needed to understand the impact of mechanical and volumetric efficiencies on the entire engine. For example, a 10% reduction in volumetric efficiency can have significant implications on the amount of air/fuel mixture into the combustion chamber, and therefore have a greater impact on the engine in comparison to a 10% increased torque in the valvetrain. Depending on the priority of mechanical of volumetric efficiency, a weighting system can be applied to the results found in this work to help further understand the optimal region. For example, if maintaining the highest volumetric efficiency is desired, this would shift the optimal oil viscosity from between 50cSt and 10cSt (depending on the lift mode and camshaft speed) to higher viscosities (300-400cSt) where volumetric efficiencies are optimal.

As illustrated in Section 7.6 it is explained any additional measurement points would generate too many tests to conduct in the time frame of this work. It is demonstrated (Chapters 7 and 8) using four oil temperatures has created novel results for understanding the UniAir performance. However, to accurately determine the optimal oil viscosity (between temperatures of 0°C and 120°C) for the UniAir performance further work is needed by investigating additional oil temperatures.

Chapter 8 identified the influence of oil viscosity on the UniAir performance with all lift modes studied. This provides further understanding into the behaviour of the UniAir and its impact on current production JLR AJ200 engines. Oil aging and oil dilution are an inevitable outcome for any internal combustion engine while in use. Sections 5.5.2 and 5.5.4 highlighted the influence of oil aging and dilution on the oil viscosity. Subsequently, this thesis has detailed the influence of oil viscosity on the UniAir operation and efficiency while defining the optimal performance for oil temperatures studied. Consequently, JLR can use this novel data to gain a better understanding of how and when oil aging and dilution will affect the performance of the UniAir over time. For example, an AJ200 engine completes 100,000 miles it can be estimated that the oil aging and dilution will change the oil viscosity. Therefore, this change in viscosity will dictate if the UniAir will be performing in the optimal region for a given speed and lift mode. When the viscosity changes the UniAir efficiency to be consistently below optimal, this will govern a time for the oil to be changed to maintain UniAir operational efficiency and thus satisfying the overall motivation for the thesis of reducing emissions.

Furthermore, it was found that not only the oil viscosity influences the efficiency, but evidence from Chapter 8 suggests oil additives also influence the UniAir efficiency. The methodology used in this work used oil temperatures for comparisons to previous experimental and simulated studies and found that oil viscosity has up to a 60% change in efficiency on the performance of the UniAir. Therefore, to gain a better understanding of how additives influence the UniAir, oils must be compared when operating at the same viscosity. This will identify the direct influence of oil additives as the oil viscosity will remain constant through comparisons. This could also help the direct impact of influence of oil aging and dilution by understanding the change in oil composition.

This thesis details a component level experimental test; therefore, it is expected that oil temperatures inside the UniAir will be heavily dictated by the mechanisms of the combustion process in an entire engine. Currently the oil used in the UniAir is circulated around the entire engine. Integrating a separate oil system specifically for the UniAir would potentially impede the amount of heat transferred into the UniAir allowing for a more controllable oil temperature to achieve the optimal oil viscosity. Furthermore, this could directly limit the amount of oil ageing and dilution the UniAir experiences as its segregated from the oil affected by the combustion process of the engine.

Chapter 8 identified the influence of the rapid change in pressure inside the UniAir specifically in EIVC and LIVO lift modes. Consequently, evidence suggests this influences the negative attributes of oil aeration. To gain a better understanding of this phenomenon it is recommended to observe the pressure inside the UniAir during operation in both the high and intermediate pressure chambers. This will identify the levels of pressure change and correlate to the influence on valve lift.

Additional EIVC and LIVO tests using different crank angles for opening and closing the solenoids will further demonstrate the influence of oil viscosity on the performance of the UniAir. It was found using a closing crank angle of 540° during EIVC lift mode, the UniAir design significantly changes at camshaft speeds above 1000rpm due to the suspected increasing oil aeration. The test rig and methodology are validated, to fully understand the influence of oil viscosity of the UniAir performance while using EIVC lift mode, a crank angle that works at all camshaft speeds is recommended.

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# 14. APPENDIX

## 14.1 OIL TYPES AND KEY DATA

| | |
|---|---|
| Castrol SAE 0W-20 A12013PBABOT936 | |
| Pour Point | -54°C |
| Viscosity | Kinematic: 42 cSt at 40°C<br>Kinematic 8.18 cSt at 100°C |
| Density | 842.1kg/m^3 at 15°C |

| | |
|---|---|
| Castrol SAE 5W-30 C1 | |
| Pour Point | -36°C |
| Viscosity | Kinematic: 52 cSt at 40°C<br>Kinematic 9.7 cSt at 100°C |
| Density | 850kg/m^3 at 15°C |

| | |
|---|---|
| Castrol Hyspin AWH-M 46 | |
| Pour Point | -42°C |
| Viscosity | Kinematic: 46 cSt at 40°C<br>Kinematic: 8.32 cSt at 100°C |
| Density | 880kg/m^3 at 15°C |

## 14.2 KEYENCE LASER DATA SHEET

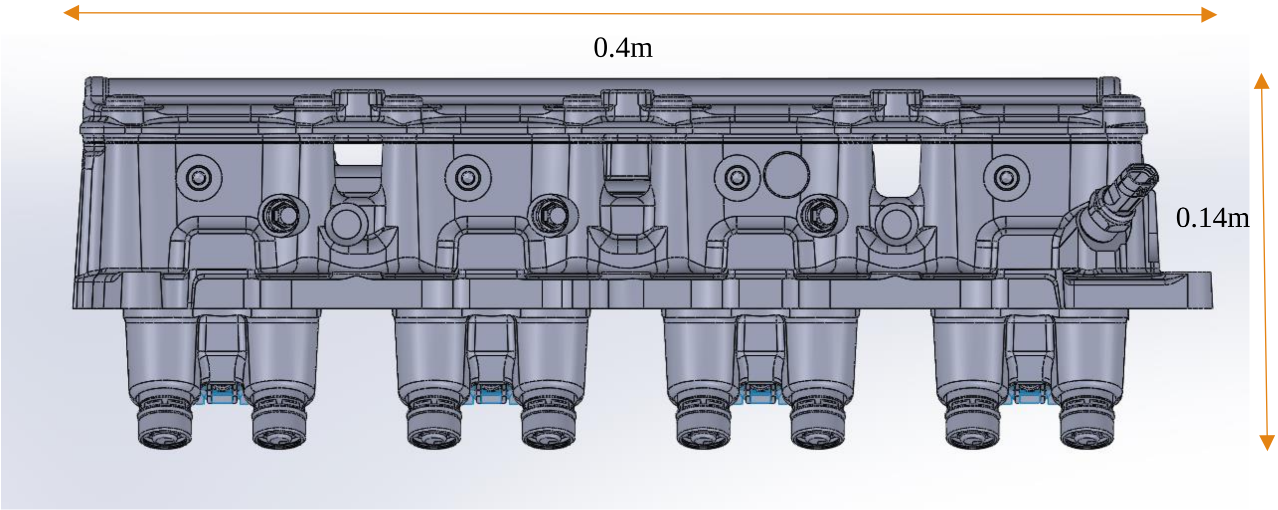
SPECIFICATIONS/SENSOR HEADS

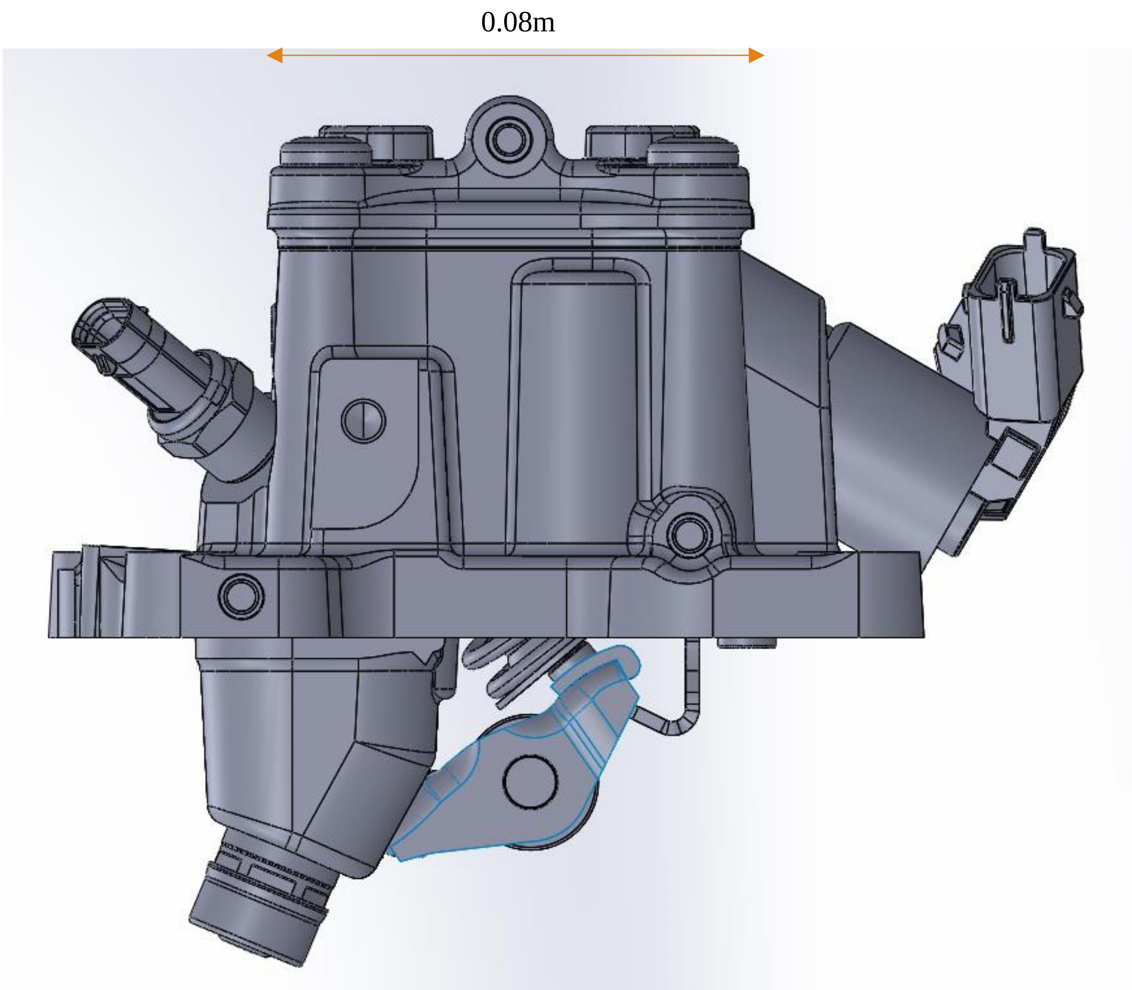
**Coarse target measurement (wide spot type)**

| Model | | LK-H008W | LK-H025 | LK-H027 | LK-H055 | LK-H057 | LK-H085 | LK-H087 | LK-H155 | LK-H157 |
|---|---|---|---|---|---|---|---|---|---|---|
| Mounting mode | | Specular reflection | Diffuse reflection | Diffuse reflection | Diffuse reflection | Diffuse reflection | Diffuse reflection | Diffuse reflection | Diffuse reflection | Diffuse reflection |
| Reference distance | | 8 mm | 20 mm | 20 mm | 50 mm | 50 mm | 80 mm | 80 mm | 150 mm | 150 mm |
| Measurement range*1 | | ±0.5 mm | ±3 mm | ±3 mm | ±10 mm | ±10 mm | ±18 mm | ±18 mm | ±40 mm | ±40 mm |
| Light source | | Red semiconductor laser | | | | | | | | |
| Light source | Wavelength | 655 nm | 650 nm | 650 nm | 650 nm | 650 nm | 655 nm | 650 nm | 655 nm | 650 nm |
| Light source | Laser class IEC 60825-1 | Class 1 | Class 3R | Class 2 | Class 3R | Class 2 | Class 3R | Class 2 | Class 3R | Class 2 |
| Light source | Laser class FDA(CDRH)21CFR Part 1040.10 | Class II | Class IIIa | Class II | Class IIIa | Class II | Class IIIa | Class II | Class IIIa | Class II |
| Light source | Output | 0.3mW | 4.8mW | 0.95mW | 4.8mW | 0.95mW | 4.8mW | 0.95mW | 4.8mW | 0.95mW |
| Spot diameter (at reference distance) | | 20 µm × 550 µm | 25 µm × 1400 µm | | 50 µm × 2000 µm | | 70 µm × 2500 µm | | 120 µm × 4200 µm | |
| Linearity*2 | | ±0.05% of F.S. (F.S.=1 mm) | ±0.02%of F.S. (F.S.= 6 mm) | | ±0.02% of F.S. (F.S.= 20 mm) | | ±0.02% of F.S. (F.S.= 36 mm) | | ±0.02% of F.S. (F.S.= 80 mm) | |
| Repeatability*3 | | 0.005 µm (0.001µm) | 0.02 µm (0.01µm) | | 0.025 µm | | 0.1µm | | 0.25 µm | |
| Sampling cycle | | 2.55/5/10/20/50/100/200/500/1000 µs (9 steps selectable) | | | | | | | | |
| Temperature fluctuation | | 0.02% of F.S./°C (F.S.=1 mm) | 0.01% of F.S./°C (F.S.= 6 mm) | | 0.01% of F.S./°C (F.S.= 20 mm) | | 0.01% of F.S./°C (F.S.= 36 mm) | | 0.01% of F.S./°C (F.S.= 80 mm) | |
| Environment resistance | Enclosure rating | IP67 | | | | | | | | |
| Environment resistance | Ambient light | Incandescent lamp or fluorescent lamp: 10000 lux max. | | | | | | | | Incandescent lamp or fluorescent lamp : 5000 lux max. |
| Environment resistance | Ambient temperature | 0 to +50°C *4 | 0 to +50°C | | | | 0 to +50°C *4 | | | |
| Environment resistance | Relative humidity | 35 to 85%RH (No condensation) | | | | | | | | |
| Environment resistance | Vibration resistance | 10 to 55 Hz, 1.5 mm double amplitude in X, Y, and Z directions, 2 hours respectively | | | | | | | | |
| Material | | Aluminium die-cast | | | | | | | | |
| Weight | | Approx. 240 g | Approx. 230g | | Approx. 260g | | Approx. 280g | | Approx. 300g | |

*1 Measurement range when the sampling cycle is 20 µs or more. *2 This value is obtained when the KEYENCE standard target (White diffuse workpiece or workpiece with a metal mirror surface only for the LK-H008W) is measured in the normal measurement mode.
*3 This value is obtained when the KEYENCE standard target (White diffuse workpiece or workpiece with a metal mirror surface only for the LK-H008W) is measured at the reference distance with the number of averaging measurements set to 16384.
The value in parentheses is a typical example of a measurement with the number of averaging measurements set to 65536 and the sampling cycle to 200 µs. *4 When the ambient temperature rises to 40°C or more, mount this on a metal plate before use.

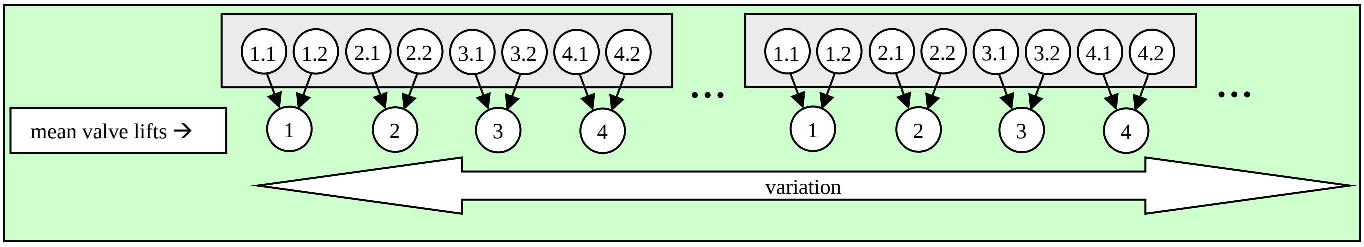
### 14.3 UniAir Dimensions



## 14.4 UNIAIR LIFT TOLERANCES

Variation and repeatability




- Variation is defined as the maximum difference of the mean valve lift of the two valves of one cylinder among all cylinders of all actuators.
- Repeatability is defined as the maximum difference of the mean valve lift of two positions of one cylinder among a certain number of valve lift events, cycle-by-cycle repeatability.

Tolerances from Schaeffler for different lift profiles and speeds at 120°C oil temperature

Full Lift:

| Oil entry | [°C] | 120 | | |
|---|---|---|---|---|
| Engine Speed | [rpm] | 750 | 3000 | 6200 |
| Max. lift height | [mm] | | | |
| | variation | 0,8 | 0,5 | 0,6 |
| | repeatability | 0,2 | 0,2 | 0,2 |

EIVC:

| Oil entry | [°C] | 120 | | | | |
|---|---|---|---|---|---|---|
| Engine Speed | [rpm] | 750 | | 3000 | | 6200 |
| EVC*mech | [°CA] | 450 | 540 | 450 | 540 | 540 |
| Max. lift height | [mm] | | | | | |
| | Variation | 0,8 | 0,8 | 0,6 | 0,6 | 0,6 |
| | repeatability | 0,3 | 0,3 | 0,2 | 0,2 | 0,3 |

LIVO:

| | | | | |
|---|---|---|---|---|
| Oil entry | [°C] | 120 | | |
| Engine Speed | [rpm] | 750 | | 3000 |
| EVC*mech | [°CA] | 420 | 440 | 420 |
| Max. lift height | [mm] | | | |
| | variation | 0,6 | 0,5 | 0,5 |
| | repeatability | 0,2 | 0,2 | 0,3 |

## 14.5 Torque transducer data sheet

### Specifications

| Type | | T40B | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| **Accuracy class** | | 0.1 | 0.05 | | | | | | | |
| **Torque measuring system** | | | | | | | | | | |
| **Nominal (rated) torque $M_{nom}$** | N·m | 50 | 100 | 200 | 500 | | | | | |
| | kN·m | | | | | 1 | 2 | 3 | 5 | 10 |
| **Nominal (rated) rotational speed** | | | | | | | | | | |
| standard speed (option M) | U/min | 20000 | | | | | 15000 | | 12000 | 10000 |
| high speed (option H) | U/min | 24000 | | | 22000 | | 16000 | | 14000 | 12000 |
| **Non-linearity including hysteresis,** related to the nominal (rated) sensitivity | | | | | | | | | | |
| Frequency output | | | | | | | | | | |
| For a max. torque in the range: | | | | | | | | | | |
| between 0% of $M_{nom}$ and 20% of $M_{nom}$ | % | < ±0.01 | | | | | | | | |
| > 20% of $M_{nom}$ and 60% of $M_{nom}$ | % | < ±0.02 | | | | | | | | |
| > 60% of $M_{nom}$ and 100% of $M_{nom}$ | % | < ±0.03 | | | | | | | | |
| Voltage output | | | | | | | | | | |
| For a max. torque in the range: | | | | | | | | | | |
| between 0% of $M_{nom}$ and 20% of $M_{nom}$ | % | < ±0.01 | | | | | | | | |
| > 20% of $M_{nom}$ and 60% of $M_{nom}$ | % | < ±0.02 | | | | | | | | |
| > 60% of $M_{nom}$ and 100% of $M_{nom}$ | % | < ±0.03 | | | | | | | | |
| **Relative standard deviation of the repeatability,** per DIN 1319, related to the variation of the output signal | | | | | | | | | | |
| Frequency output | % | < ± 0.03 | | | | | | | | |
| Voltage output | % | < ± 0.03 | | | | | | | | |
| **Temperature effect per 10 K in the nominal (rated) temperature range** | | | | | | | | | | |
| **on the output signal, related to the actual value of the signal span** | | | | | | | | | | |
| Frequency output | % | ±0.1 | ±0.05 | | | | | | | |
| Voltage output | % | ±0.4 | ±0.2 | | | | | | | |
| **on the zero signal, related to the nominal (rated) sensitivity** | | | | | | | | | | |
| Frequency output | % | ±0.1 | ±0.05 | | | | | | | |
| Voltage output | % | ±0.2 | ±0.1 | | | | | | | |
| **Nominal (rated) sensitivity** (span between torque = zero and nominal (rated) torque) | | | | | | | | | | |
| Frequency output 10 kHz / 60 kHz / 240 kHz | kHz | 5/30/120 | | | | | | | | |
| Voltage output | V | 10 | | | | | | | | |
| **Sensitivity tolerance** (deviation of the actual output quantity at $M_{nom}$ from the nominal (rated) sensitivity) | | | | | | | | | | |
| Frequency output | % | ±0.1 | | | | | | | | |
| Voltage output | % | ±0.1 | | | | | | | | |
| **Output signal at torque = zero** | | | | | | | | | | |
| Frequency output | kHz | 10/60/240 | | | | | | | | |
| Voltage output | V | 0 | | | | | | | | |
| **Nominal (rated) output signal** | | | | | | | | | | |
| Frequency output | | | | | | | | | | |
| with positive nominal (rated) torque | kHz | 15 [1]) / 90 [2]) / 360 [3]) (5 V symmetrical [4])) | | | | | | | | |
| with negative nominal (rated) torque | kHz | 5 [1]) / 30 [2]) / 120 [3]) (5 V symmetrical [4])) | | | | | | | | |
| Voltage output | | | | | | | | | | |
| with positive nominal (rated) torque | V | +10 | | | | | | | | |
| with negative nominal (rated) torque | V | −10 | | | | | | | | |
| **Load resistance** | | | | | | | | | | |
| Frequency output | kΩ | ≥ 2 | | | | | | | | |
| Voltage output | kΩ | ≥ 10 | | | | | | | | |
| **Long-term drift over 48 h at reference temperature** | | | | | | | | | | |
| Frequency output | % | < ±0.06 | < ±0.03 | | | | | | | |
| Voltage output | % | < ±0.06 | < ±0.03 | | | | | | | |
| **Measurement frequency range, −3 dB** | kHz | 1 [1]) / 3 [2]) / 6 [3]) | | | | | | | | |
| **Group delay** | µs | < 400 [1]) / < 220 [2]) / < 150 [3]) | | | | | | | | |
| **Residual ripple** | | | | | | | | | | |
| Voltage output [5]) | mV | < 40 | | | | | | | | |

[1]) Option 5, 10 ± 5 kHz (code SU2)
[2]) Option 5, 60 ± 30 kHz (code DU2)
[3]) Option 5, 240 ± 120 kHz (code HU2)
[4]) RS−422 complementary signals, note termination resistance.
[5]) Signal frequency range 0.1 to 10 kHz

## **Specifications** (continued)

| Nominal (rated) torque $M_{nom}$ | N·m | 50 | 100 | 200 | 500 | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | kN·m | | | | | 1 | 2 | 3 | 5 | 10 |
| **Maximum modulation range** [6] | | | | | | | | | | |
| Frequency output | kHz | 2.5 to 17.5 [1] / 15 to 105 [2] / 60 to 420 [3] | | | | | | | | |
| Voltage output | V | −12 to +12 | | | | | | | | |
| **Energy supply** | | | | | | | | | | |
| Nominal (rated) supply voltage (separated extra-low DC voltage) | V | 18 to 30 | | | | | | | | |
| Current consumption in measuring mode | A | < 1 | | | | | | | | |
| Current consumption in startup mode | A | < 4 (typ. 2) 50 µs | | | | | | | | |
| Nominal (rated) power consumption | W | < 10 | | | | | | | | |
| Maximum cable length | m | 50 | | | | | | | | |
| **Shunt signal** | | approx. 50% of $M_{nom}$ | | | | | | | | |
| **Tolerance of the shunt signal, related to $M_{nom}$** | % | < ± 0.05 | | | | | | | | |
| Nominal (rated) trigger voltage | V | 5 | | | | | | | | |
| Trigger voltage limit | V | 36 | | | | | | | | |
| Shunt signal ON | V | min. >2.5 | | | | | | | | |
| Shunt signal OFF | V | max. <0.7 | | | | | | | | |
| **Rotational speed measuring system** | | | | | | | | | | |
| **Measurement system** | | Magnetic, via AMR sensor (Anisotropic Resistive Effect) and magnetized plastic ring with embedded steel ring | | | | | | | | |
| **Magnetic poles** | | 72 | | | 86 | | 108 | | 126 | 156 |
| **Maximum positional variation of the poles** | | 50 angular seconds | | | | | | | | |
| **Output signal** | V | 5 V symmetrical (RS−422); 2 square wave signals approx. 90° phase shifted | | | | | | | | |
| **Pulses per revolution** | | 1024 | | | | | | | | |
| **Min. rotational speed for sufficient pulse stability** | rpm | 0 | | | | | | | | |
| **Pulse tolerance** [7] | degrees | < ± 0.05 | | | | | | | | |
| **Maximum permissible output frequency** | kHz | 420 | | | | | | | | |
| **Group delay** | µs | <150 | | | | | | | | |
| **Radial nominal (rated) distance between sensor head and magnetic ring** (mechanical distance) | mm | 1.6 | | | | | | | | |
| **Working distance range between sensor head and magnetic ring** | mm | 0.4 to 2.5 | | | | | | | | |
| **Max. permissible axial displacement of the rotor to the stator** [8] | mm | ± 1.5 | | | | | | | | |
| **Hysteresis of direction of rotation reversal in the case of relative vibrations betw. rotor and stator** | | | | | | | | | | |
| Torsional vibration of the rotor | degrees | <approx. 0.2 | | | | | | | | |
| Horizontal stator vibration displacement | mm | <approx. 0.5 | | | | | | | | |
| **Magnetic load limit** | | | | | | | | | | |
| Remanent flux density | mT | >100 | | | | | | | | |
| Coercive field strength | kA/m | >100 | | | | | | | | |
| **Permissible magnetic field strength for signal deviations** | kA/m | <0.1 | | | | | | | | |
| **Load resistance** [9] | kΩ | ≥2 | | | | | | | | |
| **Reference signal measuring system (0 index)** | | | | | | | | | | |
| **Measurement system** | | Magnetic, with Hall sensor and magnet | | | | | | | | |
| **Output signal** | V | 5 V symmetrical (RS-422) | | | | | | | | |
| **Pulses per revolution** | | 1 | | | | | | | | |
| **Min. rotational speed for sufficient pulse stability** | rpm | 2 | | | | | | | | |
| **Pulse width, approx.** | degrees | 0.088 | | | | | | | | |
| **Pulse tolerance** [7] | degrees | < ± 0.05 | | | | | | | | |
| **Group delay** | µs | <150 | | | | | | | | |
| **Axial nominal (rated) distance between sensor head and magnetic ring** (mechanical distance) | mm | 2.0 | | | | | | | | |
| **Working distance range between sensor head and magnetic ring** | mm | 0.4 to 2.5 | | | | | | | | |
| **Max. permissible axial displacement of rotor to stator** [8] | mm | ± 1.5 | | | | | | | | |

[6] Output signal range in which there is a repeatable correlation between torque and output signal.
[7] At nominal (rated) conditions.
[8] The data refers only to a central axial alignment. Deviations lead to a change in pulse tolerance.
[9] Note the termination resistances as per RS−422.

## Specifications (continued)

| Nominal (rated) torque $M_{nom}$ | N·m | 50 | 100 | 200 | 500 | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | kN·m | | | | | 1 | 2 | 3 | 5 | 10 |
| **General information** | | | | | | | | | | |
| **EMC** | | | | | | | | | | |
| **Emission** (per FCC 47, Part 15, subpart C) [10] | – | | | | | | | | | |
| **Emission** (per EN 61326–1, Section 7) | | | | | | | | | | |
| RFI field strength) [11] | – | Class B | | | | | | | | |
| **Immunity from interference** (EN 61326–1, Table 2) | | | | | | | | | | |
| Electromagnetic field (AM) | V/m | 10 | | | | | | | | |
| Magnetic field | A/m | 100 | | | | | | | | |
| Electrostatic discharge (ESD) | | | | | | | | | | |
| Contact discharge | kV | 4 | | | | | | | | |
| Air discharge | kV | 8 | | | | | | | | |
| Fast transients (burst) | kV | 1 | | | | | | | | |
| Impulse voltages (surge) | kV | 1 | | | | | | | | |
| Conducted interference (AM) | V | 10 | | | | | | | | |
| **Degree of protection per EN 60529** | | IP 54 | | | | | | | | |
| **Reference temperature** | °C | 23 | | | | | | | | |
| **Nominal (rated) temperature range** | °C | +10 to +70 | | | | | | | | |
| **Operating temperature range** [12] | °C | −20 to +85 | | | | | | | | |
| **Storage temperature range** | °C | −40 to +85 | | | | | | | | |
| **Mechanical shock per EN 60068–2–27** [13] | | | | | | | | | | |
| Number | n | 1000 | | | | | | | | |
| Duration | ms | 3 | | | | | | | | |
| Acceleration (half sine) | $m/s^2$ | 650 | | | | | | | | |
| **Vibrational stress in 3 directions per EN 60068–2–6** [13] | | | | | | | | | | |
| Frequency range | Hz | 10 to 2000 | | | | | | | | |
| Duration | h | 2.5 | | | | | | | | |
| Acceleration (amplitude) | $m/s^2$ | 200 | | | | | | | | |
| **Load limits** [14] | | | | | | | | | | |
| **Limit torque, related to $M_{nom}$** [15] | % | 400 | 200 | | | | | 160 | | |
| **Breaking torque, related to $M_{nom}$** [15] | % | 800 | > 400 | | | | | > 320 | | |
| **Longitudinal limit force** [16] | kN | 5 | 5 | 10 | 13 | 19 | 30 | 35 | 60 | 80 |
| **Lateral limit force** [16] | kN | 1 | 1 | 2 | 4 | 5 | 9 | 10 | 12 | 18 |
| **Limit bending moment** [16] | N·m | 50 | 50 | 100 | 200 | 220 | 560 | 600 | 800 | 1200 |
| **Oscillation width per DIN 50100 (peak-to-peak)** [17] | N·m | 200 | 200 | 400 | 1000 | 2000 | 4000 | 4800 | 8000 | 16000 |

[10] Option 7, Code U

[11] Option 7, Code S

[12] Heat conductance via the stator base plate necessary over 70°C. The temperature of the base plate must not exceed 85°C.

[13] The antenna ring and connector plug must be fixed.

[14] Each type of irregular stress (bending moment, lateral or longitudinal force, exceeding nominal (rated) torque), can only be permitted up to its specified load limit, provided none of the others can occur at the same time. If this condition is not met, the limit values must be reduced. If 30% of the limit bending moment and lateral limit force occur at the same time, only 40% of the longitudinal limit force is permissible and the nominal (rated) torque must not be exceeded. The effects of permissible bending moments, longitudinal and lateral forces on the measurement result are $\leq \pm 0.3\%$ (50 Nm: $\leq \pm 0.6\%$) of the nominal (rated) torque. The load limits only apply for the nominal (rated) temperature range. At temperatures $< 10$°C, load limits are expected to reduce by up to 30%, because the strength reduction increases as the temperatures fall.

[15] With a static loading.

[16] Static and dynamic.

[17] The nominal (rated) torque must not be exceeded.

## 14.6 K TYPE THERMOCOUPLE DATA SHEET

| Thermocouple base materials acc to IEC 584 | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Type | Conductors | Temperature range | µV | Tolerance acc IEC 584-2 (reference junction at 0 °C) | | | | | |
| | | | | Class 1 | | Class 2 | | Class 3 | |
| K | + Chromel (NiCr)<br>- Alumel (NiAl) | -200 °C / + 1200 °C | -5891 to +48828 | -40 °C / + 1000 °C | ± 1,5 °C<br>or ± 0,004 ./t/ | -40 °C / + 1200 °C | ± 2,5 °C<br>or ± 0,0075 ./t/ | -200 °C / + 40 °C | ± 2,5 °C<br>or ± 0,015 ./t/ |

## 14.7 APPENDIX 7

| Technical data | PR 130 / PR 140 / PR 150 / PR 190 | HT-PDX-20 |
|---|---|---|
| Overload range | 1.5 x nominal pressure | |
| Burst pressure | 3 x nominal pressure | |
| Signal type | Two wire analog (4 ... 20 mA), three wire analog (0 ... 10 VDC) | |
| Supply voltage Ub | | |
| ... at 4 ... 20 mA | 10 ... 30 VDC | 30 V |
| ... at 0 ... 10 VDC | 12 ... 32 VDC | – |
| Current consumption | 6.5 mA | 50 mA |
| Overvoltage protection | 32 VDC | |
| Error limit (of final value) | comprises the influences non-linearity, hysterese, repeatability, zero-point- and span error | |
| ... at +22 °C (room temperature) | ± 0.5 % | |
| ... at -15 ... +85°C | < ± 1.0 % | |
| ... at +85 ... +100°C | < ± 2.5 % | |
| ... at -40 ... -15°C | < ± 2.5 % | |
| Temperature error (-20 ... +80 °C) | < ± 0,03 %/°C | |
| Compensation temperature range | -40 ... +100 °C | |
| Non-linearity | < ± 0.4 % of final value | |
| Reproducability | < ± 0.1 % of final value | |
| Hysterese | < ± 0.1 % of final value | |
| Long-term stability | < ± 0.1 % of final value/year | |
| Response time | < = 1 ms (10 ... 90 %) | |
| Frequency range | < = 1 kHz | |
| Isolation resistance | min. 100 MOhm | |
| Total resistance | $R_G$ = (Ub -10 V) / 20 mA (at output signal 4 ... 20 mA) | |
| Load resistance | $R_L$ = > 5 kOhm (at output signal 0 ... 10 VDC) | |
| Number of load cycles | > 10 x $10^6$ | |
| Medium temperature | -40 ... +125 °C | |
| Environmental temperature | -40 ... +105 °C (short term +125 °C) | -40 ... +85 °C |
| Storage temperature | -40 ... +125 °C | |
| EMV test | EN 50081-2, EN 50082-2 | |
| Vibrational stability | 5 mm 10 ... 32 Hz, 20 g 32 ... 500 Hz, DIN EN 60068-2-6 | |

## 14.8 TESTING PROCEDURE

Testing procedure

Pre-Test conditions

1. Set oil bath temperature to required testing temperature.

   *Note – Dependent on required oil temperature and testing speed, oil bath temperature may not be needed as internal heating from UniAir will cover the testing temperature. For example, 3000rpm motor speed will generate internal heating on UniAir up to 90 °C over a 1-hour testing time. A combination of oil bath temperature and motor speed can be used to achieve the required temperature (see below).*

2. Set lift mode on JLR laptop (see lift mode specification sheet)

3. Generate Test ID:

Example:

FL1000rpm3bar35C2000SRC1L1L2C3L3L4 T1-T4UniAir T5T6OilBath T8Chamber T9UniAirExit T10OilFilter 10.10.19

- Lift mode (Full lift – FL, No Lift – NL, Early Intake Valve Closing - EIVC, Late Intake Valve Opening - LIVO)
- Motor speed (1000rpm)
- Oil Pressure (3bar)
- Oil testing temperature (35C)
- Sample rate (1000 sample rate at 500rpm to 6000 sample rate at 3000rpm) (2000SR)
- Laser positions (Cylinder 1 C1– Laser 1 L1 and Laser 2 L2)
- Thermocouple positions (Thermocouple name - T1, Thermocouple position UniAir)
- Date of testing (10.10.19)

4. Open virtual interface (VI) – Set
   - Test ID
   - Sample rate
   - Oil pressure
   - Oil pressure tolerance (5%) of set pressure

Testing Procedure

1. Enable pump at set pressure via VI. Wait until pressure is at +-5% of set pressure. Estimated time is 20 seconds across all testing parameters. If pressure is not achieved in this time frame, check for leaks.

2. Set and enable motor once pressure stabilized, ensure final motor speed is +-10rpm of set speed. Motor speed should be achieved in 10 seconds at maximum speed.

3. For valid measurements, measurements can only be used after 2 seconds when motor speed is changed. This will ensure stabilized motor speed and torque.

4. Wait until average oil temperature across the UniAir is within +-1 °C of required temperature. I.e. all UniAir thermocouples have surpassed set temperature range.

*Note – If doing a range of temperature tests, start with cold oil temperatures and higher speeds first, this will significantly reduce waiting times.*

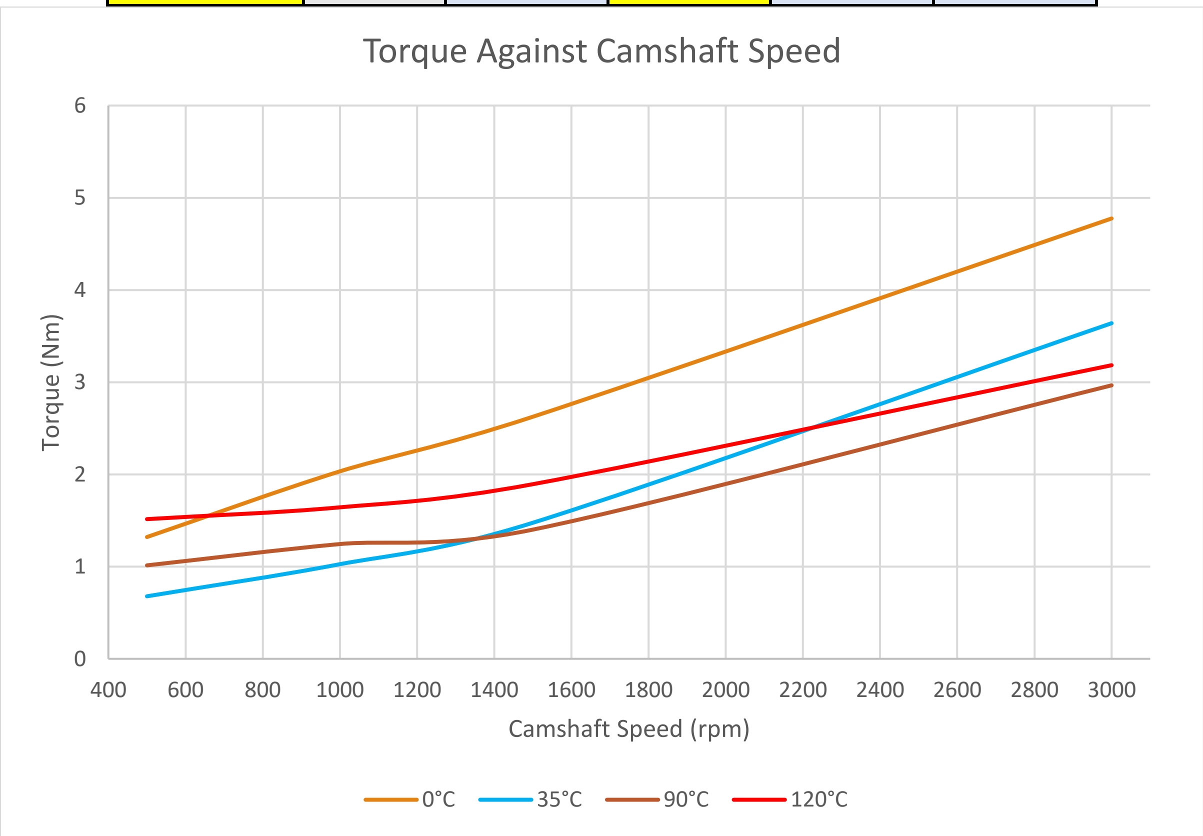
5. For valid measurement, the average oil temperature across the UniAir must be within required temperature range +-1 °C for 15 seconds or more.

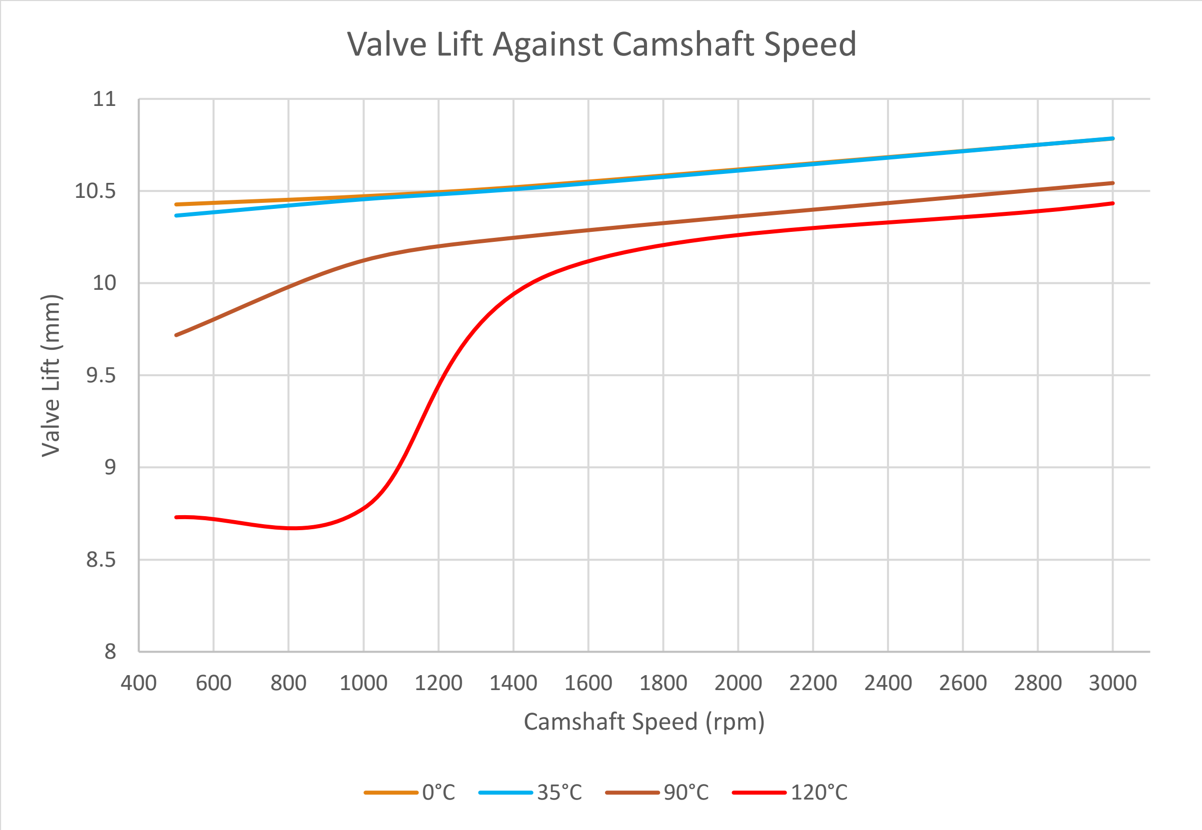
*Note – If temperature cannot be achieved after motor speed and pressure stabilized either:*

- *If too hot - wait for rig to cool down further before beginning test*
- *If too cold –*
    1. *increase oil bath temperature (Use this method when already close to achieving required temperature)*
    2. *increase motor speed to create internal UniAir heating, then reduce back to set motor speed. *REMEMBER* Measurements are only valid after 2 seconds after a motor speed change, and the measurement window must be 15 seconds or more in the set average temperature range across the UniAir*

## 14.9 5W30 Full Lift results

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 403.93 | 63.69 | 11.87 | 6.62 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 1.32 | 0.68 | 1.01 | 1.51 |
| | 1000 | 2.03 | 1.03 | 1.24 | 1.64 |
| | 1500 | 2.63 | 1.48 | 1.40 | 1.89 |
| | 3000 | 4.78 | 3.64 | 2.97 | 3.18 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 10.43 | 10.37 | 9.72 | 8.73 |
| | 1000 | 10.47 | 10.46 | 10.12 | 8.78 |
| | 1500 | 10.54 | 10.53 | 10.27 | 10.05 |
| | 3000 | 10.78 | 10.79 | 10.54 | 10.43 |



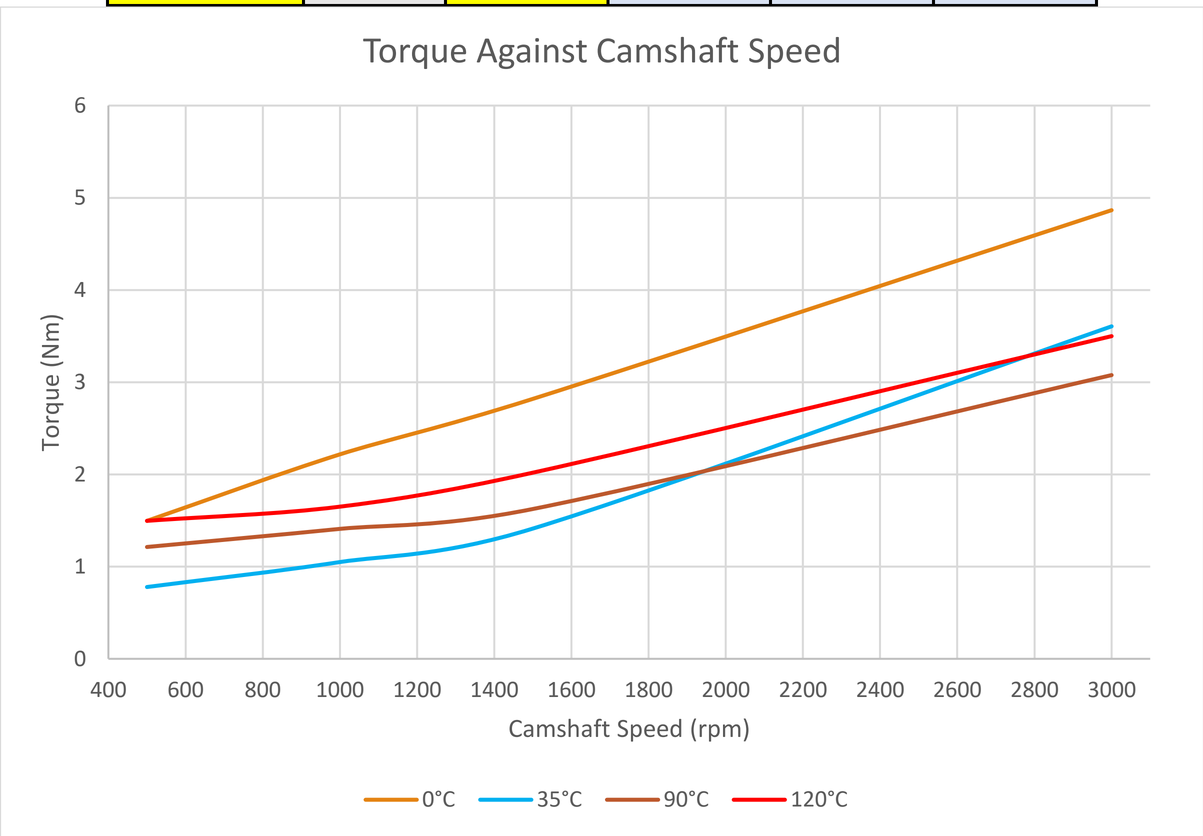


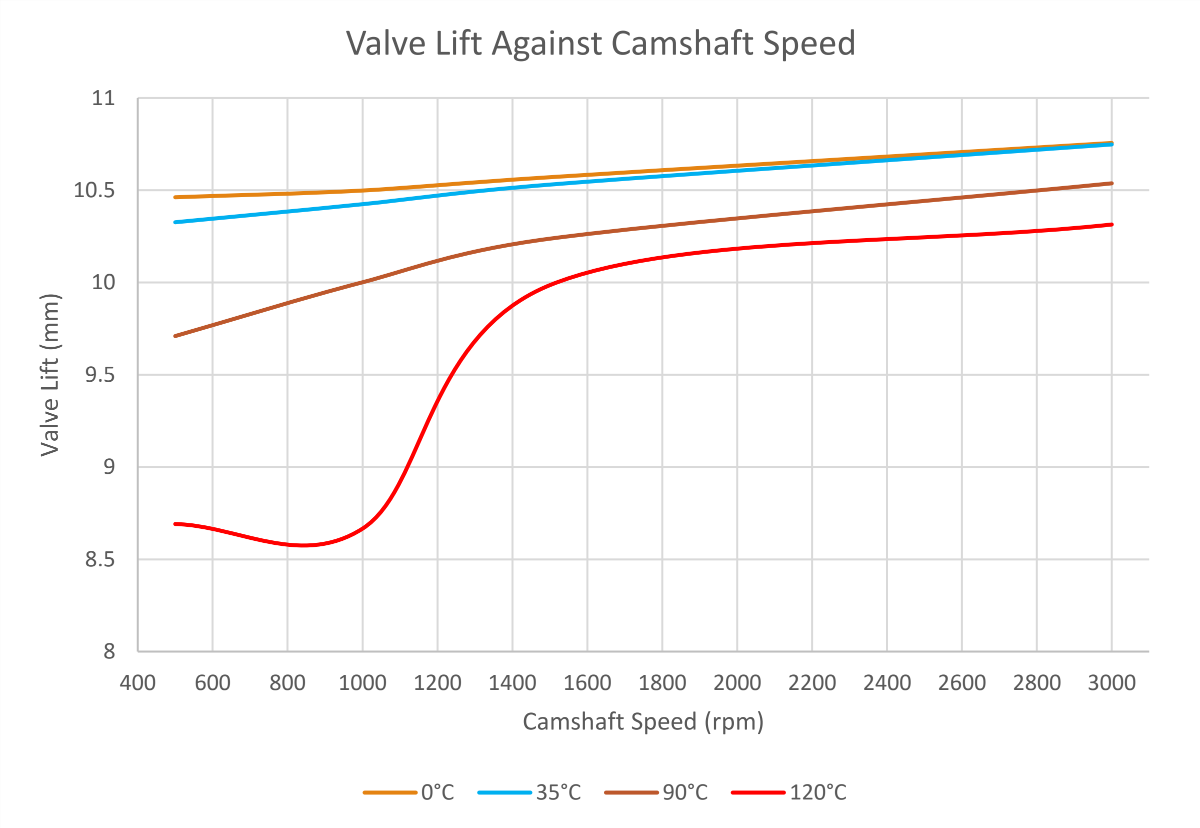

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Overall Efficiency (%) |
|---|---|---|---|---|
| 500 | 403.925 | 0.51 | 1.00 | 51.35 |
| | 63.686 | 1.00 | 0.99 | 99.42 |
| | 11.87 | 0.67 | 0.93 | 62.35 |
| | 6.615 | 0.45 | 0.84 | 37.48 |
| | | | | |
| 1000 | 403.925 | 0.50 | 1.00 | 50.48 |
| | 63.686 | 1.00 | 1.00 | 99.84 |
| | 11.87 | 0.82 | 0.97 | 79.71 |
| | 6.615 | 0.62 | 0.84 | 52.36 |
| | | | | |
| 1500 | 403.925 | 0.53 | 1.00 | 53.36 |
| | 63.686 | 0.95 | 1.00 | 94.87 |
| | 11.87 | 1.00 | 0.97 | 97.45 |
| | 6.615 | 0.74 | 0.95 | 70.57 |
| | | | | |
| 3000 | 403.925 | 0.62 | 1.00 | 62.10 |
| | 63.686 | 0.81 | 1.00 | 81.49 |
| | 11.87 | 1.00 | 0.98 | 97.75 |
| | 6.615 | 0.93 | 0.97 | 90.11 |

| Overall efficiency (%) | Viscosity (cSt) | | | |
|---|---|---|---|---|
| Camshaft Speed (rpm) | 403.925 | 63.686 | 11.87 | 6.615 |
| 500 | 51.35 | 99.42 | 62.35 | 37.48 |
| 1000 | 50.48 | 99.84 | 79.71 | 52.36 |
| 1500 | 53.36 | 94.87 | 97.45 | 70.57 |
| 3000 | 62.10 | 81.49 | 97.75 | 90.11 |
| Averaged Overall Efficiency (%) | 54.32 | 93.91 | 84.32 | 62.63 |

## 14.10 HYSPIN FULL LIFT RESULTS

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 399.575 | 56.703 | 10.055 | 5.568 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 1.50 | 0.78 | 1.21 | 1.50 |
| | 1000 | 2.22 | 1.05 | 1.41 | 1.65 |
| | 1500 | 2.82 | 1.41 | 1.62 | 2.02 |
| | 3000 | 4.87 | 3.61 | 3.08 | 3.50 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 10.46 | 10.33 | 9.71 | 8.69 |
| | 1000 | 10.50 | 10.42 | 10.00 | 8.67 |
| | 1500 | 10.57 | 10.53 | 10.24 | 9.99 |
| | 3000 | 10.76 | 10.75 | 10.54 | 10.31 |






| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Overall Efficiency (%) |
|---|---|---|---|---|
| 500 | 399.575 | 0.52 | 1.00 | 52.04 |
| | 56.703 | 1.00 | 0.99 | 98.70 |
| | 10.055 | 0.64 | 0.93 | 59.58 |
| | 5.568 | 0.52 | 0.83 | 43.23 |
| | | | | |
| 1000 | 399.575 | 0.47 | 1.00 | 47.32 |
| | 56.703 | 1.00 | 0.99 | 99.29 |
| | 10.055 | 0.74 | 0.95 | 70.94 |
| | 5.568 | 0.64 | 0.83 | 52.49 |
| | | | | |
| 1500 | 399.575 | 0.50 | 1.00 | 50.10 |
| | 56.703 | 1.00 | 1.00 | 99.62 |
| | 10.055 | 0.87 | 0.97 | 84.18 |
| | 5.568 | 0.70 | 0.94 | 66.09 |
| | | | | |
| 3000 | 399.575 | 0.63 | 1.00 | 63.24 |
| | 56.703 | 0.85 | 1.00 | 85.27 |
| | 10.055 | 1.00 | 0.98 | 97.97 |
| | 5.568 | 0.88 | 0.96 | 84.32 |

| Overall efficiency (%) | Viscosity (cSt) | | | |
|---|---|---|---|---|
| Camshaft Speed (rpm) | 399.575 | 56.703 | 10.055 | 5.568 |
| 500 | 52.04 | 98.70 | 59.58 | 43.23 |
| 1000 | 47.32 | 99.29 | 70.94 | 52.49 |
| 1500 | 50.10 | 99.62 | 84.18 | 66.09 |
| 3000 | 63.24 | 85.27 | 97.97 | 84.32 |
| Averaged Overall Efficiency (%) | 53.17 | 95.72 | 78.17 | 61.53 |

## 14.11 0W20 No Lift results

| Torque (Nm) | | Viscosity (cSt) | | | |
|---|---|---|---|---|---|
| | | 313.305 | 50.541 | 9.783 | 5.541 |
| Camshaft speeds (rpm) | 500 | 0.902677 | 0.260667 | 0.166 | 0.369 |
| | 1000 | 1.385319 | 0.701 | 0.6233333 | 0.806075 |
| | 1500 | 1.87087 | 1.086667 | 0.929 | 1.26 |
| | 3000 | 3.6 | 2.6675 | 2.2666667 | 2.4875 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency (%) |
|---|---|---|
| 500 | 313.305 | 18.38974 |
| | 50.541 | 63.68286 |
| | 9.783 | 100 |
| | 5.541 | 44.98645 |
| | | |
| 1000 | 313.305 | 44.99566 |
| | 50.541 | 88.92059 |
| | 9.783 | 100 |
| | 5.541 | 77.32945 |
| | | |
| 1500 | 313.305 | 49.65603 |
| | 50.541 | 85.4908 |
| | 9.783 | 100 |
| | 5.541 | 73.73016 |
| | | |
| 3000 | 313.305 | 62.96296 |
| | 50.541 | 84.97345 |
| | 9.783 | 100 |
| | 5.541 | 91.12228 |

| Mechanical efficiency (%) | Viscosity (cSt) | | | |
|---|---|---|---|---|
| Camshaft Speed (rpm) | 313.31 | 50.54 | 9.78 | 5.54 |
| 500 | 18.39 | 63.68 | 100.00 | 44.99 |
| 1000 | 45.00 | 88.92 | 100.00 | 77.33 |
| 1500 | 49.66 | 85.49 | 100.00 | 73.73 |
| 3000 | 62.96 | 84.97 | 100.00 | 91.12 |
| Averaged Overall Efficiency (%) | 44.00 | 80.77 | 100.00 | 71.79 |

## 14.12 5W30 No Lift results

| Torque (Nm) | | Viscosity (cSt) | | | |
|---|---|---|---|---|---|
| | | 403.925 | 63.686 | 11.87 | 6.615 |
| Camshaft speeds (rpm) | 500 | 1.051724 | 0.273703 | 0.164727 | 0.385285 |
| | 1000 | 1.595222 | 0.764939 | 0.645268 | 0.860619 |
| | 1500 | 2.055219 | 1.143903 | 0.897069 | 1.196719 |
| | 3000 | 3.710158 | 2.721603 | 2.228122 | 2.430744 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency (%) |
|---|---|---|
| 500 | 403.925 | 15.66257 |
| | 63.686 | 60.18462 |
| | 11.87 | 100 |
| | 6.615 | 42.75457 |
| | | |
| 1000 | 403.925 | 40.45006 |
| | 63.686 | 84.35553 |
| | 11.87 | 100 |
| | 6.615 | 74.97716 |
| | | |
| 1500 | 403.925 | 43.64834 |
| | 63.686 | 78.42178 |
| | 11.87 | 100 |
| | 6.615 | 74.96068 |
| | | |
| 3000 | 403.925 | 60.05465 |
| | 63.686 | 81.86801 |
| | 11.87 | 100 |
| | 6.615 | 91.66419 |

| Mechanical efficiency (%) | Viscosity (cSt) | | | |
|---|---|---|---|---|
| Camshaft Speed (rpm) | 403.925 | 63.686 | 11.87 | 6.615 |
| 500 | 15.66 | 60.18 | 100.00 | 42.75 |
| 1000 | 40.45 | 84.36 | 100.00 | 74.98 |
| 1500 | 43.65 | 78.42 | 100.00 | 74.96 |
| 3000 | 60.05 | 81.87 | 100.00 | 91.66 |
| Averaged Overall Efficiency (%) | 39.95 | 76.21 | 100.00 | 71.09 |

## 14.13 HYSPIN NO LIFT RESULTS

| Torque (Nm) | | Viscosity (cSt) | | | |
|---|---|---|---|---|---|
| | | 399.575 | 56.703 | 10.055 | 5.568 |
| Camshaft speeds (rpm) | 500 | 1.141074 | 0.316657 | 0.283131 | 0.3744 |
| | 1000 | 1.683932 | 0.794657 | 0.769965 | 0.862431 |
| | 1500 | 2.148088 | 1.136264 | 1.068119 | 1.263974 |
| | 3000 | 3.929325 | 2.739144 | 2.437453 | 2.599253 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency (%) |
|---|---|---|
| 500 | 399.575 | 24.81264 |
| | 56.703 | 100 |
| | 10.055 | 89.41245 |
| | 5.568 | 75.62256 |
| | | |
| 1000 | 399.575 | 45.7242 |
| | 56.703 | 96.89273 |
| | 10.055 | 100 |
| | 5.568 | 89.27842 |
| | | |
| 1500 | 399.575 | 49.72418 |
| | 56.703 | 94.00272 |
| | 10.055 | 100 |
| | 5.568 | 84.50482 |
| | | |
| 3000 | 399.575 | 62.03236 |
| | 56.703 | 88.98595 |
| | 10.055 | 100 |
| | 5.568 | 93.77514 |

| Mechanical efficiency (%) | Viscosity (cSt) | | | |
|---|---|---|---|---|
| Camshaft Speed (rpm) | 399.575 | 56.703 | 10.055 | 5.568 |
| 500 | 24.81264 | 89.41245 | 100 | 75.62256 |
| 1000 | 45.7242 | 96.89273 | 100 | 89.27842 |
| 1500 | 49.72418 | 94.00272 | 100 | 84.50482 |
| 3000 | 62.03236 | 88.98595 | 100 | 93.77514 |
| Averaged Overall Efficiency (%) | 45.57335 | 94.97035 | 97.35311 | 85.79523 |

## 14.14 5W30 EIVC RESULTS

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 403.93 | 63.69 | 11.87 | 6.62 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 4.57 | 3.80 | 3.58 | 3.89 |
| | 1000 | 4.93 | 4.09 | 3.94 | 4.11 |
| | 1500 | 4.52 | 3.77 | 3.53 | 3.88 |
| | 3000 | 4.60 | 3.22 | 2.83 | 3.02 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 10.39 | 10.41 | 9.81 | 9.40 |
| | 1000 | 9.32 | 9.71 | 9.59 | 9.38 |
| | 1500 | 6.89 | 7.62 | 7.95 | 8.11 |
| | 3000 | 2.16 | 2.05 | 2.65 | 2.21 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Overall Efficiency (%) |
|---|---|---|---|---|
| 500 | 403.925 | 0.78 | 1.00 | 78.04 |
| | 63.686 | 0.94 | 1.00 | 94.07 |
| | 11.87 | 1.00 | 0.94 | 94.22 |
| | 6.615 | 0.92 | 0.90 | 83.07 |
| 1000 | 403.925 | 0.80 | 0.96 | 76.72 |
| | 63.686 | 0.96 | 1.00 | 96.19 |
| | 11.87 | 1.00 | 0.99 | 98.70 |
| | 6.615 | 0.96 | 0.97 | 92.62 |
| 1500 | 403.925 | 0.78 | 0.85 | 66.34 |
| | 63.686 | 0.94 | 0.94 | 87.94 |
| | 11.87 | 1.00 | 0.98 | 98.11 |
| | 6.615 | 0.91 | 1.00 | 91.07 |
| 3000 | 403.925 | 0.61 | 0.82 | 50.24 |
| | 63.686 | 0.88 | 0.78 | 68.02 |
| | 11.87 | 1.00 | 1.00 | 100.00 |
| | 6.615 | 0.94 | 0.83 | 78.19 |

## 14.15 HYSPIN EIVC RESULTS

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
| --- | --- | --- | --- | --- | --- |
| Viscosity (cSt) | | 399.575 | 56.703 | 10.055 | 5.568 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 4.80 | 3.93 | 3.73 | 3.86 |
| | 1000 | 4.86 | 4.23 | 4.07 | 4.14 |
| | 1500 | 4.43 | 3.92 | 3.72 | 3.90 |
| | 3000 | 4.37 | 3.41 | 3.04 | 3.26 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 10.42 | 10.32 | 9.75 | 9.28 |
| | 1000 | 9.30 | 9.67 | 9.59 | 9.33 |
| | 1500 | 6.95 | 7.58 | 7.96 | 7.98 |
| | 3000 | 2.28 | 2.07 | 2.55 | 2.19 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Overall Efficiency (%) |
| --- | --- | --- | --- | --- |
| 500 | 399.575 | 0.78 | 1.00 | 77.83 |
| | 56.703 | 0.95 | 0.99 | 94.12 |
| | 10.055 | 1.00 | 0.94 | 93.61 |
| | 5.568 | 0.97 | 0.89 | 86.25 |
| | | | | |
| 1000 | 399.575 | 0.84 | 0.96 | 80.54 |
| | 56.703 | 0.96 | 1.00 | 96.33 |
| | 10.055 | 1.00 | 0.99 | 99.17 |
| | 5.568 | 0.98 | 0.96 | 94.89 |
| | | | | |
| 1500 | 399.575 | 0.84 | 0.87 | 73.19 |
| | 56.703 | 0.95 | 0.95 | 90.24 |
| | 10.055 | 1.00 | 1.00 | 99.64 |
| | 5.568 | 0.95 | 1.00 | 95.44 |
| | | | | |
| 3000 | 399.575 | 0.70 | 0.89 | 62.22 |
| | 56.703 | 0.89 | 0.81 | 72.33 |
| | 10.055 | 1.00 | 1.00 | 100.00 |
| | 5.568 | 0.93 | 0.86 | 79.86 |

## 5W30 LIVO RESULTS

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 403.93 | 63.69 | 11.87 | 6.62 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 1.37 | 0.68 | 0.76 | 1.23 |
| | 1000 | 2.12 | 1.24 | 1.23 | 1.51 |
| | 1500 | 2.87 | 1.87 | 1.67 | 1.98 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 6.60 | 6.59 | 6.52 | 6.36 |
| | 1000 | 6.91 | 6.89 | 6.80 | 6.73 |
| | 1500 | 7.11 | 7.11 | 7.03 | 7.00 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Overall Efficiency (%) |
|---|---|---|---|---|
| 500 | 403.925 | 0.50 | 1.00 | 49.89 |
| | 63.686 | 1.00 | 1.00 | 99.87 |
| | 11.87 | 0.89 | 0.99 | 88.26 |
| | 6.615 | 0.56 | 0.96 | 53.70 |
| | | | | |
| 1000 | 403.925 | 0.58 | 1.00 | 58.06 |
| | 63.686 | 0.99 | 1.00 | 98.84 |
| | 11.87 | 1.00 | 0.98 | 98.33 |
| | 6.615 | 0.81 | 0.97 | 79.21 |
| | | | | |
| 1500 | 403.925 | 0.58 | 1.00 | 58.12 |
| | 63.686 | 0.89 | 1.00 | 89.07 |
| | 11.87 | 1.00 | 0.99 | 98.88 |
| | 6.615 | 0.84 | 0.98 | 83.07 |

## 14.16 HYSPIN LIVO RESULTS

| Oil temperature (°C) | | 0 | 35 | 90 | 120 |
|---|---|---|---|---|---|
| Viscosity (cSt) | | 399.575 | 56.703 | 10.055 | 5.568 |
| Torque (Nm) | Camshaft Speed (rpm) | | | | |
| Measured Baseline torque value | 500 | 1.47 | 0.69 | 0.94 | 1.17 |
| | 1000 | 2.31 | 1.27 | 1.40 | 1.54 |
| | 1500 | 3.07 | 1.95 | 1.87 | 2.10 |
| Valve Lift (mm) | | | | | |
| Measured Baseline valve lift value | 500 | 6.56 | 6.57 | 6.52 | 6.32 |
| | 1000 | 6.80 | 6.85 | 6.80 | 6.72 |
| | 1500 | 6.96 | 7.06 | 6.98 | 6.94 |

| Camshaft Speed (rpm) | Viscosity (cSt) | Mechanical Efficiency | Volumetric Efficiency | Overall Efficiency (%) |
|---|---|---|---|---|
| 500 | 399.575 | 0.47 | 1.00 | 46.85 |
| | 56.703 | 1.00 | 1.00 | 100.00 |
| | 10.055 | 0.73 | 0.99 | 72.96 |
| | 5.568 | 0.59 | 0.96 | 56.69 |
| | | | | |
| 1000 | 399.575 | 0.55 | 0.99 | 54.32 |
| | 56.703 | 1.00 | 1.00 | 100.00 |
| | 10.055 | 0.90 | 0.99 | 89.75 |
| | 5.568 | 0.82 | 0.98 | 80.61 |
| | | | | |
| 1500 | 399.575 | 0.61 | 0.99 | 59.97 |
| | 56.703 | 0.95 | 1.00 | 95.48 |
| | 10.055 | 1.00 | 0.99 | 98.82 |
| | 5.568 | 0.89 | 0.98 | 87.27 |