

School of Engineering

Valve-Train Investigation and Comparative Analysis of Two High-Performance Single Cylinder Engines

2014 KTM Moto3 RC250R

2017 Yamaha YZ250F (GYTR Cylinder Head)

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Declaration of Authenticity

This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree

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Date: 20th April 2020

Abstract

This paper consists of an investigation and comparative analysis of the valve trains of two different high-speed, single cylinder, 250CC, naturally aspirated engines, which are used within different areas of Motorsport. The first being from a 2014 KTM Moto3 RC250R which competed in the 2019 BSB MotoStar championship and the second coming from a GYTR cylinder head from a 2017 Yamaha YZ250F competing in the FIA World Enduro Championship.

Foremost, a detailed literature review consisting of an in-depth investigation of the valve-train systems within an internal combustion engine was carried out. Next, with the use of Microsoft Excel, calculations were made to determine the characteristics of both valve trains. Then, a friction model of the valve-trains was completed and compared to one another. Finally, both valve-trains were modelled with the use of the SolidWorks software package and studied with a FEA simulation.

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1 Nomenclature

mm & m- Millimetres & Metres

mm/s & m/s – Millimetres Per Second & Metres Per Second

mm/s² & m/s²- Millimetres Per Second Squared & Metres Per Second Squared

mm/° & **m/°** - millimetres per degree & metres per degree

mm/°/° & **m/°/°** - millimetres per degree per degree & metres per degree per degree

g & **kg** – Grams & Kilograms

CC – Cubic Centre Metres

RPM – Revolutions Per Minute

Rads/s – Radians Per Second

Rads/s² - Radians Per Second Squared

N – Newtons

N/mm & **N/m** – Newton Metres & Newton Millimetres

J – Joules

W - Watts

m – Mass

m - Velocity

μ – Coefficient of Friction

a – Acceleration

F_c – Contact Force

F_f – Friction Force

τ_f – Friction Torque

W.D – Work Done

P – Power

2 Acronyms

ICE – Internal Combustion Engine

TDC – Top Dead Centre

BDC – Bottom Dead Centre

OHV – Over Head Valve

DOHC & OHC – Double Over Head Cam & Over Head Cam

CAE – Computer-Aided Engineering

CAD – Computer-Aided Design

FEA – Finite Element Analysis

2D & 3D – Two Dimensional & Three Dimensional

FIM - Fédération Internationale de Motocyclisme

FIA - Fédération Internationale de l'Automobile

BSB – British Superbike Championship

WEC – World Enduro Championship

GYTR – Genuine Yamaha Technology Racing

N/A – Naturally Aspirated

HL – Hydrodynamic Lubrication

EHL – Elastohydrodynamic Lubrication

ML – Mixed Lubrication

BL – Boundary Lubrication

DLC – Diamond Like Carbon

PVD – Physical Vapor Deposition

TiN – Titanium Nitride

CrN – Chromium Nitride.

HP - Horsepower

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7 Introduction

The ICE (*Internal Combustion Engine*) is a complex system, with an array of sub-systems working simultaneously. One of these sub-systems and the topic of this study is the valve-train, the design of which *“directly influences engine performance, exhaust emission, reliability, vibration and noise”* (Gao Wenzhi, 2004). The terminology valve-train as applied to an ICE refers to the mechanism for controlling the intake and exhaust gas flows through the cylinder head ports and combustion chamber. This mechanism is required in reciprocating ICE’s operating on the four-stroke cycle. The individual components comprising the valve train will vary depending on the type of engine and its application (Manning, 2012). The valve-train is a crucially important factor when designing an ICE, even more so when designing one to be used in motorsport, due to the harsher working conditions each component will be put under. The malfunctioning of the valve-train can result in *“fuel loss, combustion pressure, reduction of emission of gases, loss of engine productivity and consequently generating noise”* (Cavalca, 2007).

This report foremost consists of an in-depth investigation to gain an understanding of valve-train configurations, tribological processes other and characteristics. Subsequently, a comparative analysis consisting of a dynamics study, friction model and a FEA simulation was carried out on two valve-trains from two different areas of Motorsport. One being from the world of Moto3 Road Racing and the other Motocross Enduro racing, the aim of this being to understand the reasoning and relevance behind the different choices made by the manufacturers during the design process.



Figure 1 SymCirrus Motorsport's 2014 KTM RC250R



Figure 2 Team Miglio's 2017 Yamaha YZ250F (Enduro 21, n.d.)

The two valve-trains considered to complete the study were taken from the bikes shown in Figure 1 & Figure 2. The first being a 2014 KTM Moto3 RC250R rode by Edmund Best for SymCirrus Motorsport competing in the HEL Performance MotoStar support class in the 2019 BSB championship, which was kindly donated to the author. The second valve-train is from a 2017 GYTR cylinder head, taken from Team Miglio’s YZ250F used in the FIM Enduro World Championship, which was donated to a UWTSD Undergraduate named David Migliorati N’Doua, who kindly allowed the author to temporarily use the relevant components to carry out the appropriate tests to complete the study.

8 Literature Review

8.1 Valve-Train

8.1.1 Four-Stroke Cycle

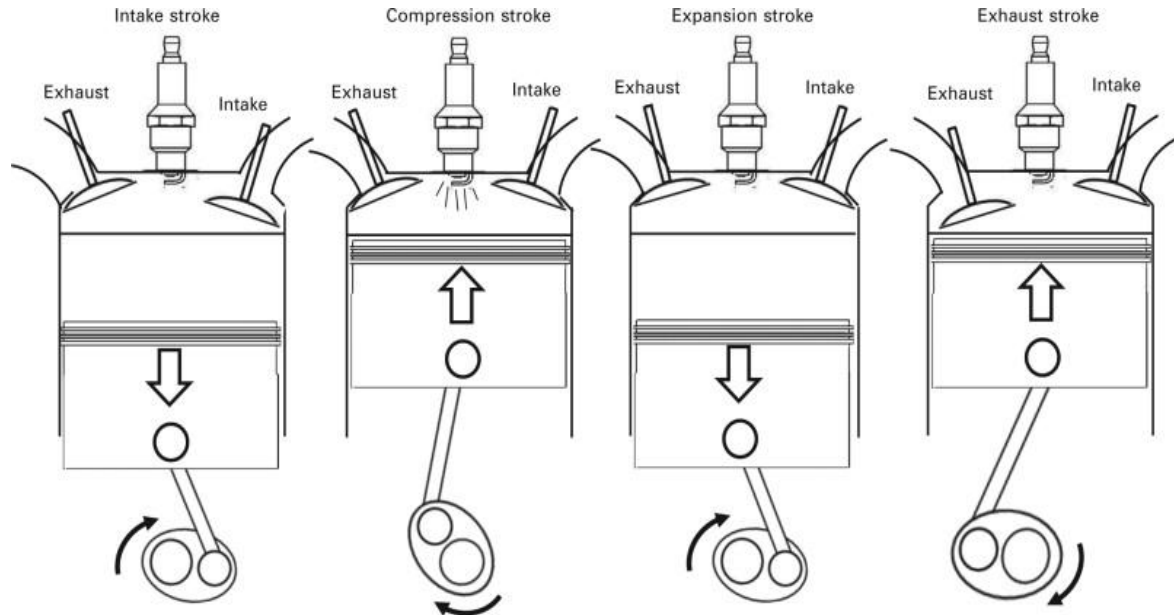


Figure 3 Four-Stroke Cycle (Johnson, 2014)

Figure 3 shows a diagram of the four-stroke cycle where every two full crankshaft revolutions (720°) the thermodynamic cycle is completed. Initially, during the intake/conduction stroke the intake valve is opened and the exhaust is closed. The valves are opened by the camshaft which rotates at half the frequency of the crankshaft “*depressing the valves against the valve springs to discharge the exhaust gas and to draw in the fresh gas*” (Bosh, 2002). Furthermore, during the intake stroke the piston descends, moving away from the cylinder head inducing a fresh charge of air and atomised petrol into the combustion chamber. An ICE which induces fresh charge by means of a depression within the cylinder is said to be N/A (*Naturally Aspirated*) (Heisler, 1999), both valve-trains considered within this study are taken from N/A engines. The next 180° of crank rotation is known as the compression stroke, where both the intake and exhaust valves are closed. The piston begins ascending towards the top of the combustion chamber compressing the air-fuel mixture which in-turn increases the pressure and temperature within the cylinder. Just before the piston reaches the top of its stroke during compression, the sparkplug ignites the dense combustible air/fuel mixture, by the time the piston reaches the innermost point of its stroke the mixture begins to burn, as a result of this the temperature and pressure within the cylinder rapidly increase until the gas forces exceed the resisting load. The burning gases then expand and in turn change the direction of the pistons motion to its outermost position (Heisler, 1999). According to the Bosh Automotive Handbook the exhaust valve opens just before BDC, and approximately 50% of the combustion gases escape under “*a supercritical pressure*

ratio during this pre-discharge phase” (Bosh, 2002). Next, when the compression stroke ends the intake valve remains closed, but the exhaust valve is opened. The Piston changes direction and as it starts ascending it sweeps essentially all of the combustion gases from the cylinder, during the exhaust stroke the gas pressure will drop from the exhaust valve opening pressure to atmospheric pressure (Heisler, 1999).

8.1.2 Valve Timing

The term “*Valve Timing*” refers to when and for how long the intake and exhaust valves are open and is a function of the following; the cam-profile and the relationship between the valve opening & closing and the position of the crankshaft.

“Various assumptions are made when calculating the theoretical thermal efficiency of an engine having a certain compression ratio” (Smith P. H., 1977). Amongst these is the assumption that each of the four-strokes occupies half of a crankshaft revolution, and that the intake and exhaust strokes both take place at atmospheric pressure. Whereas, in reality the pressure during intake barely equals atmospheric and when the exhaust stroke starts there is still a high pressure in the cylinder. In practice, the valve timing is arranged to improve the breathing and exhausting, the intake valve is arranged to open before TDC and to close after BDC and vice-versa for the exhaust; open before BDC and close after TDC. Moreover, when a valve opens so many degrees of crankshaft rotation before either BDC or TDC it is referred to as valve lead, and when a valve opens so many degrees after BDC or TDC it is referred to as valve lag (Heisler, 1999). This can be seen in Figure 4 on the subsequent page, which shows a valve timing diagram from the 1994 textbook “*Automotive Engines (Eighth Edition)*” which displays the complete cycle of events as a 720° spiral, representing two full crank revolutions. This will vary for different engines dependant on what is required of them and what applications they are used in, for more examples of spiral valve timing diagrams please refer to Appendix 14.1.

Figure 4 shows that the exhaust valve begins to open 47° before BDC on the power stroke and remains open until 21° after TDC on the intake stroke, giving more time for the exhaust gasses to exit the cylinder. Moreover, by the time the piston reaches 47° before BDC on the power stroke, the combustion pressure will have dropped significantly, therefore little power is lost by giving the exhaust gases this extra time to leave the cylinder. Figure 4 also shows that the intake valve opens 12° before TDC and remains open until 56° past BDC after the intake stroke, giving further time for the air-fuel mixture to enter the cylinder.

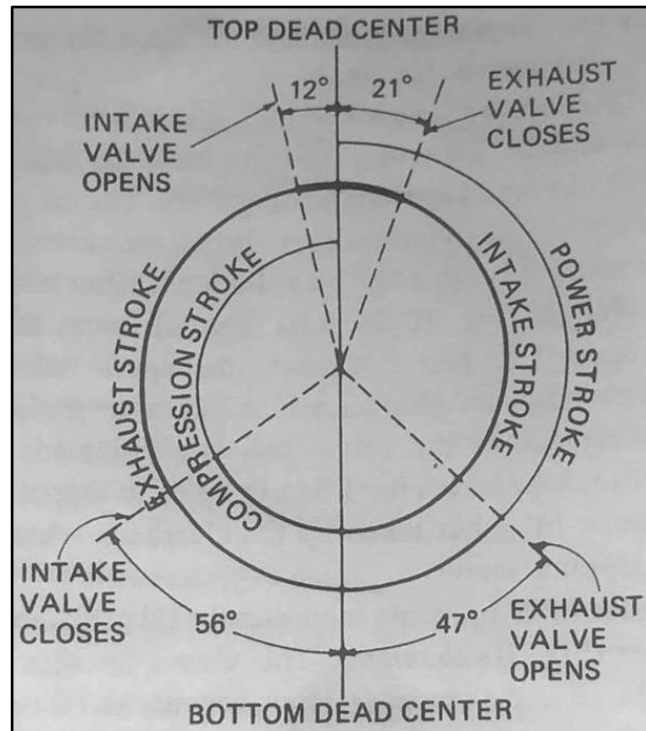


Figure 4 Valve Timing Diagram (Crouse, 1994)

As previously stated, the intake valve opens 12° before TDC and the exhaust valve opens 21° after TDC, resulting in both valves being open for a period of 33° of crankshaft rotation, this is known as “Valve Overlap”, it takes several degrees for the valves to fully open or close after passing the timing points. Valve overlap aids in scavenging or expelling the remaining exhaust gases from the cylinder (Crouse, 1994). Scavenging is the term used for when the rush of exhaust gases leaving the cylinder helps to pull in more fresh air.

8.1.3 Valve-Train Configurations

Historically, a lot of ICE's used to have the camshaft located in the cylinder block and the valves placed in the top face of the block as opposed to the cylinder head. In this situation, the valves were conducted directly by a simple cam follower, these were known as side-valve engines. These are no longer adequate due to their inferior "*efficiency, performance, fuel economy and exhaust emission*" (Manning, 2012). The most frequently used valve-train configurations are as follows; OHV, SOHC & DOHC.

8.1.3.1 OHV (Overhead Valve)

In an OHV or "*Pushrod*" ICE, the camshaft is within the cylinder block and the valve within the cylinder head, please refer to Figure 5. Within this system, there are long rods which are pushed by the cam-lobe, in turn moving the rocker arm opening the valves, hence the name "*Pushrod*".

Within an OHV engine, the cylinder head must incorporate through holes to accommodate the pushrods, these holes can also act as oil drains. Furthermore, it is crucial to take this into account during the early stages of the cylinder head design phase as they can influence the port design. (Manning, 2012).

The downside to the OHV system is that they are realistically only applicable to low engine speed applications. This is because the rocker arm and push rods have inertia, this inertia causes the components to resist changing speed and direction until enough force is applied to them. Consequently, this leads to a phenomenon called "*flexing*" where the rocker arm and pushrod bend marginally before they open the valve having a knock-on effect with valve action (Crouse, 1994).

8.1.3.2 SOHC (Single Overhead Cam)

In an OHC system whether it is single or double the camshaft is situated within the cylinder head unlike an OHV. Figure 6 shows a diagram of a SOHC in where there is one camshaft activating both intake and exhaust, meaning that there can be a maximum of two valves per cylinder, much like a pushrod engine. Moreover, in many ways the function of an SOHC is similar in methods to that of a pushrod system, however they're more robust because they don't have "*the inherent risk of bent pushrods*" (Padgett, 2020).

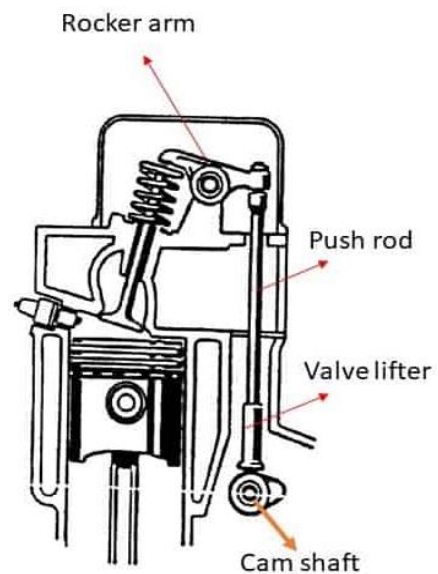


Figure 5 OHV Pushrod System (Bisa Otomotif, 2019)

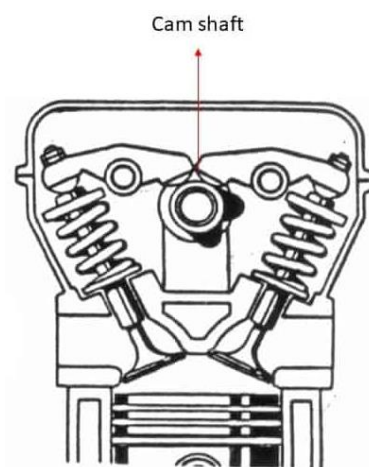


Figure 6 SOHC (Bisa Otomotif, 2019)

8.1.3.3 DOHC (Double overhead cam)

A DOHC has two camshafts, one for intake and one for exhaust, this design can provide the “*optimum valve train*” regarding high-performance engines. Furthermore, having two camshafts allows for four valves per cylinder which in turn gives a very good control of gas flow and combustion characteristics (Manning, 2012).

The additional camshaft per cylinder row aids in allowing each of the valves to open and close with greater precision, allowing close to optimal timing. Moreover, the additional valves also allow the engine to circulate fuel and exhaust with more ease. Therefore, giving DOHC systems the ability to produce more power than SOHC's or OHV's. Additionally, they are more efficient, cooler, run quieter and can be used at higher engine speeds making them the go-to valve-train configuration for high performance and race application engines. (Padgett, 2020).

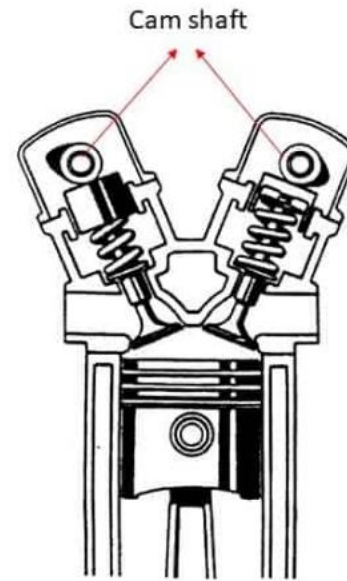


Figure 7 DOHC (Bisa Otomotif, 2019)

DOHC valve trains can incorporate direct-acting tappets (Figure 7), rockers or finger followers to transfer the cam motion to the valves. The two valve trains considered within this study are both DOHC systems; KTM (*Finger Follower*) & Yamaha (*Direct Acting*). These systems are explained in more detail in section 8.1.4.

8.1.3.4 Alternative Valve-Trains

➤ Desmodromic

- A reciprocating engine poppet valve that is positively closed by a cam and leverage system as opposed to a valve spring, these systems are used in very limited applications such as high-performance engines and racing engines (Manning, 2012).

➤ Pneumatic

- A system of mechanical valve actuation, which incorporates a small pneumatic cylinder and piston assembly which is fed with high-pressure air to close the valve, a “*air spring*” opposed to a conventional valve spring. The valve-trains are currently used in race engines with engine speeds up to 18,000RPM, and are also used within F1 engines (Manning, 2012).

➤ Electromagnetic

- Replaces the conventional cam and mechanical valve actuation system with electro-mechanical solenoids around the valve stem, one solenoid is electrically actuated to

open the valve stem and the second solenoid is electrically actuated to close it. This type of system is current only research and prototypes (Manning, 2012).

8.1.4 Yamaha YZ250F & KTM Moto3 Valve train configurations

Figure 8 displays the components within the poppet valve assembly of the 2014 KTM RC250R used within this study, the valve assembly of the GYTR head in the Yamaha YZ250F uses the same type of components.

The purpose of the helical valve spring is to return the valves back to their seated position after opening and keeping them closed. Furthermore, there are two main types of valve springs; single-coil design as used in the Yamaha YZ250F and dual-coil as used in the KTM RC250R, this is discussed in further detail in section 9.8. The spring retainers are washers, shaped to fit the top of the valve spring, the retainer is attached to the top of the valve stems by removable fasteners, known as collets. The Collets snap into a groove in the end of the valve stem and are there to ensure the valve spring, spring retainer and valve stem remain together. Finally, the valve shim is a disc of hardened metal used to adjust the clearance between the backside of the cam-lobe and the valve on which it operates.



Figure 8 KTM RC250R Valve-Train Components

8.1.4.1 Direct Acting Bucket Tappet (Yamaha)

The GYTR cylinder head from the YZ250F uses a DOHC direct acting bucket-tappet system. This type of valve-train configuration does not use a transmission link between the valves and the camshaft, instead the cam stroke is transmitted directly to the base of the bucket (Figure 9 Laterally Misaligned Bucket System Figure 9). Furthermore, the tappet then transmits the movement of the cam to the end of the valve stem, thereby directly activating the valve.



Figure 9 Laterally Misaligned Bucket System (Leva, n.d.)

Systems with direct valve actuation stand out for their high rigidity and small moving masses, in turn ensuring good performance at high speeds. On

the other hand, they're actuated by sliding contact which causes frictional losses, these losses can be reduced by using more suitable material and utilising friction-reducing coatings, such as DLC etc. These processes are discussed in further detail in section 8.2. Further reduction in wear and tear can be achieved when the cam is ground at an angle and attached opposite the bucket tappet in lateral misalignment as shown in Figure 9, rotating the tappet through a certain angle at each stroke (Ruvile, 2018), this technique is used within the GYTR YZ250F valve-train as shown in Figure 10 below which is taken from the owner's manual.

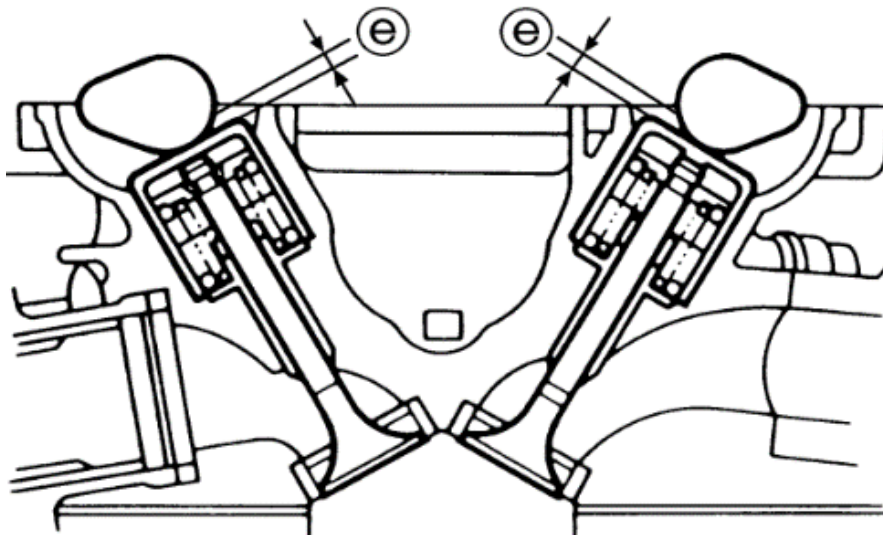


Figure 10 YZ250F Valve-Train Diagram (Yamaha, 2017)

Within this configuration there must always be a defined clearance between the base circle of the cam-lobe and the base of the bucket, this balances out the changes in length of valve train due to thermal expansion (Schaeffler; Rep-Xpert, 2017). The clearance is adjusted by means of the shim thickness; therefore shims can be supplied in various thicknesses.

The shims can either be loosely inserted in the tappet body, known as shim under bucket (Figure 11) or vice-versa on top of the tappet; shim over bucket (Figure 12). Furthermore, there are also tappets systems where adjusting shims are not utilised and instead the clearance is set through bucket-tappets with different base thicknesses.



Figure 11 Shim Under Bucket (Schaeffler; Rep-Xpert, 2017)



Figure 12 Shim Over Bucket (Schaeffler; Rep-Xpert, 2017)

“Shim Pop” may occur, where the shim pops out of its seated position, a diagram explaining this process can be found in Appendix 14.2. Moreover, the YZ250F positions the shim under the bucket-tappet

which has reduced spring forces and frictional power due to the lower mass of the design variant (Schaeffler; Rep-Xpert, 2017).

8.1.4.2 Finger Follower (KTM)

The KTM Moto3 uses a finger follower system as opposed to the bucket tappet actuation system used in the GYTR YZ250F. Therefore, instead of a bucket the finger follower acts directly on top of the valve (*a shim is used to set the clearances*). In a conventional bucket-tappet system, the bucket has to go up and down with the valve stem which results in extra moving mass. Furthermore, the 2019 Kawasaki Ninja ZX-10R also utilise the finger-follower valve actuation system (*Figure 13*). The ZX-10R project leader Azuma-San stated during an interview, that as a

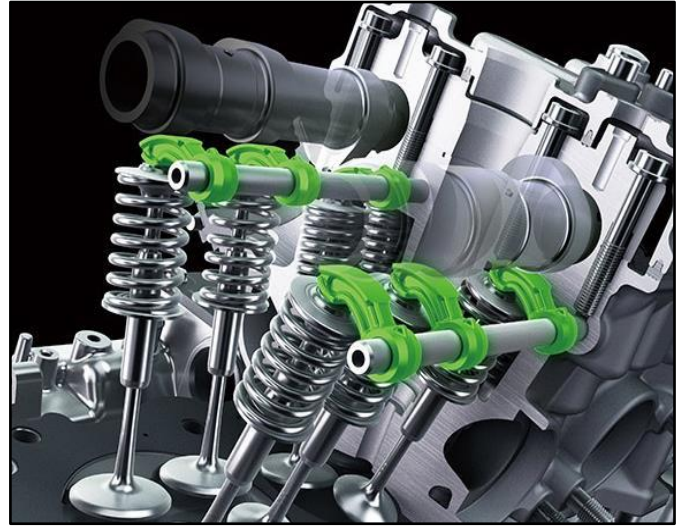


Figure 13 2019 Kawasaki Ninja ZX-10R Valve Train (Urry, 2018)

result of the finger follower acting directly upon the valve there is approximately 20% less moving mass. Furthermore, due to this reduction in moving mass *“the higher you can safely allow the motor to rev without compromising reliability”* (Azuma-san, 2018). In addition to this, in 2017 Suzuki decided to move from a bucket and shim valve actuation system to finger followers in the GSX-R 1000 and as a result increased the rev limit by 1,000 RPM to 14,500 RPM (ZigWheels, 2016). Also, each finger follower in the GSX-R weighs 10 grams, 6 grams less than the previously used bucket-tappet system which is a reduction of 37.5% (Bridge Motorcycles, 2016). The KTM RC250R Moto3 is a high-revving and extremely lightweight race bike making the finger follower the ideal valve-train configuration. Additionally, in the BSB Motostar Championship, FIM Moto3 Junior Championship and the World Moto3 Championship the engine speed is limited to 13,500RPM, the regulations stating this rule for all three championships can be found in Appendix 14.3.

Valve-train systems that utilise bucket-tappet's are limited in how aggressive the cam profile can be due to the buckets diameter, as space is limited within the cylinder head. On the other hand, with finger followers the cam-lobe is striking a smaller area (*Just the back of the follower*), so a more aggressive cam profile can be used, which comes with power benefits (Azuma-san, 2018). This further aids in explaining why KTM use finger followers as *“To gain low-end torque you want a shorter duration, however for top-end power you need a higher lift”* (Azuma-san, 2018), and with a more aggressive cam profile higher valve-lift can be achieved. A Moto3 engine is usually being revved between 9,000-13,500RPM and *“a Moto3 rider will also be using maximum power for about 70% of that time”* (Motocross Action , 2014). Furthermore, in Enduro racing the rider will never be on full throttle for

anywhere near the same amount of time as a Moto3 and also favours low-end torque, therefore the bucket-tappet system used in the Yamaha YZ250F is adequate.

8.1.5 Camshaft

The dictionary defines the word “*Cam*” as a rotating or sliding piece in a mechanical linkage used to transform rotary motion into linear motion and vice versa (Merriam-Webster, 2020). The camshaft is one of the main components within the valve-train which directly correlates to the action of the valves, “*Valve timing – and thus gas exchange – are regulated by a control shaft (camshaft) rotating at half the frequency of the crankshaft by which it is driven*” (Bosh, 2002). A Technical paper from 1961 titled “*Recent Developments in Cam Design*” stated that “*one might regard to the automotive engine type cam as not more than a smooth lump on another wise round shaft*” (J. H. Nourse, 1961). However, these smooth lumps known as the lobes are vital in any camshaft, as the shaft spins the lobes open and close the valves in time with the motion of the piston, there is a direct correlation between the shape of the cam lobe and the way the engine performs in different speed ranges. The shape of the lobe is referred to as the cam-profile.

8.1.5.1 Cam-Profile Terminology

8.1.5.1.1 Base Circle

The base circle also known as the “*Heel*” is the concentric or round portion of the cam lobe and is the lowest point of the lobe where the valve is in closed position. Furthermore, it is where valve clearance adjustments are made.

8.1.5.1.2 Opening & Closing Ramps

Immediately adjacent to the base circle, the cam has a portion with low velocity removing slack from the valve-train at the start of the lift sequence so there is not a collision. In addition to this, a similar closing ramp is used so the valve gently returns to its seat and will not cause “*Bounce*” (Tilden Technologfies, 2007).

8.1.5.1.3 Flank

The flanks are the portion of the lobe with large velocity and acceleration to move the valve as fast as possible (Tilden Technologfies, 2007).

8.1.5.1.4 Nose

The nose or “*Toe*” is the part of the lobe with the highest point of lift for the valve, it is where the valve is kept open for as long as possible before the transition to closing.

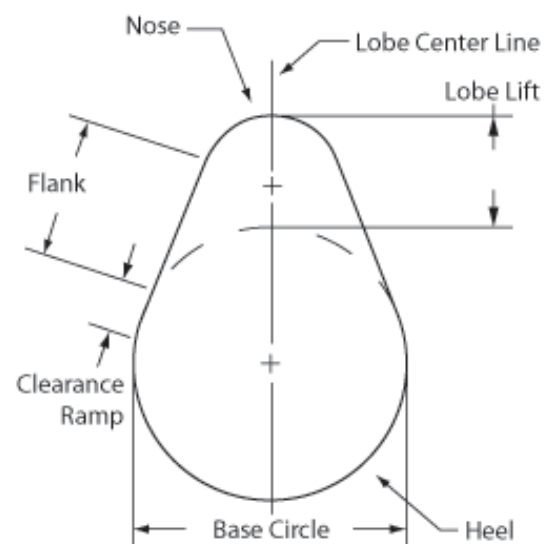


Figure 14 Cam-Profile Diagram (Lunati, 2020)

8.1.5.1.5 Lobe Separation Angle

The lobe separation angle is the angle between the centre lines of the intake and exhaust cam lobes. For example a 110° separation angle means that the peak opening points of the intake and exhaust lobes are 110° apart, this measurement is another way of expressing valve overlap. “Measurements typically range between $104-115^\circ$ ” (Lubken, 2016). A tighter angle will come with the following characteristics; lower RPM torque range, higher maximum torque and cylinder pressure, lower idle vacuum, rougher idle quality and more valve overlap, as opposed to a wider lobe separation angle which will have an opposite effect on these characteristics (Lubken, 2016).

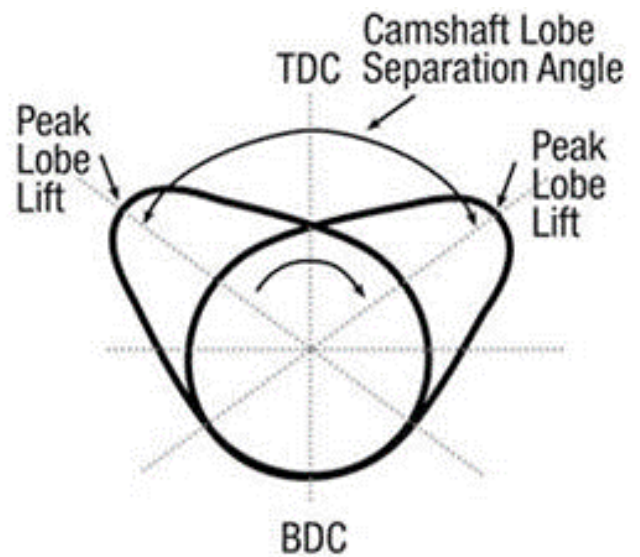


Figure 15 Lobe Separation Angle (Lubken, 2016)

8.1.5.2 Symmetrical & Asymmetrical Cam-Profiles

The camshaft found in the GYTR head of the Yamaha YZ250F is “Symmetrical” in design meaning it has identical opening and closing ramps and flanks on either side of the lobe. Therefore each side is “a mirror image of the other” (Summit Racing Equipment, 2016), meaning the intake and exhaust valves open and close at the same rate. Symmetrical cam profiles are appropriate for low to medium speed engines, but when the speed of an engine reaches a significantly high point, the valve velocity, acceleration, vibration and noise from valve loading becomes so intense the abrasion and wear of the valve will be accelerated and the durability of the camshaft will be deteriorated (Gao Wenzhi, 2004).

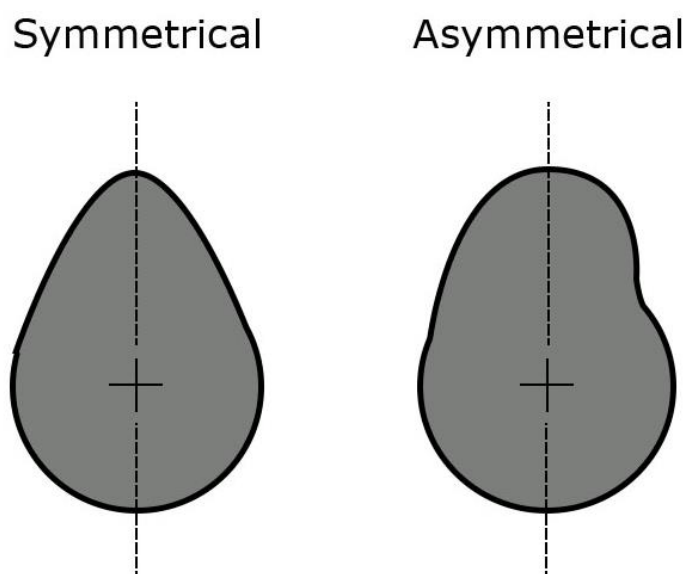


Figure 16 Symmetrical & Asymmetrical Diagram (Summit Racing Equipment, 2016)

The KTM RC250R Moto3 uses an “*asymmetrical*” camshaft. An asymmetrical cam profile has different opening and closing ramps/flanks unlike the Yamaha YZ250F, as shown in Figure 17 & Figure 18. With an asymmetric design an opening profile with a high velocity and a closing profile with a slower velocity can be achieved. With this process the valve can be set down more gently than the rate that it was opened (Lunati, 2020).



Figure 17 Yamaha YZ250F Cam-Profile



Figure 18 KTM RC250R Moto3 Cam-Profile

8.2 Tribological Processes in the Valve-Train

Tribology is the study of friction, wear & lubrication of interacting surfaces in relative motion. The valve-train makes a consequential input into the overall frictional losses within an internal combustion engine, the total amount depends on the following variables; Valve-train type, engine architecture, engine speed, lubricant & temperature amongst numerous other contributing factors. The power losses within an ICE valve-train are a function of the following;

- **Contact Load**
 - “a function of spring force, inertia forces and vibration.”
- **Sliding Velocity**
 - “determined by the cam profile, the geometry of the mechanism and the engine speed.”
- **Friction Coefficient**
 - “a complex variable dependent on contact force, sliding speed, lubricant film thickness, lubricant viscosity, lubricant temperature, surface texture, materials combination etc.”

(Michele Calabretta, 2010)

Maria Bouchet & Makota Kano wrote a chapter titled Superlubricity of Diamond/Glycerol Technology Applied to Automotive Gasoline in “*Superlubricity*”, in which it was said that the friction produced between the sliding interfaces between the cam-lobe and follower is accountable for around 80% of the friction within the valve-train and 20% of the total engine friction (Kano, 2007). Whereas, another study states that in a direct-acting mechanical bucket tappet-type valve-train, the sliding contact between the cam-lobe and tappet is responsible for about 85% of the total valve train losses but merely contributes between 6-10% of the total friction within an ICE. Moreover, this number is comparatively minor with losses due to other components within an engine but can still be further reduced with a range of methods (Arup Gangopadhyay, 2012).

A study carried out in 1996 claimed that the wear of valve train components had progressively become a problem for durability and reliability within an ICE, as over the years there has been many changes to engine component designs in order to meet the requirements of emission legislation and the high performance of automotive engines. Moreover, these changes normally put engine components in a more severe working condition, for example, modern valve trains have higher valve lift and are more complex than older systems. These changes have had a knock-on effect on valve train loading and other operating factors which often cause surface damage to valve train components (Lisheng Yang, 1996). Therefore, the study to understand the tribological behaviour of valve train components is a crucial factor when designing an engine. *“More recently designers have become more energy-conscious and the friction power loss associated with the cam mechanism has become more important”* (Emes, 1998). There are four lubrication regimes that describe the interaction between contacting surfaces.

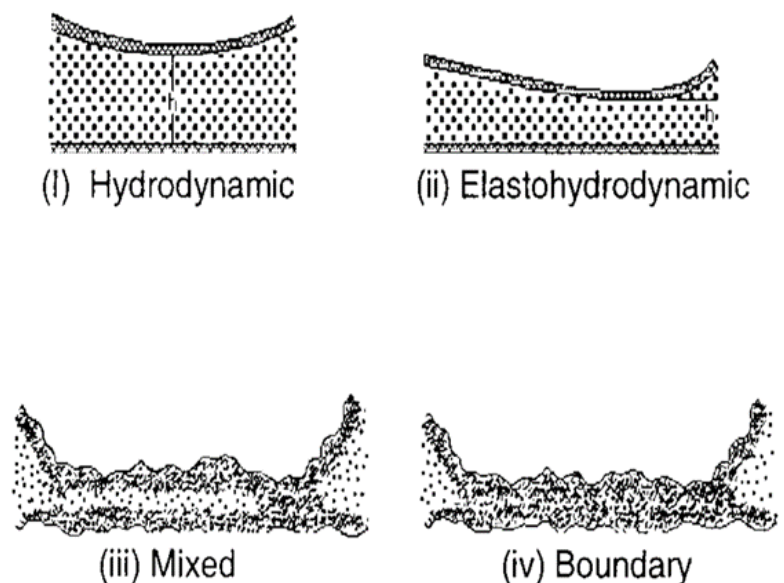


Figure 19 Schematic of Lubrication Regimes (Emes, 1998)

8.2.1 Lubrication Regimes

Hydrodynamic lubrication was first studied by Osborne Reynolds in the late 1800's where he discovered that when a lubricant was applied to a shaft and bearing, the rotating shaft pulled a converging wedge of lubricant between both components. Reynolds also noted that as the velocity of the shaft increased, the rate of flow of the liquid between them also increased as the *"lubricant is viscous, produces a liquid pressure in the lubricant wedge that is sufficient to keep the two surfaces separated"*. Moreover he stated that assuming ideal conditions the liquid pressure is great enough to keep the two bodies from experiencing physical contact and the only friction put upon the system was the viscous resistance of the lubricant (Marine Diesels, 2008). Moreover, this requires low load concentration, high sliding & high lubricant viscosity. Friction losses are small and are due to shearing of the fluid film which enables the friction force to be predicted using conventional fluid mechanics.

$$Force_f = \frac{Viscosity \times Area \times Velocity}{Film Thickness}$$

Equation 1 Hydrodynamic Friction Force (Emes, 1998)

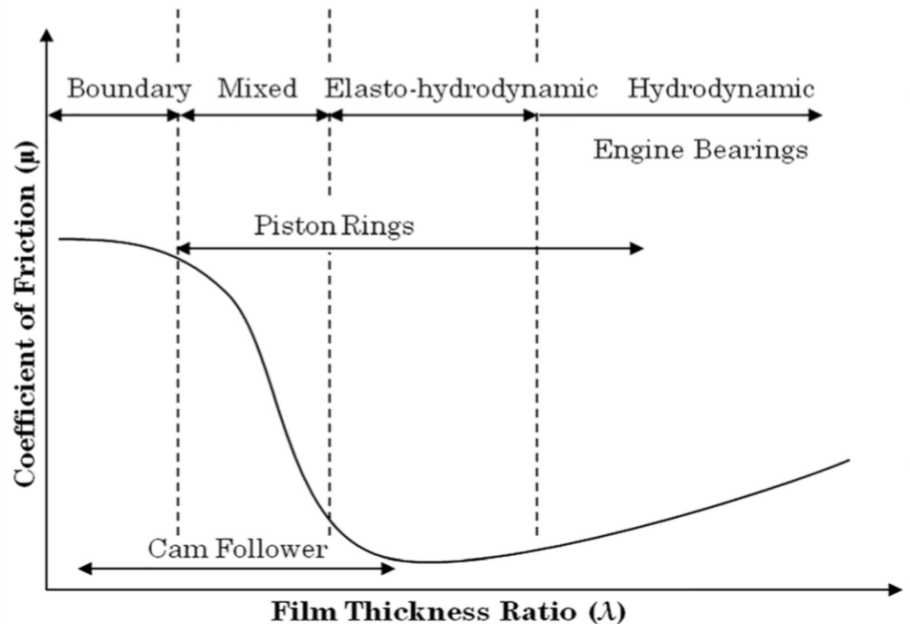
As shown in Equation 1, frictional force is inversely proportional to film thickness. Therefore, by decreasing the film thickness of a given geometry will in turn increase the systems friction force. On the other hand a thinner film develops larger fluid film pressures and therefore will carry higher loads, *"the net result is that as the normal loading increases, the coefficient of friction decreases"* (Emes, 1998).

EHL (*Elastohydrodynamic Lubrication*) is the quintessential regime for friction pairs with elastic contact. EHL is a development of HL (*Hydrodynamic Lubrication*) considering the elastic deflection of solid contact forces. Moreover, film thickness is considerably smaller than in conventional HL and under these conditions, a continuous hydrodynamic film can only be accomplished through elastic deflection customarily in consolidation with a growth in viscosity as a result to the high contact pressure (Zhang, 2018) & (Marinescu, 2004).

ML (*Mixed Lubrication*) is delineated as a mixture of solid contact and liquid lubrication, it is an intermediate stage between boundary lubrication and either HL or EHL. Furthermore, in a conformal contact the regime will move from HL into mixed and finally boundary as the load escalates. A non-conformation contact will move from EHL into ML and subsequently to BL (*boundary lubrication*), and *"as with all lubrication regimes the transition into and out of mixed lubrication is not clearly defined"* (Tribology News, 2009). A section within *"Plant Engineer's Reference Book"* states that BL could perhaps utmost be defined as the lubrication of surfaces by fluid films so thin that the coefficient of friction is affected by the type and nature of the surface and is predominantly independent of viscosity (Sloman, 2003). BL ensues when any of the fundamental factors that determine full-fluid film formation

between contacting surfaces are inadequate, such as when the viscosity of fluid is deficient to avert surface contact. Moreover, in comparison with HL, EHL & ML regimes, wear and friction under boundary conditions are the most uncompromising conditions for the lubricant to control. (Jan C.J. Bart, 2013).

Graph 1 displays a Stribeck Curve showing different regimes of lubrication regarding ICE components whilst under sliding motion and shows that the cam follower is situated between BL and the start of EHL. Two more Stribeck Curve graphs were attained from two different studies of the same nature which also found that valve train components fall under the same lubrication regime within an ICE, please refer to Appendix 14.4. In addition to this, another



Graph 1 Stribeck curve (Agarwal, 2019)

study tells that the lubrication modes within the valve-train range from “*predominantly hydrodynamic to boundary lubrication and mixed lubrication*”. (Tung, 2016). The study further states that valve-train friction is relatively more consequential at lower speeds, demonstrating that the majority of valve-train friction comes from BL & ML.

8.2.2 Coatings & Surface Finishing

Friction and wear within the valve-train can be further reduced by surface finish and thin-film coating. Within racing applications there are an array of contributing factors regarding the use of coatings;

- “Corrosion Prevention”
- “To reduce abrasive wear”
- “To improve the friction coefficient”
- “Improvement in materials compatibility problems”
- “Modification of other material problems”

(Wallwork, 2020).

There are numerous types of thin-film coatings motorsport engine manufacturers consider when designing the valve-train. Furthermore, these coating allow ICE's to run at greater temperatures and at a faster rate with lower frictional losses, without the use of thin-film coatings within motorsport many engines would not function optimally. TiN (*Titanium Nitride*) is a frequently used coating on valves reducing frictional effects in both "*austenitic stainless steel & titanium alloy valve stems*" (Wallwork, 2020). Moreover, valve spring retainers, especially ones made from titanium are typically coated using TiN or CrN (*Chromium Nitride*) in order to prevent damage brought on by wear from the interface with the flat edges of the valve spring. CrN offers the same properties as TiN but has a greater resistance to temperature which makes it the popular choice for components exposed in high temperature environments.

TiN and CrN are both products of PVD (*Physical Vapor Deposition*) which is an assortment of vacuum deposition methods and is a widely used process for the fabrication of surface coatings and thin films. All reactive PVD hard coating processes combine the following steps;

- "*A method of depositing the metal*"
- "*Combination with an active gas, such as Nitrogen, Oxygen or Methane*"
- "*Plasma bombardment of the substance to ensure a dense, hard coating*"

The methods for PVD differ but the primary methods are ion plating, ion implantation, laser surface alloying and sputtering. (Zeng, 2013). Additionally, the most typically used PVD coating is TiN and is practiced primarily on precision metal parts as it has an optimal balance of "*hardness, toughness, adhesion & inertness*". (Titanium Exposed, 2013).

Cam followers withstand high contact pressures and sliding velocities which can have a detrimental effect on contact temperatures and abrasive wear. Therefore, a super-polished CrN is known to be used as a coating for the benefits it provides when a component experiences elevated contact temperature. Furthermore, valves are also known to use CrN as it offers thermal barrier benefits which will aid in fuel conversion efficiency as on the combustion face there is "*reduced conduction through the stem*" and also improved cylinder filling by "*having a cooler valve head and seat*" (Wallwork, 2020). Subsequently, DLC (*Diamond Like Carbon*) coatings are applied to valves and essentially always used when coating rockers to mitigate the frictional forces exerted upon on them. DLC is prevalently when coating camshafts "*cams have been coated for a number of years to reduce the fiction effect in contact with the associated counter face*" (Wallwork, 2020).

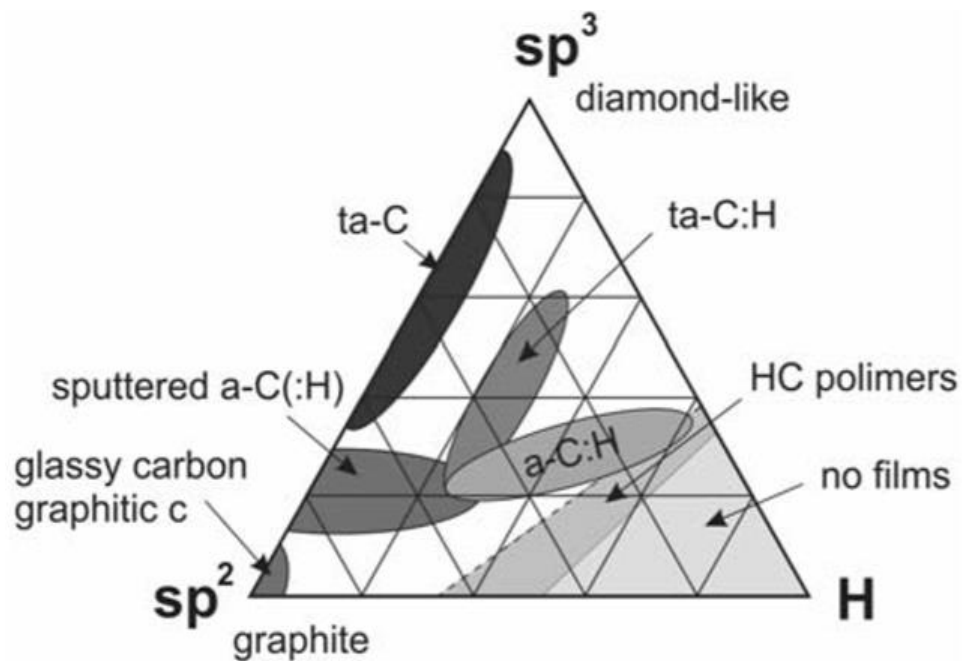


Figure 20 Ternary phase diagram of DLC (M. Kalin, 2008)

Figure 20 shows a ternary phase diagram of sp^2 , sp^3 Hydrogen contents of various forms of DLC and was extracted from a study on BL mechanisms of DLC coatings. Essentially, a Ternary Phase Diagram represents the phase behaviour of mixtures containing three components in a triangle. Additionally, there is a range of DLC coatings which vary in levels of sp^2 and sp^3 bonded carbon atoms, ranging from a considerable amount of Hydrogen to almost none. DLC can be split into two main sections regarding its chemical structure;

- **Non-Doped**
 - Exclusively consists of Carbon and/or Hydrogen atoms.
- **Doped**
 - Contains additional metal or non-metal elements.

DLC coating can be further dissected into the following areas which can be seen in Figure 20.

- **Amorphous**
 - Non-Hydrogenated ($a-C$)
 - Hydrogenated ($a-C:H$)
- **Tetragonal**
 - A negligible amount of Hydrogen ($ta-C$)
 - Substantial amount of Hydrogen ($ta-C:H$)

(M. Kalin, 2008)

Figure 21 shows the difference in hardness of different types of DLC coatings displaying whether there is Hydrogen content or not in the chemical structure, it was sourced from an article on the Industrial Machinery Digest website. In addition to this, the article states that the most frequently used DLC coating type is hydrogenated amorphous carbon (a-C:H) and is typically applied through PACVD (*Plasma-Assisted Chemical Vapor Deposition*). The PACVD deposition method causes a chemical reaction through plasma excitation and ionization that then develops a coating hardness of approximately 15-30Gpa, which is a considerably high number, yet it is on the lower end of the DLC family. Furthermore, the same article then goes on to say that Hydrogen-free DLC coatings are even harder and offer a very low coefficient of friction, please refer to Figure 21. Hydrogen free coatings are typically applied by a method of PVD and are used in the most demanding environments “including high performance vehicles for friction, wear and contact areas of the engine and valve train” (Industrial Machinery Digest , 2020)

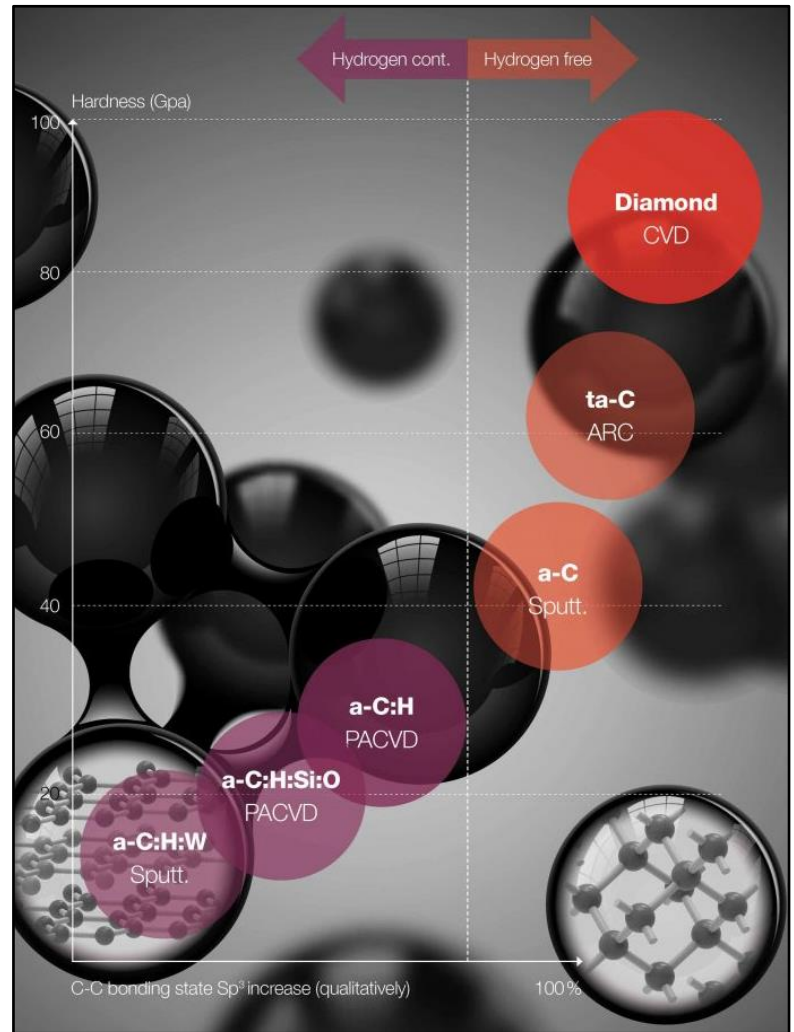


Figure 21 DLC Hardness by Type (Industrial Machinery Digest , 2020)

Furthermore, coatings are usually applied to valve train components in order to reduce friction. However, when looking at the collets if a coating is applied it is not done with the intention of reducing friction, as it is more beneficial to have a higher friction coefficient at the interface between the retainer and collet, in order to reduce movement that could cause the valves to fail. These coatings include soft metals such as silver, also in cases when there is no concern over material and no coating is required it is typical for collets to have been subjected to ‘Oxide’ surface treatments with the hope of averting corrosion. (Ward, Collets - Surface Treatments & Coatings, 2014).

In situations where a coating is not used, it is worth noting that improving the finish of a surface is proven to decrease friction within the valve-train. One study found that a noticeable reduction in frictional torque in the valve-train is achieved if the roughness of the camshaft and inserts was reduced. Finally, the same study found that friction could be reduced by 10-40% in Nissan HR & MR engines

by reducing the composite surface roughness of the cam-lobe's and bucket tappets by 50% (Arup Gangopadhyay, 2012).

8.3 Computer-Aided Engineering

“Valve-train design and development has been one of the earliest applications of CAE to engine engineering functions” (Keribar, 2000). CAE is the broad practice of computer software to assist in engineering analysis tasks, consisting of three steps *“Pre-processing, solving and post-processing”*.

➤ **Pre-Processing**

- The Geometry is modelled and the physical properties of the design, alongside applied loads or constraints.

➤ **Solving**

- The model is solved using an applicable mathematical formulation of the elemental physics.

➤ **Post-Processing**

- The outcome is conferred with the engineer.

(Siemens, 2020)

In order to sufficiently complete the project, two areas of CAE were used; Computer-Aided Design & Finite Element Analysis. Moreover, both methods were carried out with the use of Dassault Systèmes SolidWorks, a 3D CAD design software, analysis and product data management software package (SolidWorks Dassault Systèmes, 2020).

8.3.1 CAD & FEA

CAD is used to help create, change, analyse and optimize models. Foremost, it aids engineers create better quality, more precise and neater designs and is used in most, if not all streams of the engineering industry. The concept of CAD has developed from simplistic 2D designs into multi-layered 3D structures with kinematic movement, it dates to 1957 when the *“Father of Computer-Aided Design”* Dr Patrick Hanratty created the first numerical control system, which later became CAD (Part Solutions, 2018). Since then, CAD has come on leaps and bounds with Dassault Systèmes releasing the first edition of the SolidWorks software package in 1995.

“SolidWorks is a world leader in 3D solutions that help millions of engineers and designers succeed through innovation” (Dassault Systèmes, 2012), it allows the user to produce 3D solid models of any part and assembly, regardless of size and complexity. Subsequently, within SolidWorks there is a portfolio of available analysis tools that utilise FEA to predict a components real-world physical behaviour by carrying out virtual testing on its CAD model. As a result of the ever-increasing complexity of ICE components, a FEA simulation needs to be completed in order to determine the

feasibility of its structural performance as well as its kinematics and dynamics before a prototype can be manufactured. In addition to this, FEA works by a method of breaking the component down into a multitude of smaller pieces and predicts how it will react to physical effects such as forces etc, *“breaking down a real object into many finite elements. Mathematical equations help predict the behaviour of each element, adding up all the individual behaviours predicts the behaviour of the actual object”* (AutoDesk, 2018).

9 Methodology

9.1 Removal of Components

9.1.1 KTM Moto3



Figure 22 KTM RC250R Moto3 Engine (Without Gearbox)



Figure 23 KTM RC250R Moto3 Engine Without Cylinder Head



Figure 24 KTM RC250R Moto3 Cylinder Head

In order to gather data relevant to the investigation (*Cam Lift, Valve Lift, Spring rates, etc.*) the valve-train components needed to be removed from the engine of the KTM RC250R. Firstly, the cylinder head was removed from the block.



Figure 25 KTM RC250R Moto3 Cylinder Head without cover



Figure 26 KTM RC250R Moto3 Cylinder Head Without Bearing Cap



Figure 27 KTM RC250R Moto3 Intake Camshaft

Next, the cover was removed and then the camshaft bearing cap so that both the intake and exhaust camshaft could be freely removed from the cylinder head. Figure 27 shows the intake camshaft, which is hollow in order to reduce weight. Furthermore, upon inspection it was discovered that the exhaust camshaft is an auto decompression cam, please see Figure 28 & Figure 29 on the subsequent page.

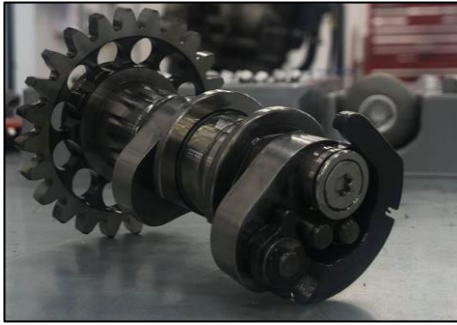


Figure 28 KTM RC250R Moto3 Exhaust Camshaft



Figure 29 Decompressor

“Decompressor systems are often used with single cylinder 4-strokes engines” (Cyclepedia Press, 2014), as previously stated both the KTM Moto3 and the Yamaha YZ250F are both single cylinder 4-strokes, and both utilise auto decompression camshafts. On the KTM Moto 3 the decompressor is situated on the lobe end of the camshaft (Figure 28), whereas on the YZ250F it is found on the sprocket end of the camshaft (Figure 30). The decomposer system works by a method of reducing the cylinder compression at low RPM for easy starting. Furthermore, the compression is reduced by momentarily opening an exhaust valve on the compression stroke to moderately vent the combustion chamber. Finally, reducing engine compression in turn reduces the force that is required to start the engine meaning smaller, less powerful electric starting components are needed. These mechanisms often use spring loaded weights to operate a pin within the camshaft, at rest and below idle engine speeds the weights are held in by the springs and the pin will contact the exhaust valve to release compression (Cyclepedia Press, 2014).

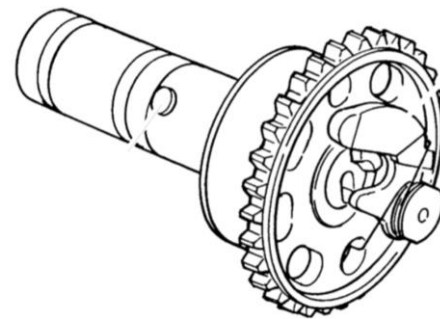


Figure 30 2017 Yamaha YZ250F Exhaust Diagram (Yamaha, 2017)

The following steps were then undertaken to remove the remaining KTM Moto3 valve-train components; the shims were removed, then the pins seen in Figure 32 were removed allowing the shafts (Figure 33) keeping the finger followers in place to be removed and then the followers. An initial examination of the finger followers found they were coated with DLC.



Figure 31 KTM Shim



Figure 32 KTM Pins



Figure 33 KTM Follower Shaft



Figure 34 KTM Finger Follower



Figure 35 Valve Spring Compressor (1)



Figure 36 Valve Spring Compressor (2)

Next, with the use of a valve spring compressor in the automotive workshop the collets were removed, in turn freeing the valve springs and allowing the rest of the valve-assembly to be removed from the cylinder head. Figure 38 displays the KTM intake valve-assembly, to see the exhaust valve-assembly please refer to Appendix KTM Moto3 Exhaust14.5.1.

Furthermore, upon inspection of the cylinder head it was discovered the valves are in radial arrangement. Moreover, SI engines with the purpose of higher specific outputs favour valves inclined towards each other, as shown with the KTM cylinder head in Figure 38, this is also the case with the Yamaha YZ250F, this configuration allows larger valve diameters for a given cylinder bore “while also providing greater latitude for optimizing intake and exhaust passage design” (Bosh, 2002).



Figure 38 Intake Valve-Assembly



Figure 37 KTM Inclined Valve Alignment

9.1.2 Yamaha YZ250F

The valve-train components had already been removed from the GYTR cylinder head used in the YZ250F, meaning they were immediately accessible to the author. Figure 39 shows an exploded view diagram of the valve assembly.

Shown below are images of the Intake valve-train from the GYTR cylinder head of the YZ250, for the same images regarding the exhaust valve-train please see Appendix 14.5.2.



Figure 40 Yamaha YZ250F Intake Camshaft

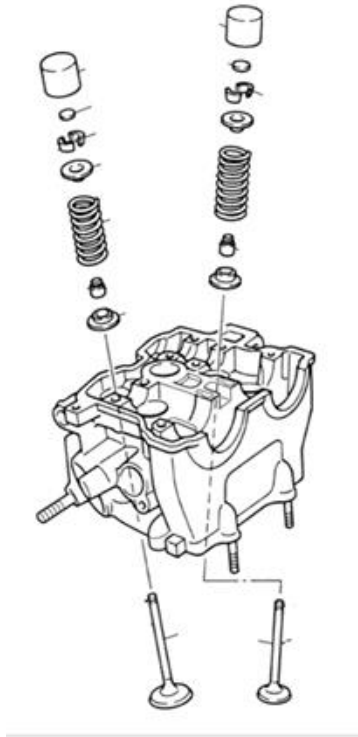


Figure 39 Valve-Assembly Exploded Diagram (Yamaha, 2017)



Figure 41 Yamaha YZ250F Bucket (Top Plane)



Figure 42 Yamaha YZ250F Shim



Figure 43 Yamaha YZ250 Intake Valve

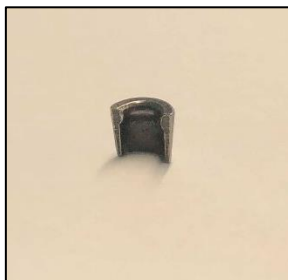


Figure 44 Yamaha YZ250F Collett



Figure 45 Yamaha YZ250F Spring Retainer



Figure 46 Yamaha YZ250F Intake Valve Spring

9.2 Masses

The masses of the components within both valve-trains were determined with the use of scales acquired from the Automotive Workshop, as shown in Table 1 below.

	KTM	Yamaha	
Inlet Valve	21.00	20.00	g
Exhaust Valve	18.00	14.00	g
Shim	1.00	1.10	g
Spring Retainer	7.00	5.50	g
Valve Spring	35.00	30.00	g
Collett	0.50	0.40	g
Intake Camshaft	262.00	291.00	g
Exhaust Camshaft	264.00	311.00	g
Finger Follower	8.00	N/A	g
Bucket	N/A	12.50	g

Table 1 Mass of components

The moving mass of each valve train was then calculated using the following equations;

$$YZ250 \text{ Mass}_{\text{Moving}} = \text{Valve} + \text{Bucket} + \text{Shim} + (2 \times \text{Collett}) + \text{Spring Retainer} + \frac{1}{2} \text{Spring}$$

Equation 2 YZ250 Moving Mass

$$\text{Moto3 Mass}_{\text{Moving}} = \text{Valve} + \text{Follower} + \text{Shim} + (2 \times \text{Collett}) + \text{Spring Retainer} + \frac{1}{2} \text{Spring}$$

Equation 3 Moto3 Moving Mass

The finger follower of the KTM is 36% lighter than the bucket-tappet used in the YZ250F. Furthermore, as previously stated in section 8.1.4.2, Suzuki stated that the finger followers used in the valve-train of the 2017 GSX-R1000 are 37.5% lighter than a conventional bucket-tappet. Moreover, as also previously discussed

	Moving Mass		
	KTM	Yamaha	
Intake	47.50	54.90	g
Exhaust	44.50	48.90	g

Table 2 Moving Mass

Kawasaki said that when the Ninja ZX-10R switched from a bucket-tappet system to a finger follower system in 2019, the moving mass of the valve-train was reduced by 20%. Regarding the KTM Moto3 (*Finger Follower*) and the Yamaha YZ250F (*Bucket-Tappet*), the KTM had 13.48% less moving mass within the intake valve-train, 9% less within the exhaust valve-train and 11.37% less overall.

9.3 Materials

The materials of the components were unknown and unattainable, particularly regarding the KTM Moto3 where the literature was so scarce. Therefore, research was carried out to discover common materials used for valve-train component in high-performance engines, based upon that research the following materials were chosen;

Component	Materials	
	KTM	Yamaha
Camshafts	Billet Steel Alloy	
Valves	Titanium Alloy (Ti-6Al-4V)	
Finger Follower	Cast Steel (DLC Coated)	N/A
Bucket-Tappet	N/A	Steel Alloy
Shim	Stainless Steel	
Collett	Titanium Alloy (Ti-6Al-4V)	
Valve Spring	Chrome-Silicon Steel Alloys (ASTM A877)	
Spring Retainer	Titanium Alloy (Ti-6Al-4V)	

Table 3 Materials

9.3.1.1 Camshafts

Mass produced camshafts which are distributed in higher volumes are generally made from a chilled iron casting and then nitride coated. However, as shown in Table 3 was not the material used for the camshafts of the high-performance valvetrains in the KTM or GYTR cylinder head, “*Aftermarket & high-performance camshafts are made out of billet steel*”, they are initially machined on a CNC and then rough ground to a somewhat oversized specification. Finally, they are heat treated and finish ground to the exact measurement (Motocross Action, 2012). Steel billets is the second stage product produced in the process of making steel bars. Yamaha state on their online shop that their GYTR Performance camshafts are in fact made of “*Special billet alloy*” and although no information was attained on the material of the KTM’s camshafts, the assumption was made to use a billet steel alloy for both.

9.3.1.2 Valves

The peak RPM of the YZ250F is 12,900 and as previously stated the KTM is limited to 13,500RPM, therefore both are considered “*high revving engines*”. Engine Builder Magazine stated in 2006 that titanium valves are favoured in high revving engines, as they allow a more aggressive cam profile that open and close the valves faster, also whilst titanium valves are more expensive, they “*Weigh about 40% percent less than stainless steel*” (Carley, Performance Valves, 2006). Finally, the FIM World Championship GP Regulations state that the valves must be made from “*either ferrous or titanium-based alloy*” (FIM, 2019).

Titanium Valve tips and are often coated with thin-film coating (*DLC, TiN, CrN, etc.*) as previously discussed within this report. Unfortunately, regarding the valves of the YZ250F and KTM it was unknown whether thin-film coatings were utilised. The friction model and FEA studies within this paper do not take the coefficient of friction, UTS and other characteristics of the valves into account, and as coatings are usually around 3 microns it will have an inconsequential effect on the overall mass of the valve. Therefore, the assumption was made that all the valves within this paper are made from titanium-based alloys.

9.3.1.3 Bucket-Tappet & Finger Followers

The finger followers are made from a cast steel allowing them a *“very complex component geometry, which enables an especially high resilience”* (Schaeffler; Rep-Xpert, 2017). Furthermore, the finger followers use a DLC coating, making them extremely durable, with a coefficient of friction of 0.05 (Fransen, 2020). The assumption was made that the Bucket-tappet of the YZ250F was made of steel (Juan R. L. Trasorras, 1998), did not use a thin-film coating and had a friction coefficient of 0.16 (Engineering ToolBox, 2004).

9.3.1.4 Shim & Collets

Valve shims are discs of a hardened metal of precisely calibrated height. Shims can be made out of an array of metallic materials, for the purpose of this report the assumption was made to use stainless steel.

There are situations within racing where aluminium has been used for collets in order to reduce weight, but this offers *“offers very little weight-saving potential”* compared to larger components within the valve train. Furthermore, titanium are often used when making collets and are known to be used in valve-trains being used with *“titanium valves and retainers”*, which is the case for both valve-trains in this study (Ward, Collets - surface treatments and coating, 2014) & (Ward, Aluminium in valvetrain applications, 2014).

9.3.1.5 Valve Springs & Spring Retainers

The valve springs are tremendously high-stressed components within an ICE's valve train (*Discussed further in Section 9.8*). During an interview the Product Engineering Manager at PAC Racing Springs stated that *“Almost all high-performance valve springs are produced from chrome-silicone steel alloys specified in material standard ASTM A877”* (Osborn, 2019).

As previously stated, titanium valves are typically 40% lighter than steel valves, and *“titanium retainers are usually half the weight of stock steel retainers”* (Carley, Understanding High Performance Valvetrain Technology). Therefore, the decision was made to use titanium for the valve spring retainers.

9.3.1.6 SolidWorks Validation

As a validation tool the chosen materials were applied to SolidWorks models of each component. The SolidWorks models and assemblies are thoroughly explained further on within the paper.

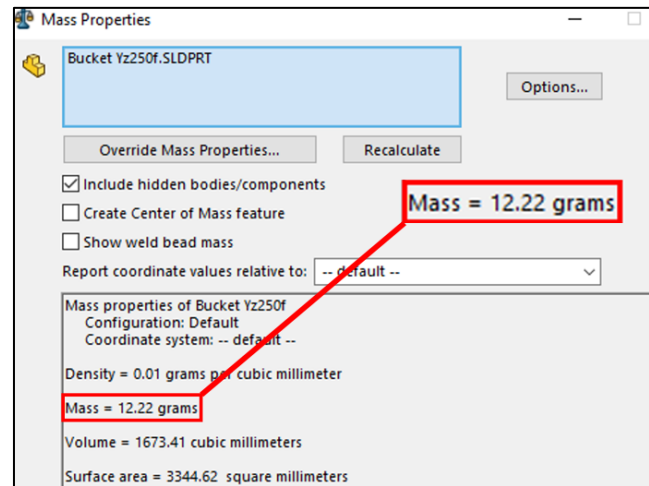


Figure 47 SolidWorks Mass Properties Tool YZ250 Bucket-Tappet

Once a material has been applied, SolidWorks automatically generates the mass of the component based on the volume and density, this can be accessed using the mass properties function as shown in Figure 47. The masses generated from the SolidWorks model was then compared to the actual masses of the components and the difference in percentage between the two was calculated.

Component	Actual Mass (g)		SolidWorks Mass (g)		Difference (%)	
	KTM	Yamaha	KTM	Yamaha	KTM	Yamaha
Intake Camshaft	262.00	291.00	257.50	288.00	1.73%	1.04%
Exhaust Camshaft	264.00	311.00	261.20	314.00	1.07%	0.96%
Intake Valve	21.00	20.00	21.80	19.60	3.74%	2.02%
Exhaust Valve	18.00	14.00	17.89	14.16	0.61%	1.14%
Finger Follower	8.00	N/A	8.04	N/A	0.50%	N/A
Bucket-Tappet	N/A	12.50	N/A	12.22	N/A	2.27%
Shim	1.00	1.10	0.98	1.05	2.02%	4.65%
Collett	0.50	0.40	0.48	0.39	4.08%	2.53%
Spring retainer	7.00	5.50	6.78	5.56	3.19%	1.08%

Table 4 Masses

All components were within a 5% error margin; therefore the assumption was made going forward that these materials were to be used. The Camshafts, Bucket-Tappets & Finger Followers were the most imperative components to correctly determine the materials of. This is because the characteristics of these play a crucial role within the friction model & FEA Study sections of this paper, all of these components had a percentage difference under 3%. Finally, the valve springs were not physically modelled within SolidWorks, as the “Spring” tool was utilised allowing the spring rate to be inputted without having to actually model the component.

9.4 Intake & Exhaust Valves Measurements

	Valve Head Diameter		
	KTM	Yamaha	
Intake	34.50	31.00	mm
Exhaust	27.00	25.00	mm

Table 5 Valve Head Diameter

With the use of a Vernier-Calliper the intake and exhaust valve diameters were measured, please see Table 5. The intake valve on both the KTM Moto3 and the Yamaha YZ250 are considerably larger than that of the exhaust, this is visibly shown in Figure 48. “When the intake valve is open, the only force moving air-fuel mixture into the cylinder is atmospheric pressure” (Crouse, 1994), therefore the intake diameter is bigger so that more volume of air can enter the combustion chamber during the intake stroke. Furthermore, during the exhaust stroke less room is needed for the exhaust gases to exit, as there is still high-pressure within the cylinder. “A smaller exhaust valve provides enough space for the high-pressure exhaust gases to get out of the cylinder” (Crouse, 1994).



Figure 48 KTM RC250R Moto3 Intake & Exhaust Valves

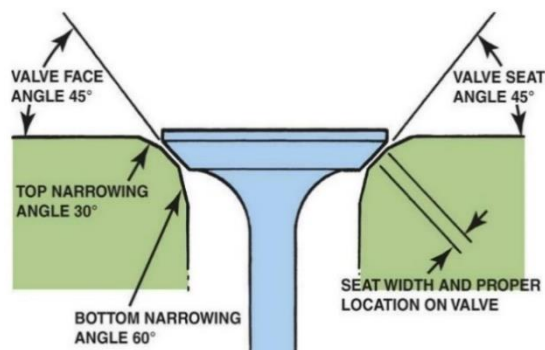


Figure 49 Valve Head Diagram (Halderman, 2012)

Figure 49 shows the different angles that can be found on certain valves; this diagram is applicable for both the KTM Moto3 & the Yamaha YZ250F. The angle of the valve face is usually 45° to the stem, although angles of 30° are known to have been used for intake valves.

Both the intake and exhaust valves of the YZ250F have a 45° as shown in Figure 49 from the owner’s manual. Subsequently, due to the limited literature regarding the KTM Moto3 valve-train, the valve face angle was unknown, an assumption was made that it was 45°. An article on Cycle World states that the late Kenny Augustine, who dedicated the majority of his professional career trying to get air into engines once said “you can make air go fast, and you can make it turn

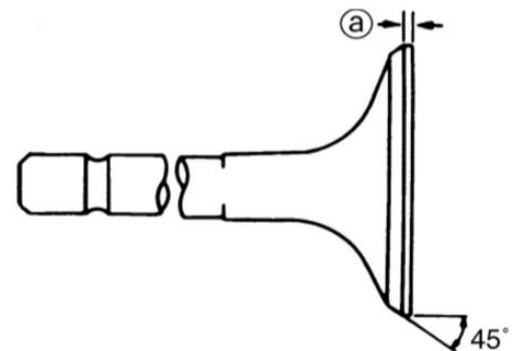


Figure 50 YZ250F Valve Head Diagram (Yamaha, 2017)

corners. *But you can't do both*". Thus, airflow can't make sudden turns, such as flowing straight along the port to suddenly turning outward 45° , "*the flow separates from the inner edge of the seat at the change to 45° because it can't make such a sharp turn*". When the flow fails to make the 45° turn, it separates from the surface, the separated flow isn't able to fill the entirety of the space between valve and seat, resulting in reduced flow, particularly at low lift. Therefore, the bottom and top narrowing angles (Figure 49) break down the 45° turn into three smaller turns, each turn is less sharp, making it easier for the air to turn. This increases the air flow, especially at low lift (Cameron, 2016). Furthermore, the section labelled "*a*" in Figure 50 is the margin part of the valve head, this is the width on the edge of the valve head between the top of the valve and the edge of the face. Too narrow a margin causes preignition and can cause the valve to over-heat resulting in damage. The YZ250F margin values are as follows; Intake – 0.8mm, Exhaust – 0.7mm, and the KTM Moto3 are; Intake – 0.9 mm. Exhaust – 0.9 mm.

9.5 Cam Lift

Cam lift is defined as the "*difference between the radius of the base circle and that of the nose*" (Westbury, 1956). For this investigation the cam lift of both the Moto3 and YZ250F was determined using the optical diving head in the Automotive Workshop.



Figure 51 Measuring KTM Moto3 Cam-Profile with Optical Dividing head

Figure 50 shows the intake camshaft of the KTM Moto3 RC250R when fitted in the Matrix Optical Dividing head, the same method was used for the exhaust camshaft of the KTM and for both camshafts of the Yamaha YZ250F.

The Following steps were carried out to successfully gather the cam lift data;

- Firstly, end caps needed to be inserted into either side of the camshaft allowing it to be fitted to the dividing head. Workshop technician Mark Fall was able to provide these for the KTM but unfortunately, they did not fit the camshafts of the Yamaha, therefore a pair needed to be fabricated from a billet of aluminium. The technical drawing and an image of the billet of aluminium can be found in Appendix 14.6, Figure 52 shows the final product.
- Secondly, when the end caps are in either side of the camshaft, the footstock can be moved as closely to the headstock as possible with the camshaft in place, the bolts allowing the footstock to move back and forth are then tightened with a spanner, keeping the camshaft in place.
- Next, when the camshaft is secured in the dividing head it became apparent that when the dial was rotated the camshaft would not rotate with the dividing head and instead would just rotate freely. Therefore, in order to fix this a bolt was screwed into the headstock of the dividing head with nuts and washers to go either side of the camshaft sprocket, then they tightened against one another ensuring that the camshaft only rotates with the dial.
- To ensure the camshaft is set up correctly and horizontally in the dividing head, a DTI gauge was set up and placed on the journal of the camshaft, the dial was then rotated a full 360°, one full revolution of the camshaft. As the journal has a constant radius, providing the camshaft is not on an angle and is completely horizontal in the dividing head, the DTI gauge should read 0mm (+0.02mm Tolerance) throughout the revolution. Both the intake and

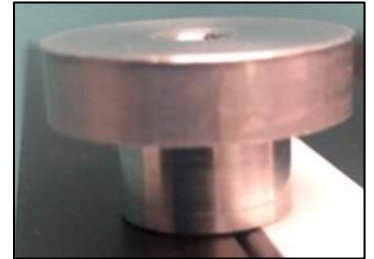


Figure 52 Aluminium End Cap

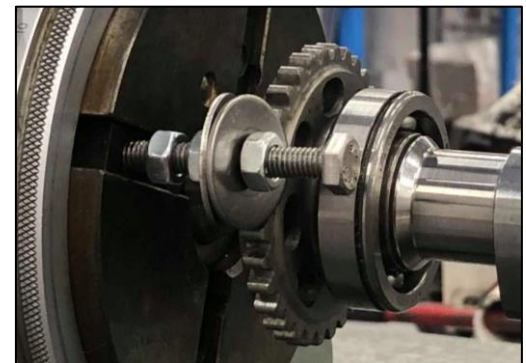


Figure 53 Bolts, Nut & Washer to hold sprocket

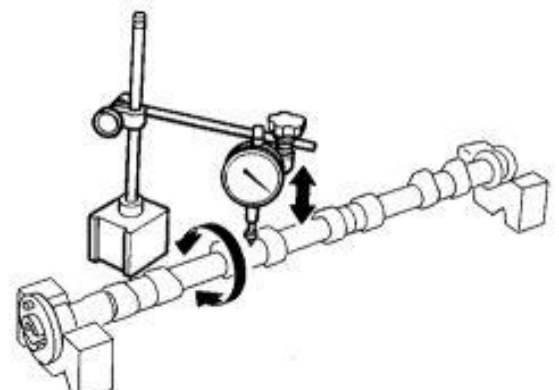


Figure 54 Camshaft Runout Check (AutoZone, n.d.)

exhaust camshafts of the KTM and the YZ250F were within tolerance when tested. This method is usually utilised when checking the runout of a camshaft journal, therefore this test also ensured the runout of the camshaft journals of the KTM & YZ250F were within tolerance.

- Following the validity test, the DTI gauge was set to 0mm and the dial on the dividing head was rotated until it read 0°. The gauge was then placed at the bottom most point of the base circle (*the furthest point from full lift*). The scale displaying the angle of rotation also has a Vernier element for parts of a degree, allowing more intricate measurements to be taken in smaller increments. The fine-tuning dial, which can be seen in Figure 56 was used to rotate the camshaft, recording the lift displayed on the DTI gauge at every degree until the camshaft had completed a full revolution. The results were then inputted into a Microsoft Excel spreadsheet.



Figure 55 Initial Position

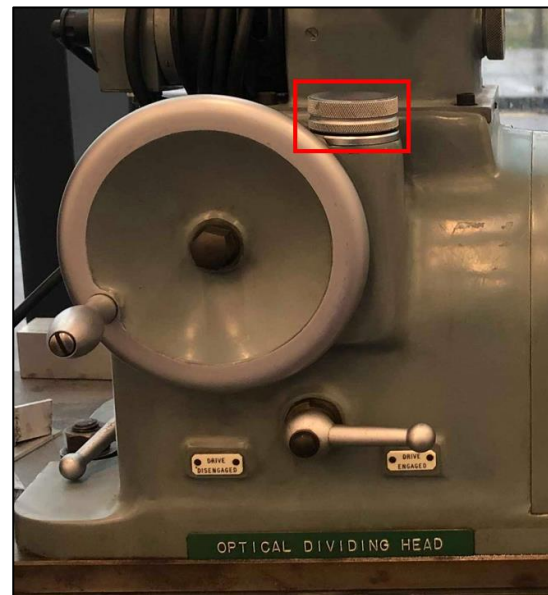


Figure 56 Fine-Tuning Dial

9.6 Valve Lift

9.6.1 Yamaha YZ250F

The valve Lift of the YZ250F for both the intake and exhaust was calculated from the cam lift data using the following methods.

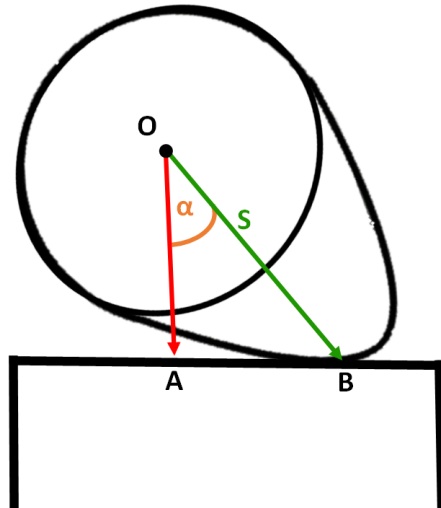


Figure 57 O, S, B & A Cam Diagram (1)

“S” Figure 57 is expressed as;

$$OB = S = (R + L)$$

Equation 4 Base Circle + Cam Lift

Where “L” is cam lift and “R” is the radius of the base circle. For the YZ250F the base circle radius of the intake was 11.25mm and 12.65mm for the exhaust.

“α” is the angle in degrees between the contact point of the cam lobe and bucket-tappet to OA (*Lift*), and is calculated using the following equation;

$$\alpha = 180 - (\theta + \varphi)$$

Equation 5 Alpha

Phi (φ) is the angle in degrees of cam rotation that the nose of the lobe has rotated from the vertical axis, as shown in Figure 58 below.

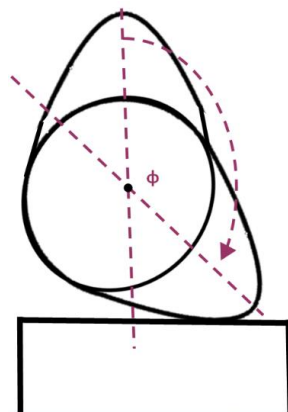


Figure 58 O, S, B & A Cam Diagram (2)

Theta (θ) is the angle in degrees from the nose of the cam lobe to the contact point of the cam and bucket-tappet, as shown in Figure 59 below.

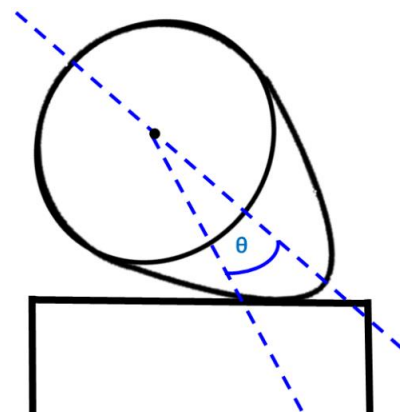


Figure 59 O, S, B & A Cam Diagram (3)

Lift can then be calculated by using equation 6 below;

$$OA = S \times \cos(\alpha)$$

$$\therefore = S \times \cos(180 - (\theta + \varphi))$$

Equation 6 Lift

Equation 6 was then inputted into a 360 by 360 matrix in Microsoft Excel, calculating for every angle of ϕ & θ .

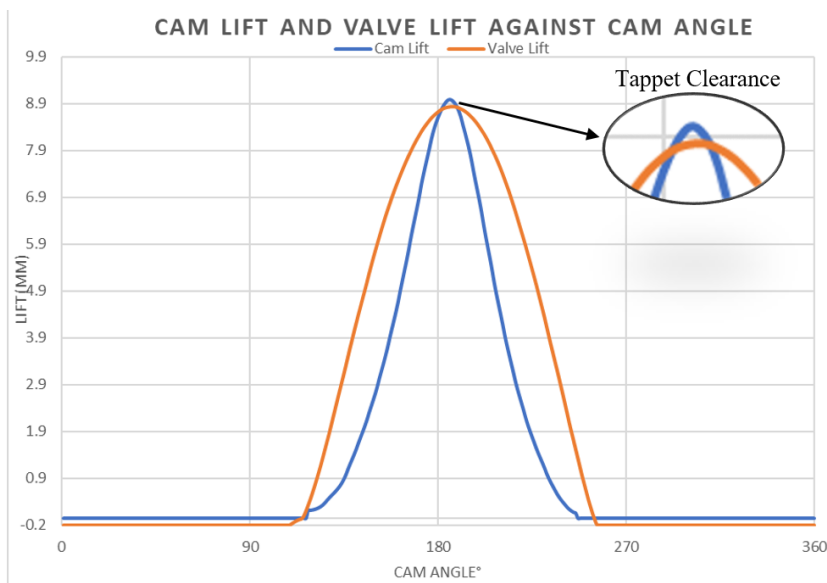
		ϕ						
θ	S	0	1	2		358	359	360
0	S ₀							
1	S ₁							
2	S ₂							
								
358	S ₃₅₈							
359	S ₃₅₉							
360	S ₃₆₀							

Figure 60 360 by 360 Matrix

The subsequent equation was then used to calculate valve lift in mm;

$$\text{Valve Lift} = OA - \text{Tappet Clearance} - R$$

Equation 7 Valve Lift



Graph 2 Yamaha YZ250F Intake Cam Lift & Valve Lift Against Cam Angle

Where OA was the maximum value from each column, allowing valve lift to be calculated for every degree of cam rotation (ϕ). The intake tappet clearance was 0.15mm and the exhaust tappet clearance was 0.22mm (Yamaha, 2017).

When the Cam Lift and Valve lift are plotted together on a scatter graph the tappet clearance can be seen at full lift. Furthermore, when the valve lift displays a negative value before opening and after closing, it is also due to tappet clearance. Graph 2 shows the cam and valve lift for the

intake valve-train this will be analysed in more detail in section 10.1.1, the same graph for the exhaust valve train can be found in Appendix 14.7.

9.6.2 KTM Moto3 RC250R

Figure 61 shows the initial contact point between the cam lobe and the finger follower of the KTM. As the camshaft rotates, the lobe pushes down on the finger follower sliding across its curved surface where contact point is constantly changing. Whereas, the YZ250F's cam lobe slides across the flat surface of the bucket-tappet, therefore the methods used when calculating the valve lift of the YZ250F are not applicable for the KTM.

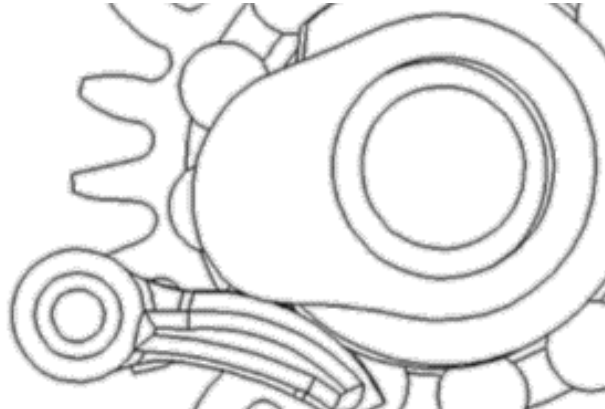


Figure 61 KTM Initial Contact Point

The following steps were taken to determine the valve lift for both the intake and exhaust of the KTM Moto3;

- Firstly, a valve was re-fitted with a lighter spring allowing the camshaft to be turned by hand. Figure 62 shows the lighter spring alongside the intake valve spring.
- Next, a degree wheel was bolted into the thread on the sprocket end of the camshaft, as shown in Figure 63 & Figure 64.



Figure 62 KTM Intake Spring and Lighter Spring



Figure 63 Thread



Figure 64 Degree Wheel

- Then, the cylinder head was placed upside down on top of two steel bars situated on either side of the head, ensuring the camshaft was free so it could be rotated by hand. Subsequently, a G clamp was used to secure the cylinder head.
- A scribe was then attached to one of the steel bars with a jubilee clip, the scribe end was on the degree wheel and was used as a reference point for how many degrees the camshaft had been rotated from the starting point. Additionally, a DTI gauge was angled at the same angle as the valve and placed on the valve head.
- Finally, the camshaft was rotated by hand recording the lift in mm's displayed on the DTI gauge, for every degree until the camshaft completed a full revolution. This method was carried out for both the intake and the exhaust and the result put into a Microsoft Excel Spreadsheet.

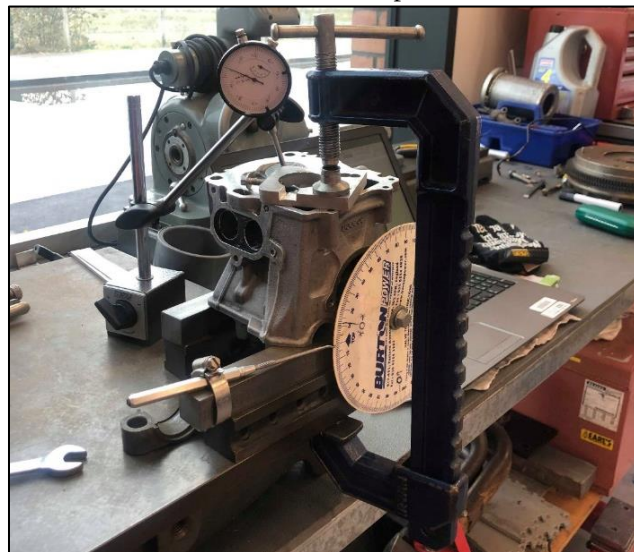


Figure 65 Measuring KTM Valve Lift

9.7 Interpretation of Valve Lift Data

9.7.1 Valve Velocity & Acceleration

From the valve lift data the velocity and acceleration of the valves at every degree of cam rotation was then be calculated. Time derivatives are the standard way of representing instantaneous velocities and accelerations, velocity is defined as the rate at which the position of a body changes with respect to time and, acceleration is the derivative of velocity with respect to time. This can be more simply expressed as;

$$v = \frac{dy}{dt}$$

Equation 8 First Derivative of Distance with Respect to time

$$a = \frac{dv}{dt} = \frac{d^2y}{dt^2}$$

Equation 9 Second Derivative of Distance with Respect to time

To calculate the velocity of the valves in mm/° and the acceleration of valves in mm/°/° the following two equations were used;

$$\text{Valve Velocity} = \Delta \text{Valve Lift}$$

Equation 10 Valve Velocity in mm/°

$$\text{Valve Acceleration} = \Delta \text{Valve Velocity}$$

Equation 11 Valve Acceleration in mm/°/°

The number of cam degrees per second can be expressed as;

$$\begin{aligned} &= \frac{RPM_{\text{camshaft}}}{60} \times 360 \\ \therefore &= 6 \times RPM_{\text{camshaft}} \end{aligned}$$

Equation 12 Cam Degrees Per Second

Where camshaft RPM is half the speed of the crankshaft. Therefore, to calculate the valve velocity in metres per second and valve acceleration in metres per second² the following two equations were utilised;

$$= \frac{(\text{Velocity in mm per degree}) \times (6 \times RPM_{\text{camshaft}})}{1000}$$

Equation 13 Valve Velocity in m/s

$$= \frac{(\text{Acceleration mm per degree per degree}) \times (6 \times RPM_{\text{camshaft}})}{1000}$$

Equation 14 Valve Acceleration in m/s²

9.8 Valve Springs

The valves springs are one of the most crucial components within the ICE valve-train. *“Tasked with the critical job of controlling the valve train”* (Smith E. J., 2019), the valve springs operate in harsh environments and are responsible for keeping the valve closed securely against their seats until the camshaft opens the valve, in return releasing pressure from the spring. The camshaft continues rotating through its revolution releasing the valve, the spring then retracts the valve back to its seat in the head, *“The closing force for the valves is applied by the valve springs, which are also responsible for maintaining contact between the cam lobe and the valve.”* (Bosh, 2002) & (B, 2010) .

Figure 66 shows the Intake Valve Springs of the KTM Moto3 (Left) and the Yamaha YZ250F (Right). There are numerous kinds of valve springs available in the automotive industry such as; Single-Coil, Beehive, Conical, Dual-Coil and even Triple-Coil Springs. Regarding the valve springs used within this paper, the KTM utilises Dual-Coil springs and the YZ250F single-coil, this can be seen in Figure 66.

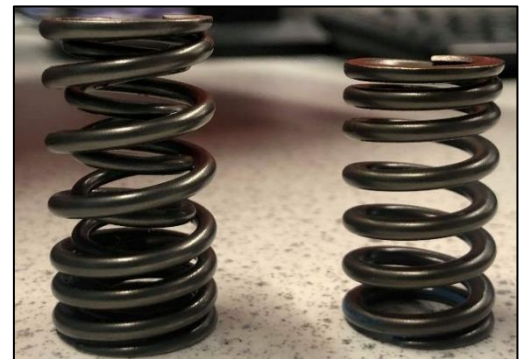
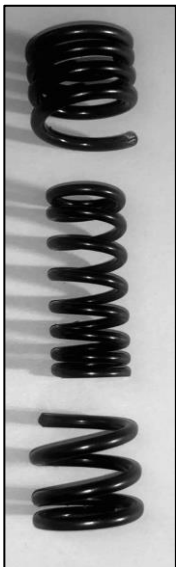


Figure 66 Intake KTM and Yamaha Valve Spring

Single-coil springs are the most commonly found valve springs, especially in engines produced in high volumes. On the other hand, they produce more friction, spring wear and limits valve lift (Summit Racing, 2020). Furthermore, valve float occurs earlier than that of alternative valve springs. Valve float is an adverse condition which occurs when the valves do not properly follow the closure phase of the cam profile. Also, weak valve springs can cause valve bounce which is when the valve hits its seat and then quite literally bounces back off it.



Dual-Coil valve springs are frequently used in high-performance engines and consist of a smaller diameter spring within a larger diameter spring, Figure 67 shows one of the KTM exhaust valve springs broken into two, showing the inner and outer coil. Essentially, the inner and outer springs rub against one another and act as a damper, the second spring also acts as a safety measure, *“broken springs can be found before they drop a valve into the engine”* (Summit Racing, 2020).

Figure 67 KTM Exhaust Valve;
outer and inner coils

9.8.1 Valve Spring Measurement

The valve spring compression tester was provided by workshop technician John Kinsella in the Automotive Workshop. As shown in Figure 68, the valve spring was placed in the tester, the digital Vernier-Caliper was set to 0mm at the point of contact between the tester and the top of the valve spring. The wheel was then turned compressing the spring in increments of 1mm at a time, recording the force being exerted upon the spring as shown on the testers digital screen. This data was inputted into an Excel Spreadsheet, where the force was converted from pounds to newtons by multiplying it by 4.45 and the compression distance into metres by multiplying it by 10^{-3} . This process was repeated twice for the intake and exhaust springs of both the KTM Moto3 and the Yamaha YZ250F to ensure consistency of results and that there was no anomalies.



Figure 68 Valve Spring Tester

9.8.2 k – Spring Constant

In order to complete the friction model, FEA study and to determine the valve float RPM of the valve-trains, the spring constant needed to be considered. This information was attainable from the data acquired whilst measuring the valve springs.

Within Hooke's law " k " is a constant called the spring constant, its S.I unit is N/m. Essentially Hooke's law states that the more a spring is deformed, the more force is required to deform it further. Mathematically, Hooke's law is expressed as;

$$F = k \times x$$

Equation 15 Hooke's Law (AIM Coil, 2020)

Where;

F – Spring Force (N)

x – Displacement of Spring (m)

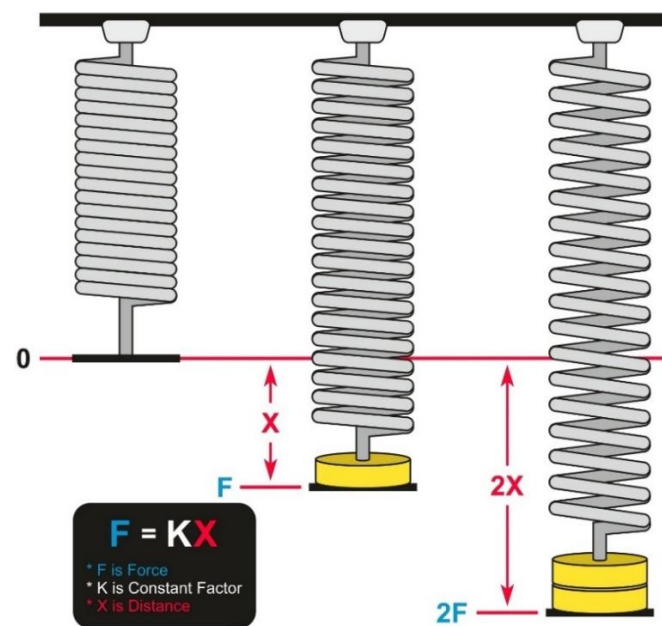
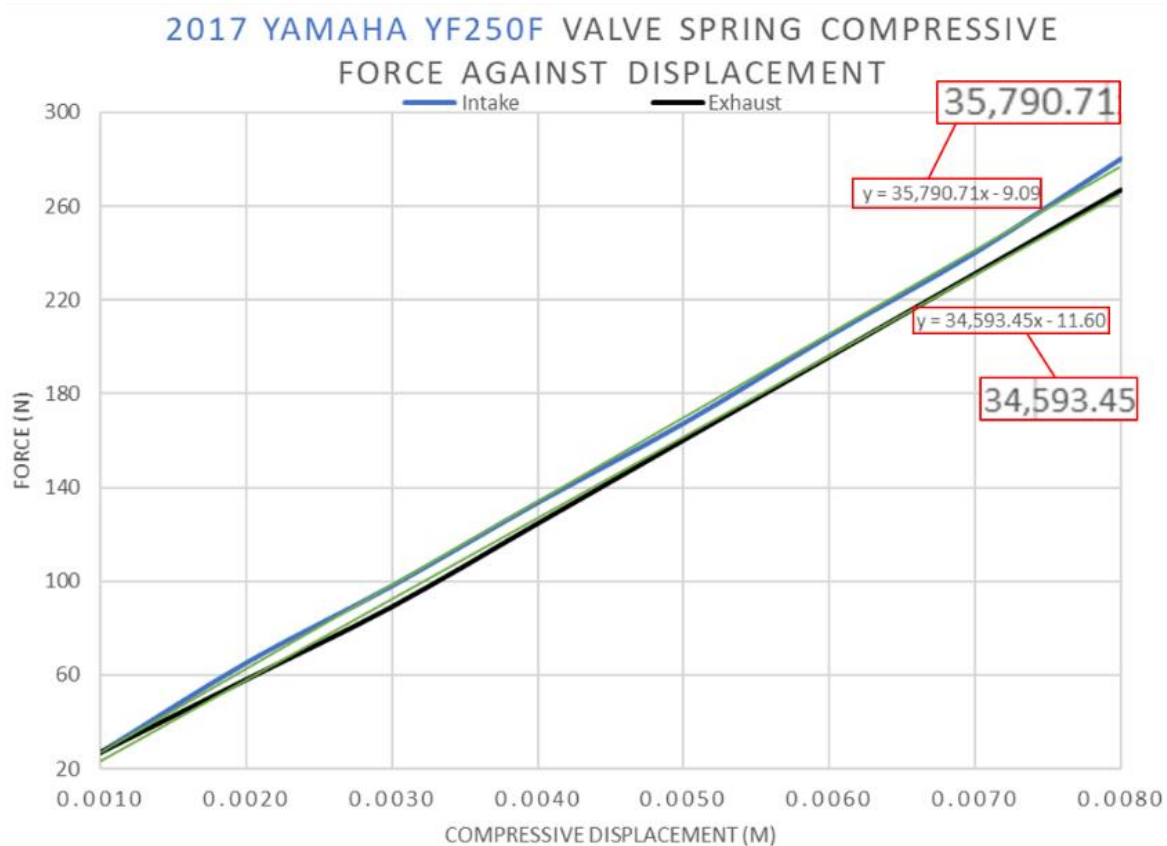


Figure 69 Hooke's Law (AIM Coil, 2020)

The valve spring data of the YZ250F was plotted on a scatter graph, with force on the y axis and displacement on the x axis, with the purpose of finding the spring constant.

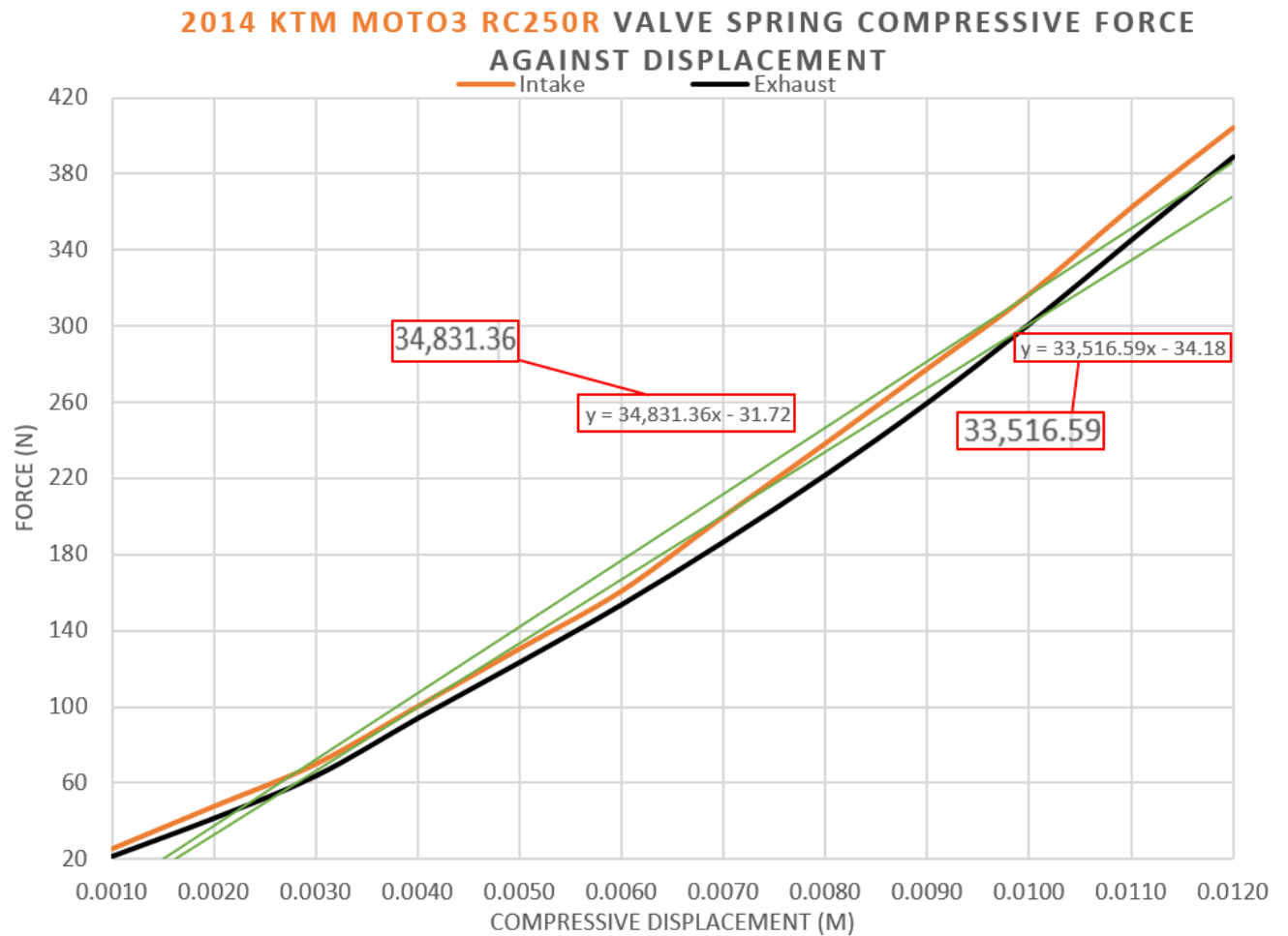


Graph 3 YZ250F Valve Spring Compressive Force Against Displacement

Graph 3 YZ250F Valve Spring Compressive Force Against Displacement Graph 3 shows a clear linear rate of change for the force as compressive displacement increases, “springs that obey Hooke’s law fall into the category of “linear force” springs” (AIM Coil, 2020). To determine the spring rate a linear trendline was applied to both the intake and exhaust, the equation of said trendline was then applied to the graph and the k value was taken from this, as shown in Graph 3.

Not all springs obey Hooke’s law, there are exceptions such as variable pitch springs (KTM). Valve springs with a constant diameter but a variable pitch, such as the springs used in the valve train of the KTM Moto3 (Figure 66) will not display a linear trend when plotted on a graph. “Variable pitch springs are a third example of a spring type that does not obey Hooke’s Law. Variable pitch springs are often compression springs with constant coil diameters, but varying pitch.” (AIM Coil, 2020).

Even though the KTM springs do not obey Hooke’s law a k value needed to be determined in order to be able to complete the friction model and to enable a working motion simulation within SolidWorks. Therefore, an assumption was made to use the same method as the YZ250F.



Graph 4 KTM Moto3 Valve Spring Compressive Force Against Displacement

Whilst the graph shows a non-linear growth, it was considered reasonable enough to use the value from the trendline equation as the spring constant. Table 6 displays the spring constant values in SI units (N/m) of both valve trains.

	K	
	Intake	Exhaust
KTM	34831.00	33954.00
Yamaha	35791.00	34593.00

Table 6 Spring Constant Values

9.8.3 y_o – Initial Seated Compression

As well as the spring constant, in order to successfully complete the friction model and to determine the engine speed at which valve float will occur, the initial seated compression denoted as “ y_o ” needed to be acquired. Figure 70, shows three different heights of a valve spring, A is the free height of the spring, B is the installed height when the valve is closed and finally C is the height of the spring whilst the valve is open.

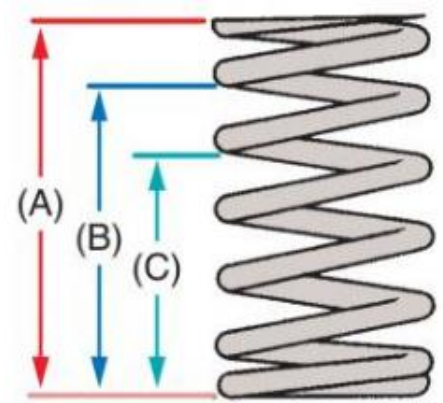


Figure 70 Valve Spring in Sections, A, B & C
(Tim Gilles, 2019)

All of the valve springs were measured with a Vernier-Calliper to determine their free height. Then, the valve-train was reassembled in the KTM Moto3 and the installed height of the springs were measured.

Regarding the YZ250F, the cylinder head was not accessible to the author to be able to reassemble the valve-train. Fortunately, the installed height was found in the owner's manual, as well as the spring free height to verify that the Vernier measurements were reliable, please see Appendix 14.8. Next, the initial seated compression (y_o) was calculated;

$$\text{Initial Seated Compression} = \text{Free Height} - \text{Installed Height}$$

Equation 16 Valve Spring Initial Seated Compression

	Spring Free Height (mm)	Initial seated Spring Compression (mm)	Installed Height (mm)
KTM (Intake)	42.10	8.60	33.50
KTM (Exhaust)	41.40	8.10	33.30
Yamaha (Intake)	37.81	8.68	29.13
Yamaha (Exhaust)	37.54	8.24	29.30

Table 7 Valve Spring Heights

Table 7 shows the results in mm's, this was converted to metres when used within calculations.

9.9 Contact Force

9.9.1 Yamaha YZ250F

Newtons second law states that the acceleration of an object is dependent upon two variables; the force acting upon it and its mass.

$$Force = mass \times acceleration$$

Equation 17 Newtons Second Law

Newtons second law can be applied to the valve-train of the YZ250F to calculate the contact force between the cam lobe and the bucket-tappet, as shown in Equation 18 below.

$$F_c = ma + k(y_o + y)$$

Equation 18 Contact Force

Where;

- **m** = Moving Mass (kg)
- **a** = Valve Acceleration (m/s^2)
- **k** = Spring Constant (N/m)
- **y_o** = Initial Valve Spring Compression (m)
- **y** = Valve Lift (m)

9.9.2 KTM Moto3 RC250R

As previously discussed, the surface of the finger followers are curved, whereas the surface of the bucket-tappets are flat. Therefore the method used to calculate the contact force of the YZ250F was not applicable when calculating the contact force between the cam lobe and finger follower of the KTM.

Firstly, the valve force was calculated, this is the force from the valve spring attempting to return the valve back to its seat. Which is pushing up on the bottom surface of the finger follower, as shown in Figure 72. The equation is the same as that used when calculating the contact force of the YZ250F (*Equation 18*).

$$F_v = ma + k(y_o + y)$$

Equation 19 Valve Force

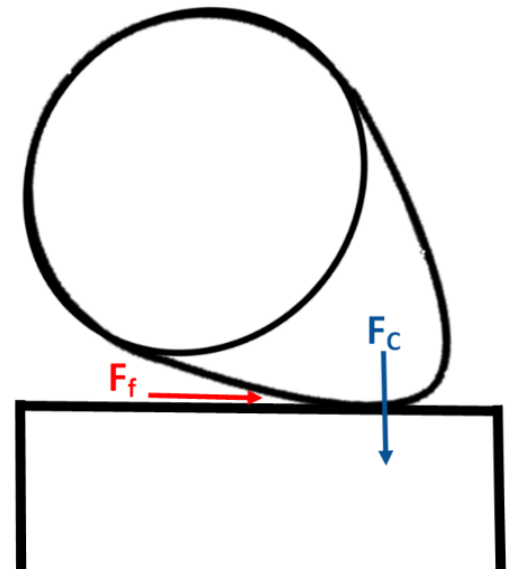


Figure 71 bucket-tappet Contact Force & Friction Force Diagram

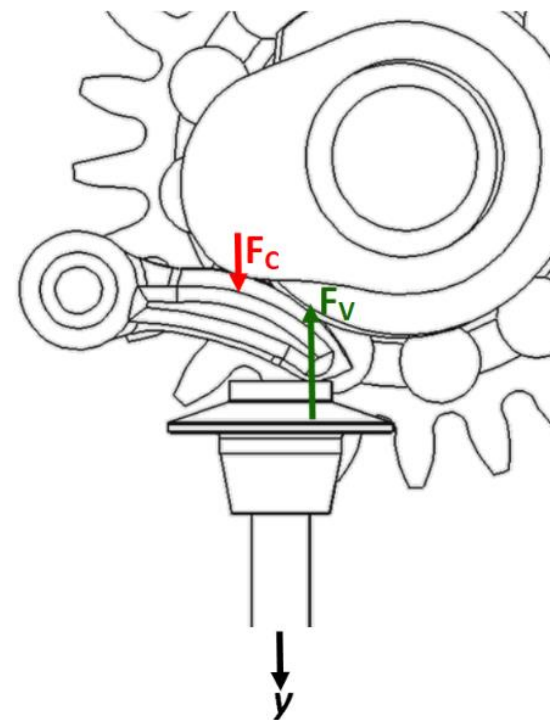


Figure 72 KTM Contact Force & Friction Force Diagram

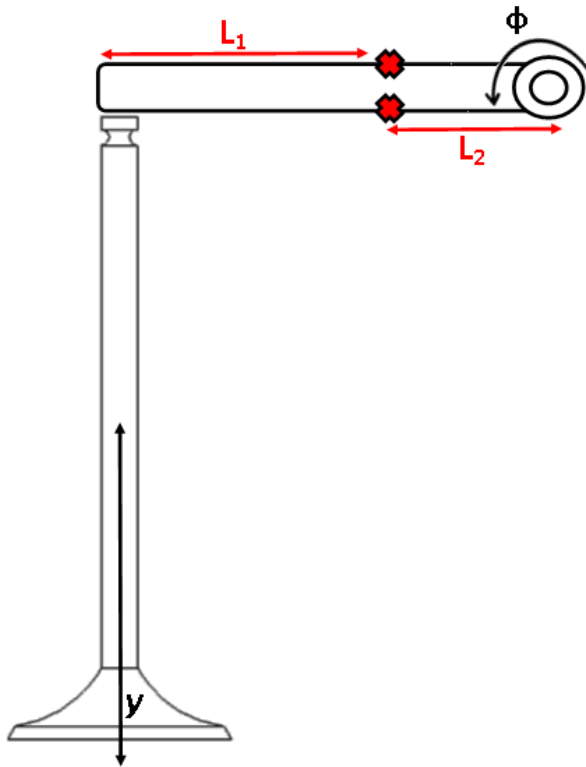


Figure 73 Follower Length 1 & 2 Diagram

Next, the angle at which the follower rotates from its initial position during the camshaft's revolution needed to be determined. Two lengths needed to be attained to calculate the follower angle, length one being the length from the contact point to the nose of the follower and length two being from the contact point to the pivot point of the follower. Due to the curved shape of the finger follower L_1 and L_2 are continually changing as the camshaft rotates. Therefore, with a Vernier these lengths were approximately measured and then the assumption was made to treat the follower as if the surface was flat so L_1 and L_2 stay the same throughout the full 360° of camshaft rotation (Figure 72). $L_1 = 0.017\text{m}$ & $L_2 = 0.014$.

When L_1 & L_2 had been attained, the follower angle (ϕ) through the camshaft's revolution was calculated in radians;

$$y = (L_1 + L_2) \times \phi$$

$$\therefore \phi = \frac{y}{(L_1 + L_2)}$$

Equation 20 Finger Follower Angle

Next, the finger follower velocity (Rad/s) and then the acceleration (Rad/s^2) was calculated using the subsequent two equations;

$$\text{velocity}_{ff} = \Delta\phi$$

Equation 21 Finger Follower Velocity Rad/s

$$\text{acceleration}_{ff} = \Delta\text{velocity}$$

Equation 22 Finger Follower Acceleration Rad/s²

The finger follower's moment of inertia also needed to be calculated, the moment of inertia is the inertia of a rotating body with respect to its rotation. This was attained from the SolidWorks model of the finger follower component (SolidWorks modelling discussed in Section 9.11).

The table displaying the moment of inertia in SolidWorks can be seen in Appendix 14.9. SolidWorks provides the data in g.mm^2 , it was then multiplied by 10^{-9} as it needed to be in kg.m^2 for use in the contact force equation.

Moment of Inertia	
117.9894	g.mm^2
1.18E-07	Kg.m^2

Table 8 Finger Follower Moment of Inertia

Next, the contact force between the cam lobe and the finger follower was calculated using Equation 23.

$$F_c = F_v \times \frac{(L_1 + L_2)}{L_2} \times \frac{I}{L_2(L_1 + L_2)} \times acceleration_{ff}$$

Equation 23 Contact Force KTM Finger Follower

9.10 Friction Model

When the contact force had been calculated, the friction force of the cam-lobe sliding across the surface of the bucket-tappet/finger follower as shown in Figure 71 & Figure 72 was calculated using Equation 24.

$$F_f = \mu \times F_c$$

Equation 24 Friction Force

μ is the coefficient of friction of the cam lobe sliding across the bucket-tappet/finger follower which as previously stated is 0.16 for the YZ250F and 0.05 for the DLC coated finger follower.

Torque can be calculated as;

$$\tau = Force \times Distance$$

Equation 25 Torque

In order to calculate the frictional torque in the valve train the friction force (*Newtons*) was multiplied by *S* (*Base Circle Radius + Lift*) in metres.

$$\tau_f = F_f \times S$$

Equation 26 Friction Torque

When the frictional torque of the valve-train was attained the work done per degree was calculated, followed by the overall work done per full camshaft revolution.

$$W.D \left(\frac{Torque_1 + Torque_2}{2} \right) \times \frac{2\pi}{360}$$

Equation 27 Work Done Per Degree

$$Total W.D = \sum W.D \text{ per Degree}$$

Equation 28 Total W.D

Finally, the power losses of the valve-train due to the friction between the cam-lobes and the tappets/followers could be calculated as;

$$Power\ Loss = 2 \times \left(Total\ W.D \times \frac{RPM}{2 \times 60} \right)$$

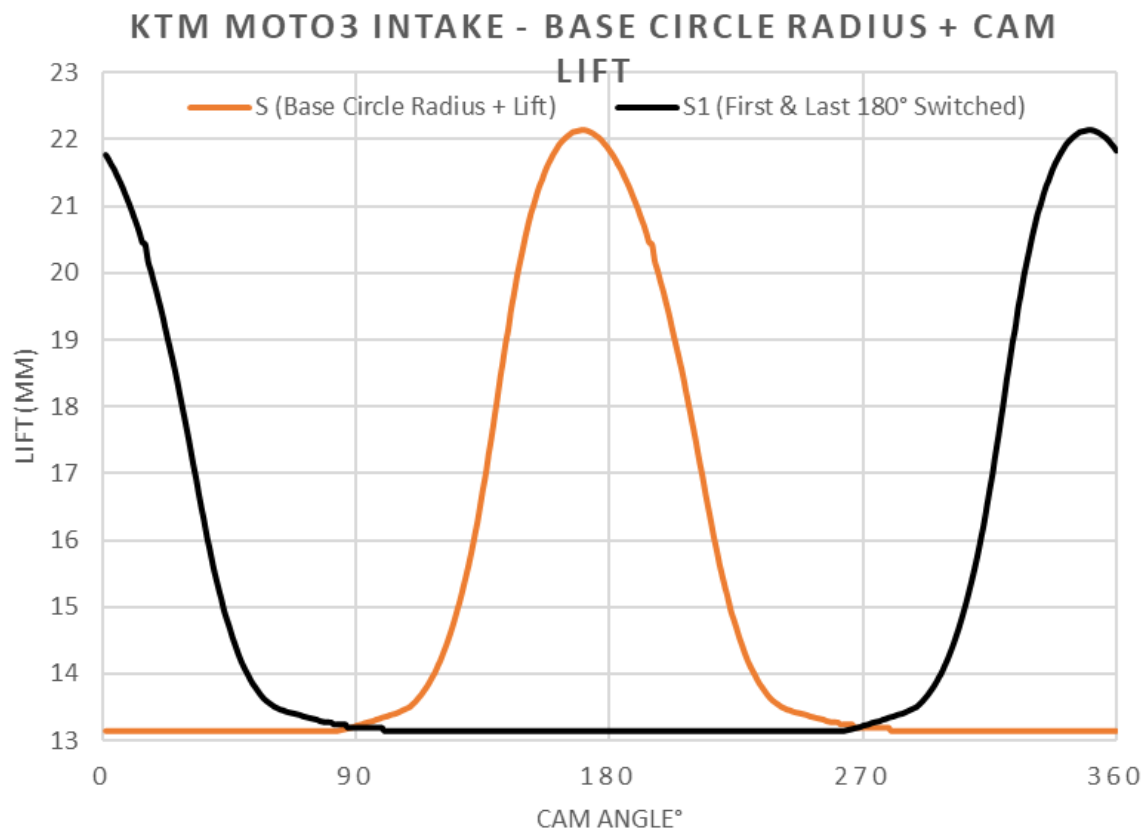
Equation 29 Power Loss

The multiplication of two within Equation 29 Equation 29 Power Loss represents the amount of cam-lobes on the camshaft, both the YZ250F and Moto3 have 2 lobes per camshaft on account of them being single cylinder engines.

9.11 CAD Modelling

9.11.1 Cam Profile

In order to accurately replicate the cam profile of both intake and exhaust camshafts for both engines within SolidWorks, the cam lift data was manipulated to generate X & Y co-ordinates. Foremost, the first and second 180° of S (*Base Circle + Lift*) were switched in place, this was denoted as S₁ as shown in Graph 5, displaying the S and S₁ values of the intake camshaft of the KTM.



Graph 5 KTM Moto3 Intake - Base Circle Radius

Furthermore, the S₁ values were used within the following trigonometric functions in order to plot a graph within excel displaying the shape of the cam profile.

$$y = A \sin(Bx)$$

Equation 30 Sinusoidal Sine Function

$$y = A \cos (Bx)$$

Equation 31 Sinusoidal Cosine Function

The A in front of the Sine or Cosine function affects the amplitude and B is the number of cycles the graph completes in an interval from 0 to 2π , or in the case of this study 0 to 360° of rotation. These functions were applied to the cam data in order to determine the X & Y co-ordinates, using the following two equations where ϕ is the angle of the camshaft in radians.

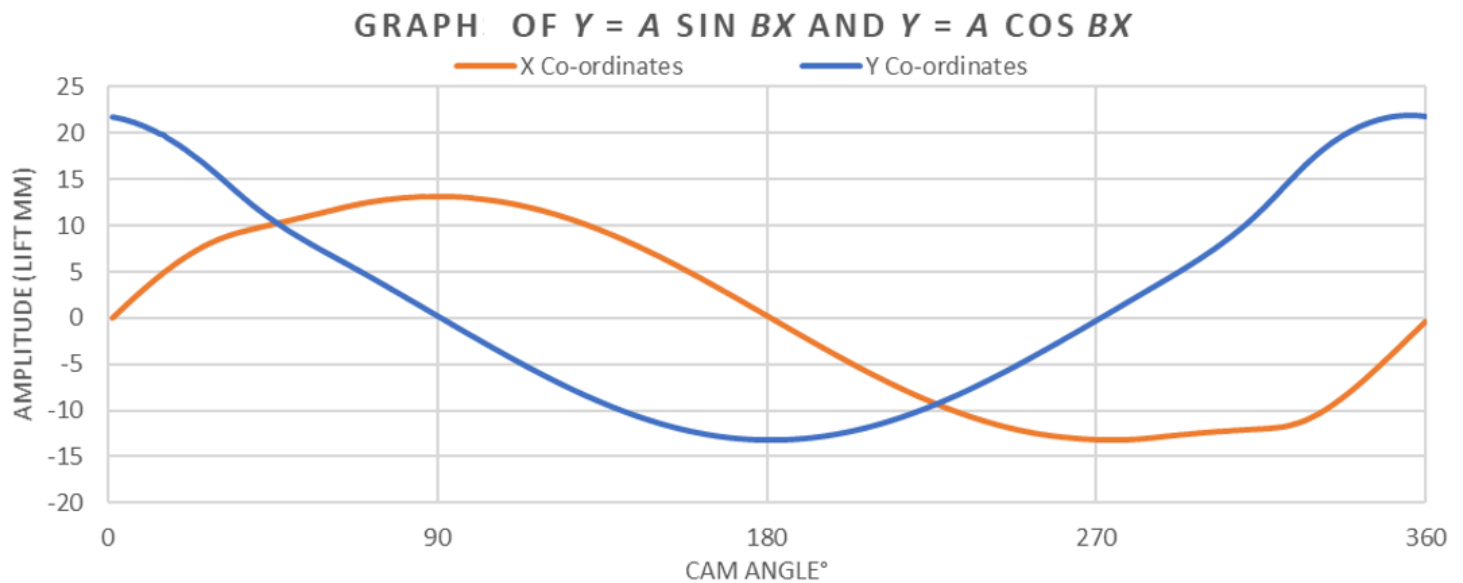
$$X = S_1 \times \sin(\phi)$$

Equation 32 X co-ordinates

$$Y = S_1 \times \cos(\phi)$$

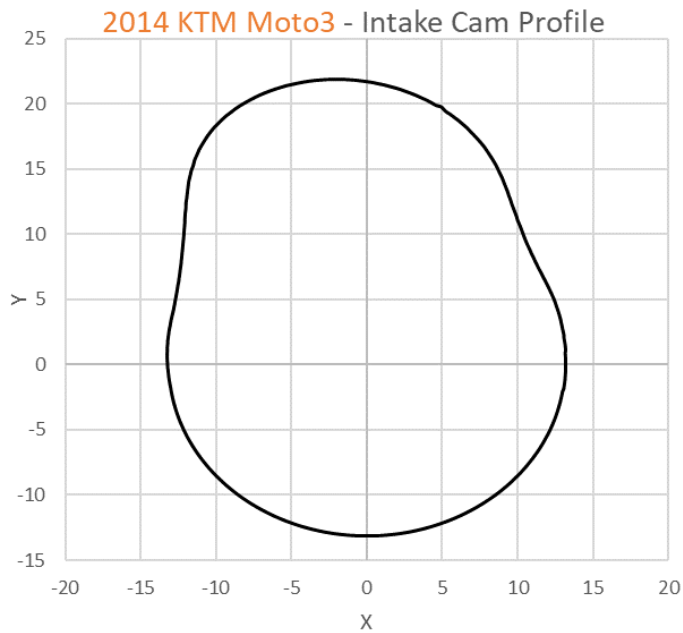
Equation Y co-ordinates

Graph 6 shows the X & Y values plotted at each degree of camshaft rotation for the intake of the KTM Moto3.

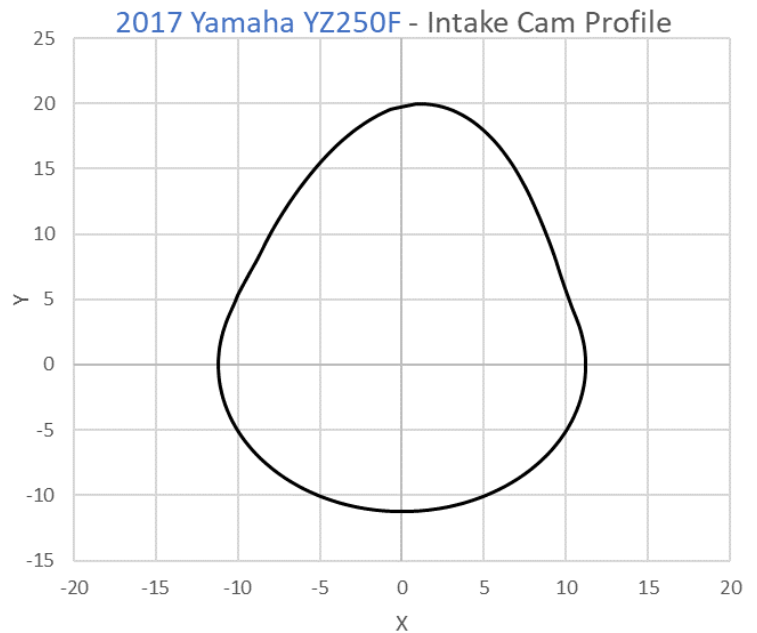


Graph 6 Graphs of $y = A \sin Bx$ & $y = A \cos Bx$

Next, the X and Y data was plotted on scatter graphs within Excel to ensure they represented the correct shape of the cam lobes before being inputted into the SolidWorks, Graph 7 & Graph 8.



Graph 7 KTM Moto3 - Intake Cam Profile



Graph 8 Yamaha YZ250F - Intake Cam Profile

The same graphs regarding the exhaust camshafts of both engines can be found in Appendix 14.11. The X & Y data was inputted into SolidWorks using the Curve function, generating a sketch of the cam-profile, of which the component was then modelled around.

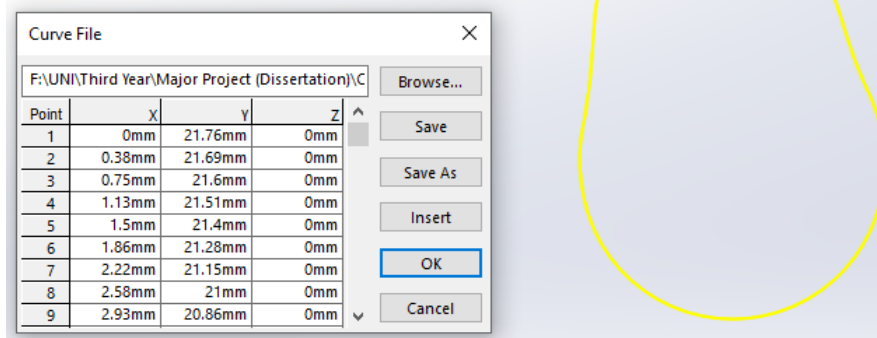


Figure 74 SolidWorks Curve File Tool

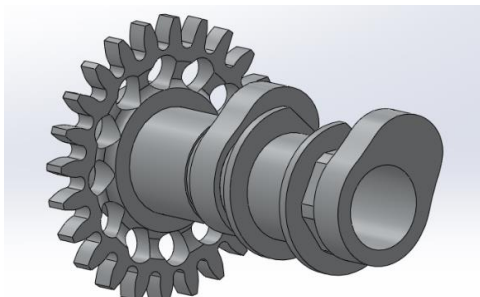


Figure 75 KTM Moto3 Intake Camshaft - SolidWorks Model

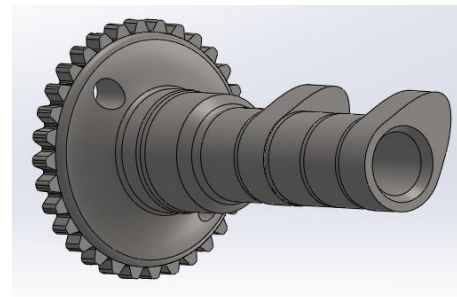


Figure 76 Yamaha YZ250F Intake Camshaft - SolidWorks Model

9.11.2 DLC Coated Cast Steel Material

The steel finger followers of the KTM Moto3 had a thin-film DLC coating. As previously stated in Section 8.2.2, the form of DLC coatings used within high-demanding environments such as ICE valve-trains are in ta-C form. Therefore, the assumption was made that the finger follower used a ta-C form DLC coating with a thickness of 3 microns (μm). Thin-film coatings are not available within the SolidWorks package; therefore a custom material was created. Research was conducted to discover the mechanical properties of thin-film DLC coatings, please refer to 14.10 (Sung-Jin Cho, 1999) & (Davidson, 2000). The available Cast Steel SolidWorks material was selected and then modified to suit that of DLC coated cast steel, this was saved as a new material and applied to the finger follower, as shown in Figure 77.

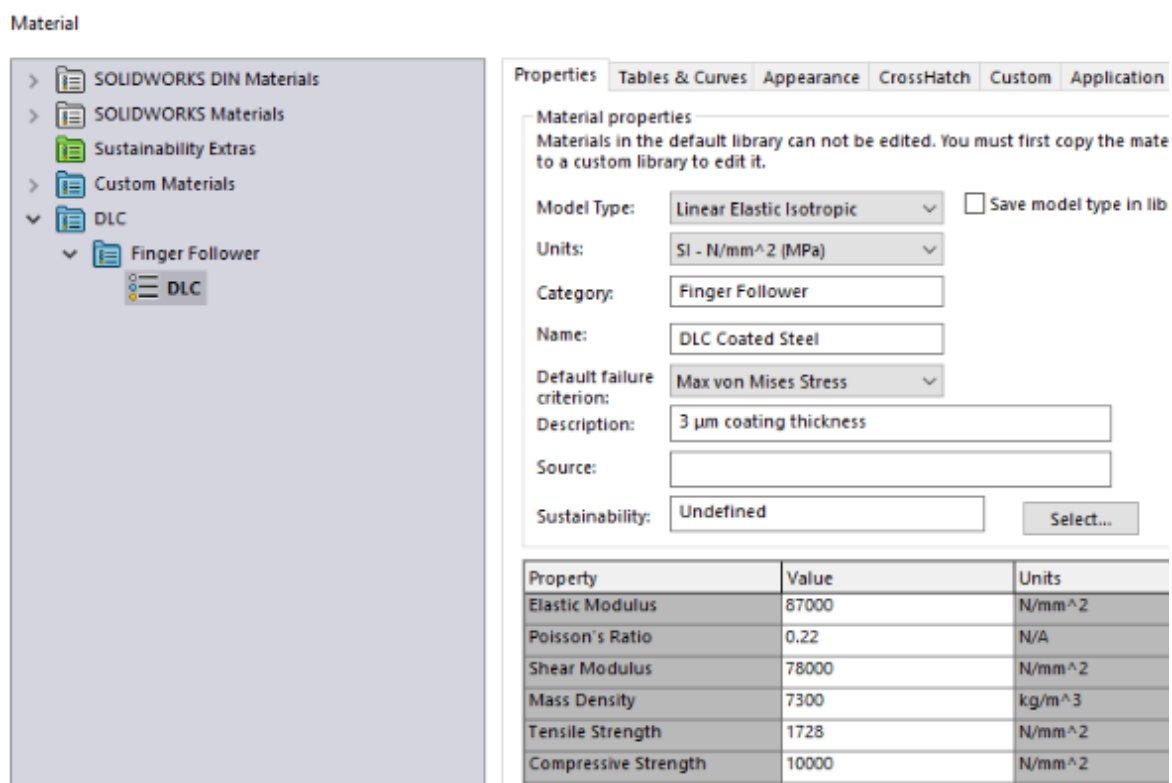


Figure 77 DLC Coated Steel SolidWorks Material

9.11.3 Motion Simulation

Once all the components of each valve-train within this study had been modelled they were compiled into assemblies, Figure 78 & Figure 79 shows the intake assemblies. This was done using a method of mating all the required components to one another, mechanical mates were used when applicable to reduce redundancies.

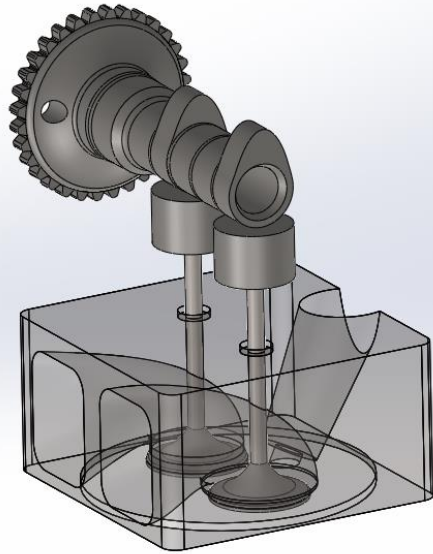


Figure 78 YZ250F Intake SolidWorks Assembly Model

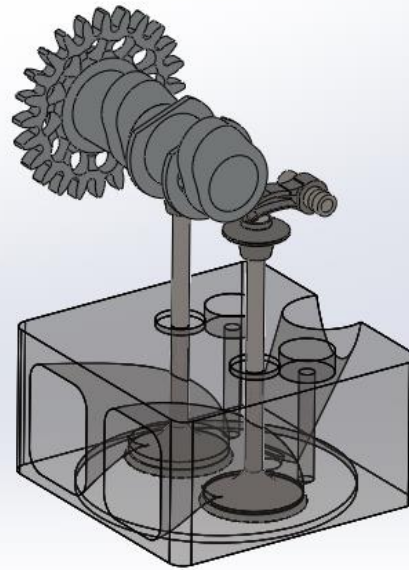


Figure 79 KTM Moto3 Intake SolidWorks Assembly Model

In order to carry out an FEA study, a working motion simulation of each valve-train was required. Firstly, a rotary motor was applied to the camshaft running at a constant speed, these were set to maximum RPM for each valve-train, as the purpose of the FEA simulation was to study the forces being exerted onto the finger followers/bucket-tappets during maximum engine speed. As previously stated, the KTM is limited to 13,500RPM and the maximum engine speed of the YZ250F is 12,900RPM. Therefore, the rotary motor for the KTM was set to 6,750RPM and 6,450RPM for the YZ250F as;

$$\text{Camshaft Speed} = \frac{\text{Crankshaft Speed}}{2}$$

Equation 33 Camshaft Speed

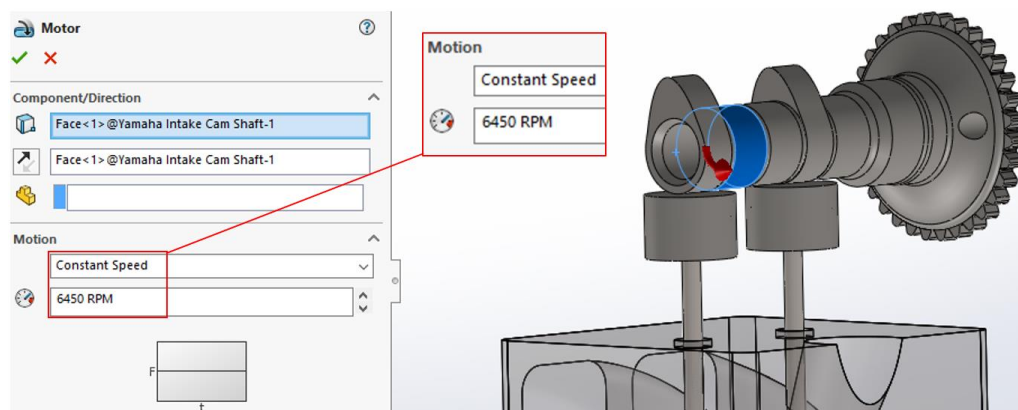


Figure 80 YZ250F Intake - Rotary Motor

The motion simulation consists of one camshaft revolution, this was all that was needed to simulate the forces exerted onto the finger follower/bucket-tappet. Furthermore, only having a motion simulation consisting of one revolution saves on computational time compared to numerous unnecessary revolutions, the KTM only needed 0.0089 seconds to complete one full revolution and 0.0093 for the YZ250F.

KTM RC250R Moto3

$$\text{Revolutions Per Second} = \frac{6,750}{60}$$

$$\therefore = 112.5$$

Equation 34 KTM Revolutions Per Second

$$\text{Time Taken for Per Revolution} = \frac{1}{112.5}$$

$$\therefore = 0.0089s$$

Equation 35 KTM Time Taken for Per Revolution

Yamaha YZ250F

$$\text{Revolutions Per Second} = \frac{6,450}{60}$$

$$\therefore = 107.5$$

Equation 36 YZ250F Revolutions Per Second

$$\text{Time Taken for Per Revolution} = \frac{1}{107.5}$$

$$\therefore = 0.0093s$$

Equation 37 YZ250F Time Taken for Per Revolution

Subsequently, the number of frames required within SolidWorks for each valve-train could be calculated using the revolutions per second data calculated in Equation 34 & Equation 36.

KTM RC250R Moto3

$$\text{Frames Needed} = 112.5 \times 360$$

$$\therefore = 40,500$$

Equation 38 KTM Frames Needed

Yamaha YZ250F

$$\text{Frames Needed} = 107.5 \times 360$$

$$\therefore = 38,700$$

Equation 39 YZ250F Frames Needed

Next, the spring tool was utilised to input the valve spring data, the k values determined earlier on within this paper were entered. Gravity was added to the model with a rate of acceleration of 9806.65 mm/s², ensuring validity of results.

The contact points within the assemblies needed to be modelled; lobe to finger follower/bucket-tappet, finger follower to shim and valves to cylinder head, so that when the valve spring retracts the valves they stop when they contact the valve seats. These contact points were modelled with the use of the Solid Body Contact function, which also allowed the friction coefficients previously discussed to be applied to the model.

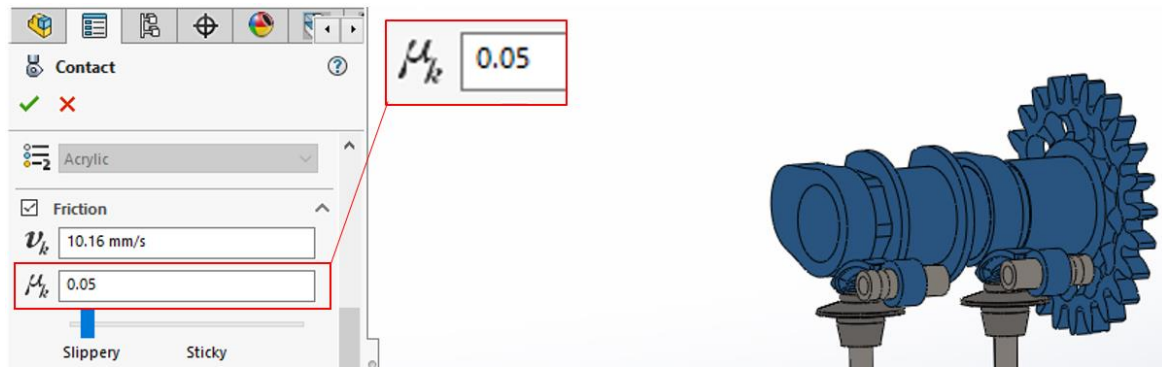


Figure 81 KTM Intake - Solid Body Contact Between Lobes & Finger Follower

The motion simulations could now run effectively, these methods were carried out for each valve-train considered within this study.

9.12 Finite Element Analysis

Once a working motion simulation for each valve-train had been established, the forces acting upon components within the simulation were exported out to calculate their stresses and displacements. This was done with the use of the Import Motion Loads function available in SolidWorks Simulation, allowing the user to generate a design scenario for each chosen time frame populated with motion loads developed at any given frame.

Once this method was completed for both the KTM finger follower and the YZ250F bucket-tappet, the individual components were opened, and scenarios were run. Upon completion; it was clear that the forces exerted onto the finger follower/bucket-tappet from the cam-lobe and the force from the valve pushing back up on the KTM 's finger follower had not been exported from the motion loads function. Only gravity, centrifugal and remote loads were exported as shown in Figure 83 & Figure 84.

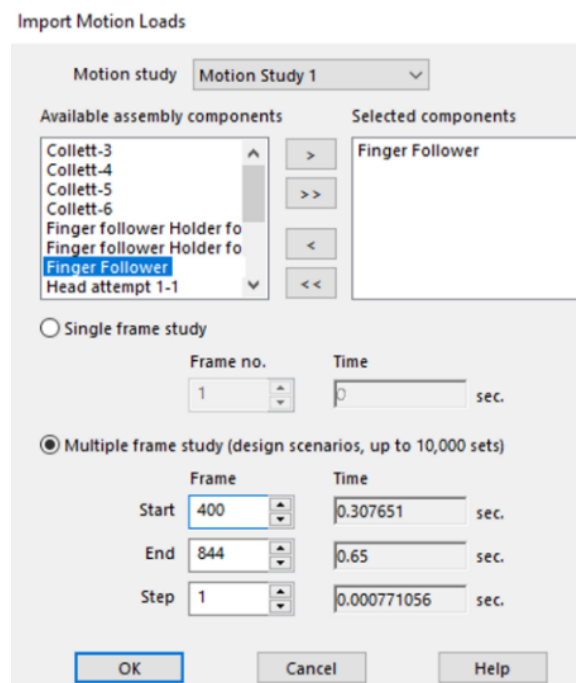


Figure 82 Import Motion Loads Function SolidWorks

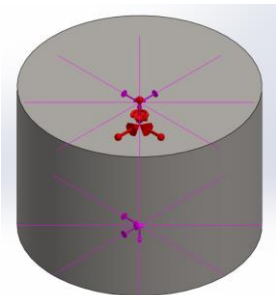


Figure 83 Bucket-Tappet Initial Forces

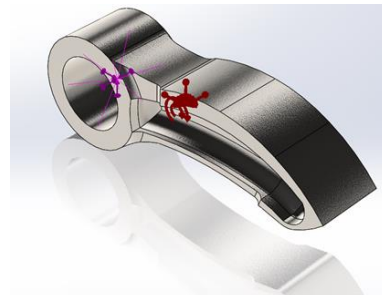


Figure 84 Finger-Follower Initial Forces

To rectify this problem for the YZ250F, the highest contact force at maximum engine speed was taken from the Excel Spreadsheet (*As previously calculated using Equation 18*) and applied directly acting upon the top of the bucket-tappet. This method would not work for the finger followers within the KTM valve-train as the contact point between the lobe and follower is continuously changing across the curved surface, meaning it would be inaccurate to apply the maximum force to the entirety of the top plane surface of the finger follower, as maximum force is only applicable for one section of the curved surface. This being the cam-lobes position on the finger follower during maximum contact force, this area was determined from studying the motion simulation, the surface was then split, allowing forces to be applied specifically upon that area as opposed to the entirety of the top surface. This method was then repeated for the contact point between the shim and follower so the opposing valve force against the finger follower could also be added to the model ensuring accuracy.

The contact force of the force acting upon the YZ250F was 826.12N & 1609.25N for the KTM with an opposing valve force of 726.76N, these values were to be validated, but unfortunately couldn't due to unforeseen circumstances, discussed further in section 12.

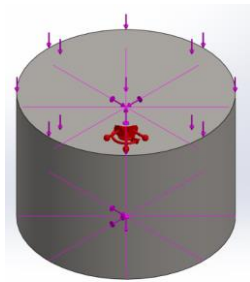


Figure 85 YZ250F All Forces Applied

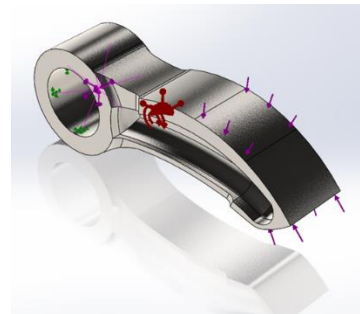


Figure 86 KTM Finger Follower All Forces Applied

The Import Motion Loads function was used again to evaluate each scenario, but this time the contact forces were applied within the FEA study. Furthermore, regarding the KTM finger follower the inner diameter at the pivot points was fixed in space so that the geometry was a fixed constraint. Finally, a mesh was applied, and the studies could be run. The refinement mesh was made as fine as SolidWorks would allow in order to calculate the greatest accuracy in results. A coarser mesh divides the component into larger elements, whereas the finer mesh subdivides the component into much smaller elements.

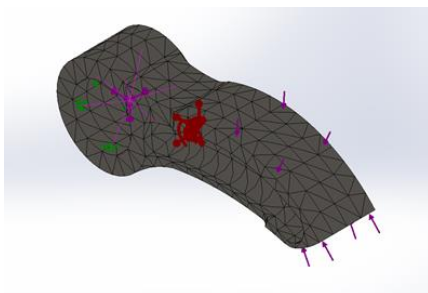


Figure 87 KTM Finger Follower Coarse Mesh

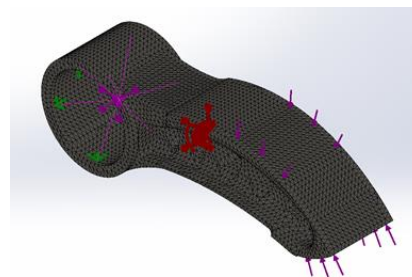


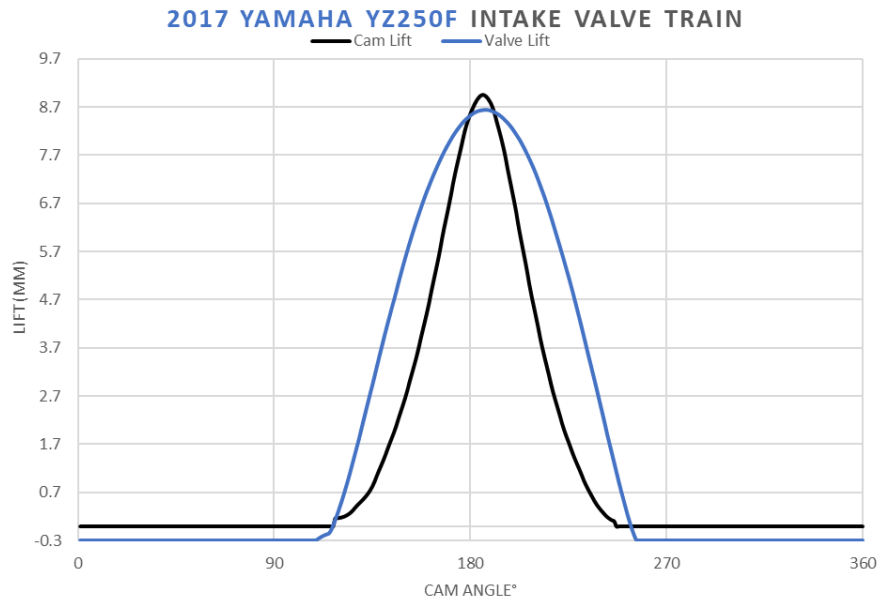
Figure 88 KTM Finger Follower Fine Mesh

10 Results & Analysis

10.1 Valve-Train

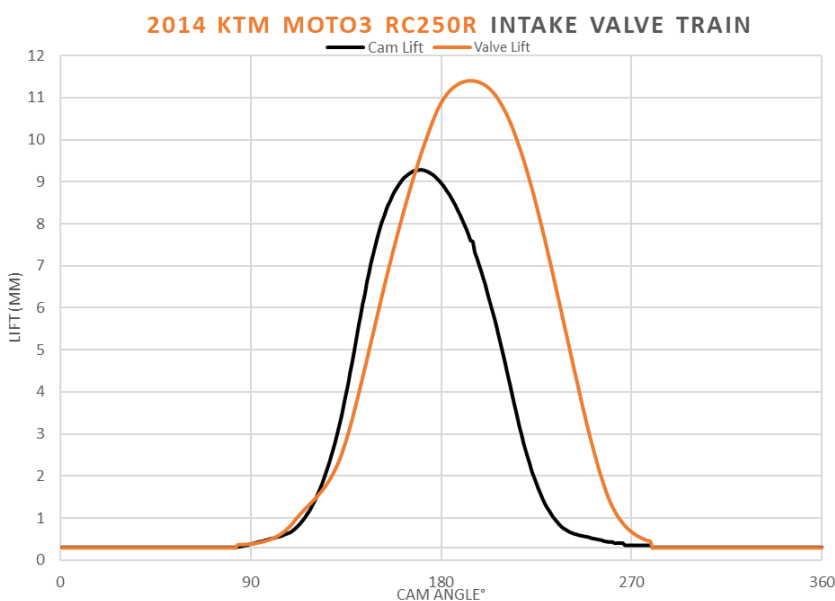
10.1.1 Cam & Valve Lift

As previously discussed, the camshaft used within the valve-train of the YZ250F is “Symmetrical” in shape meaning that the opening and closing ramps of the cam lobe are mirror images of one another, this is clearly demonstrated in Graph 9 where the pattern of the opening and closing phases are identical. Moreover, as stated in section 9.6.1, there is 0.3mm tappet clearance which can be seen at maximum lift and when there is no lift, explaining the -0.3mm.



Graph 9 2017 YZ250F Intake Valve Lift

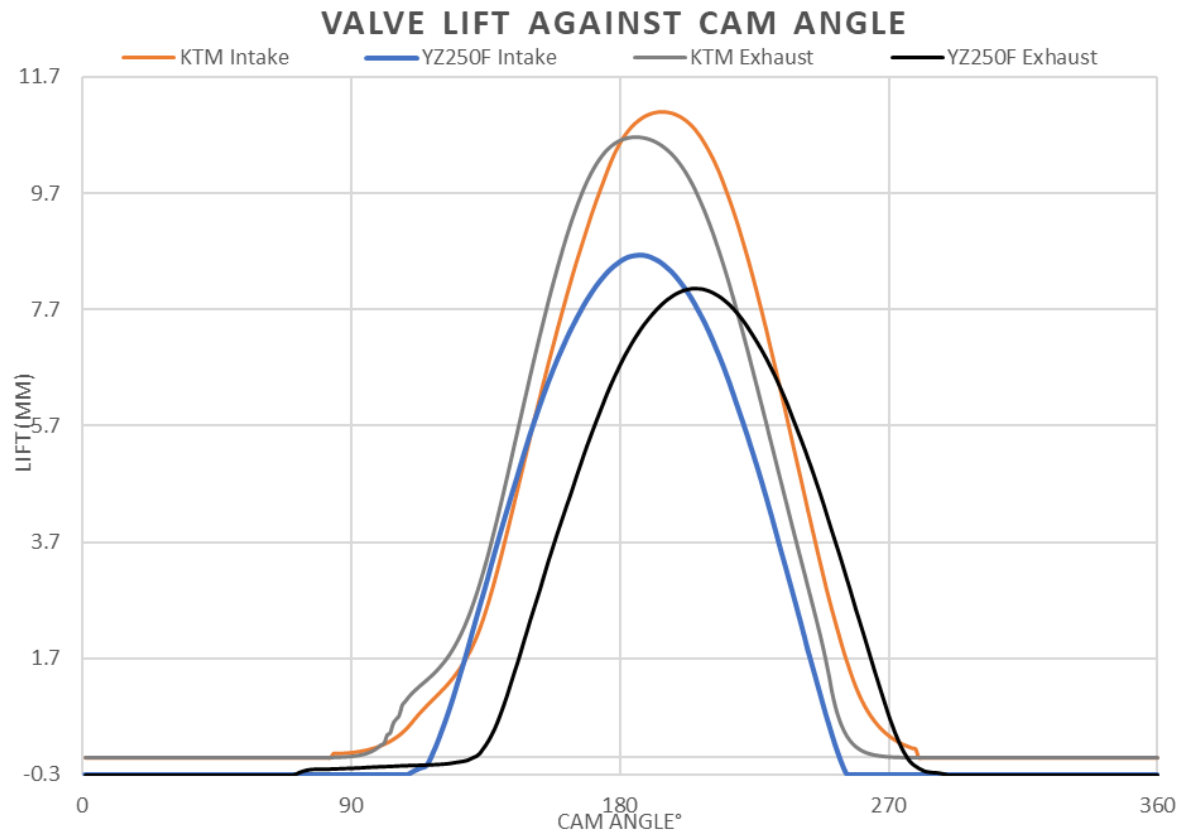
Graph 10 shows that whilst the duration of cam and valve lift are the same, the valve lift is higher for longer, an assumption was made that this was an effect of the finger follower. It is worth mentioning that the author intended to remeasure both the cam and valve lift of both the intake and exhaust to check the validity of these results, but was unable to due to unforeseen circumstances, this is discussed further in section 12. Additionally, unlike the YZ250F there is a greater



Graph 10 KTM Intake Valve Lift

difference between cam and valve lift at their maximum values, this once again is a result of the finger follower actuating system as opposed to bucket-tappet. The maximum valve lift of the intake valve-train was 2.12mm higher than the cam lift, and 1.92mm higher in the exhaust valve-train. Furthermore, as previously discussed in the literature review the camshaft used within the valve-train of the KTM is

asymmetric, when reviewing the results this was made clear. The closing duration of the intake camshaft is 22.9% longer than the opening phase and 18.06% for the exhaust, thus the valves open at a quicker rate than they close. The aim is to yank the valve off its seat to “*accelerate it over the nose as quickly as possible*”, whilst reducing velocity during the closing ramp in order to avoid “*destabilizing valve bounce*” (Siczek, 2016), this was further proven when analysing the results as it was found that the velocity of the intakes opening phase was 20% faster than that of the closing. The same graphs regarding the exhaust valve-trains of both engines can be found in Appendix 14.13 & Appendix 14.7

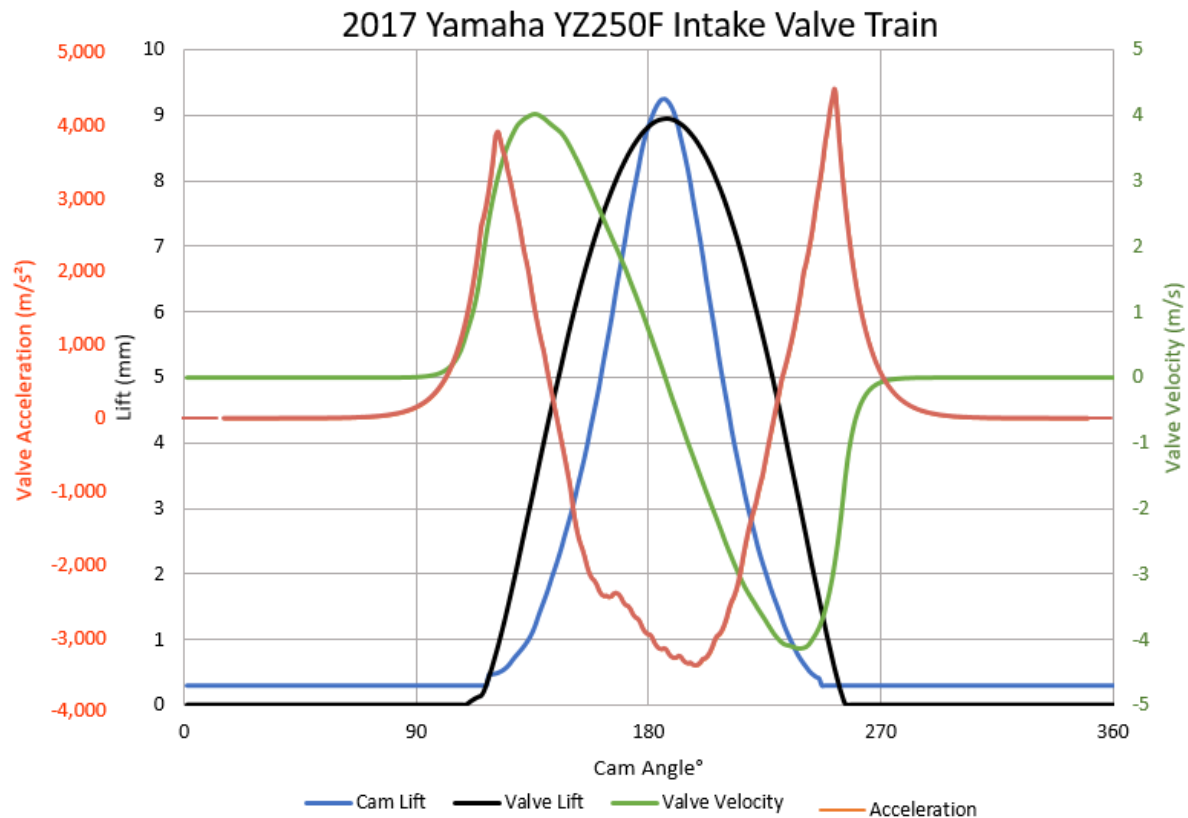


Graph 11 Valve Lift Against Cam Lift

As previously discussed during section 8.1.4.2, the finger follower actuating system used within the KTM allows for a more aggressive cam profile and in turn higher valve lift. This is shown in Graph 11 where the valve lift of both intake and exhaust valve-trains of the KTM are considerably higher than that of the YZ250F. The KTM which is used within road racing will favour higher valve lift for more top-end power, and as previously stated a Moto3 rider will be at full power for round 70% of the time, whereas an Enduro rider using the YZ250F will never be on full throttle for any near the same amount of time as a Moto3 rider “*To gain low-end torque you want a shorter duration, however for top-end power you need a higher lift*” (Azuma-san, 2018).

10.1.2 Valve Velocity & Acceleration

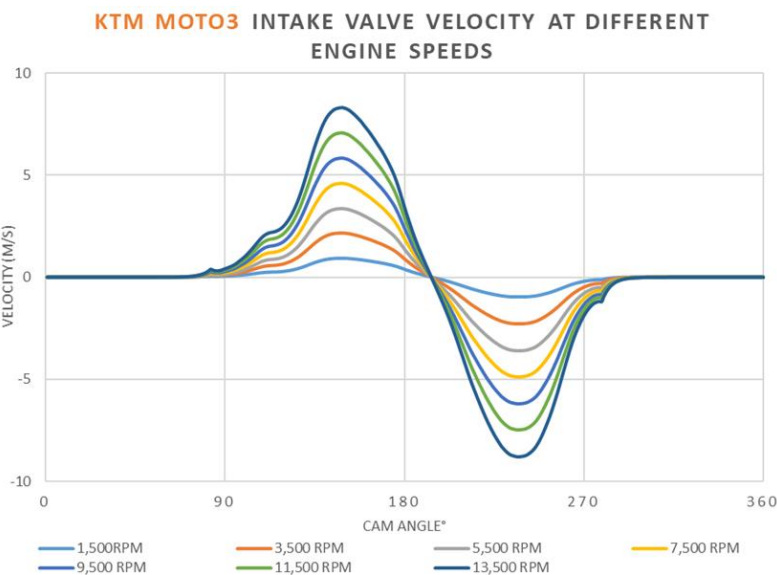
As previously stated, velocity and acceleration are the first and second derivatives of displacement with respect to time, the third being jerk and then jounce and it goes on even further, “*These higher derivatives become increasingly esoteric*” (Tilden Technologies, 2007). The most important derivatives regarding valve-train dynamics are valve velocity and acceleration. Furthermore, it is worth mentioning that jerk also has influence on the valve-trains dynamics but is of lesser importance, thus it was not included within this study.



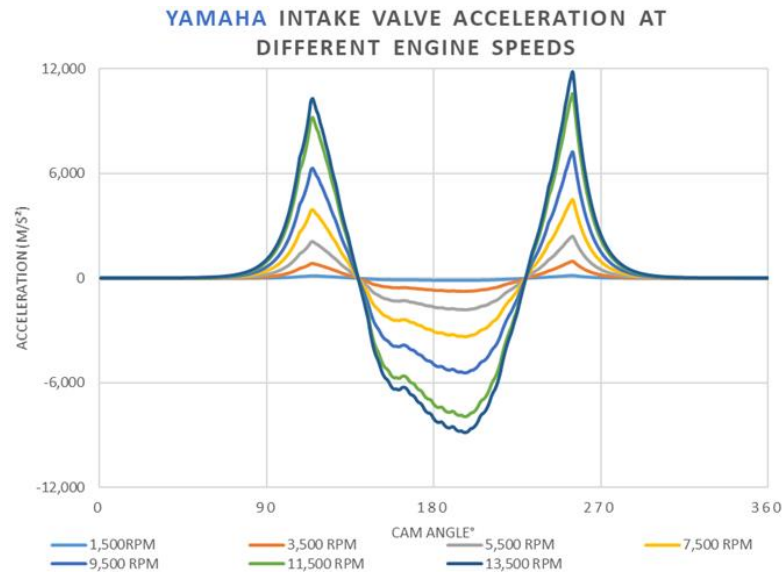
Graph 12 YZ250F Intake Valve Train

Graph 12 displays the lifts, velocity and acceleration of the intake valve-train of the YZ250F. As the opening phase begins, the velocity increases at a rapid rate as does the acceleration. Then, as the valve approaches maximum opening point the velocity begins to decrease causing rapid deceleration, this is clearly shown in Graph 12. When the valve is at maximum opening point the velocity is at 0 m/s^2 and the acceleration is at its lowest point. Furthermore, as the valve then begins to return to its seat the velocity shows the same trend as it did during opening but moving in the opposite direction as the valve is also now moving in the opposite direction. As the camshaft of the YZ250F is symmetric the velocity and acceleration should show an identical pattern during the opening and closing phase. From reviewing Graph 12 it was made apparent that the opening and closing phase of the YZ250F's valve-train were

not quite symmetric, the valve acceleration during the closing ramp is clearly higher than that of the opening. As a result of this the credibility of this data is questionable, therefore the author intended to remeasure the cam profile of the YZ250F to see if an error had been made, unfortunately once again due to unforeseen circumstances this was not possible, thus an assumption was made going forward that the results were valid. The same graph regarding the intake valve-train of the KTM Moto3 can be found in Appendix 14.14.



Graph 13 KTM Moto3 Intake Valve Velocity at Different Engine Speeds

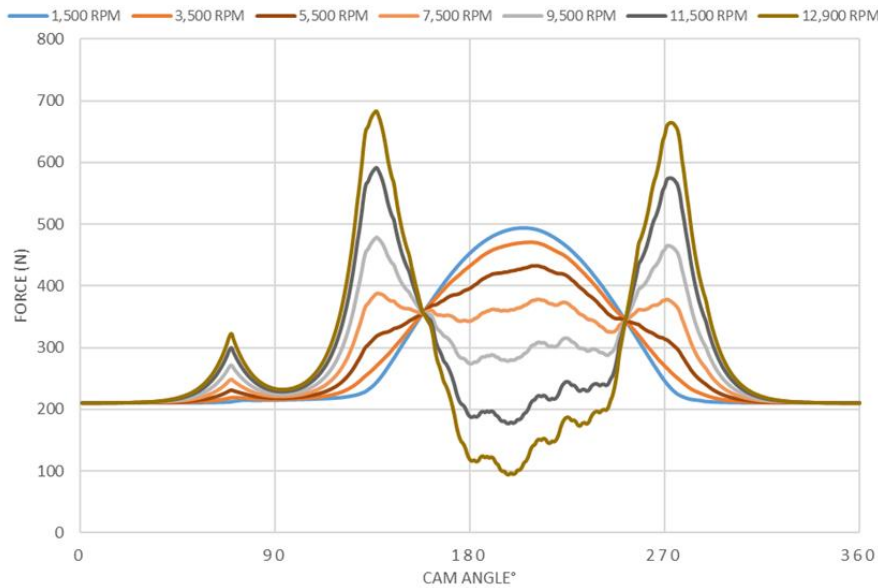


Graph 14 Yamaha Intake Valve Acceleration at Different Engine Speeds

Graph 13 shows the KTM Moto3's intake valve velocity as engine speed increases in increments of 1000RPM from 1,500-13,500RPM. Additionally, Graph 14 shows the same for the valve acceleration in regard to the intake valve-train of the YZ250F. The purpose of both graphs is to show that as engine speed increases so does velocity and acceleration, but the pattern of the trends remains the same. Both the intake and exhaust valve-trains of each engine follow the same trends as shown in these two graphs.

10.1.3 Contact Force & Valve Float

YAMAHA EXHAUST CONTACT FORCE AT VARYING ENGINE SPEEDS



Graph 15 Yamaha Exhaust Contact Force at Varying Engine Speeds

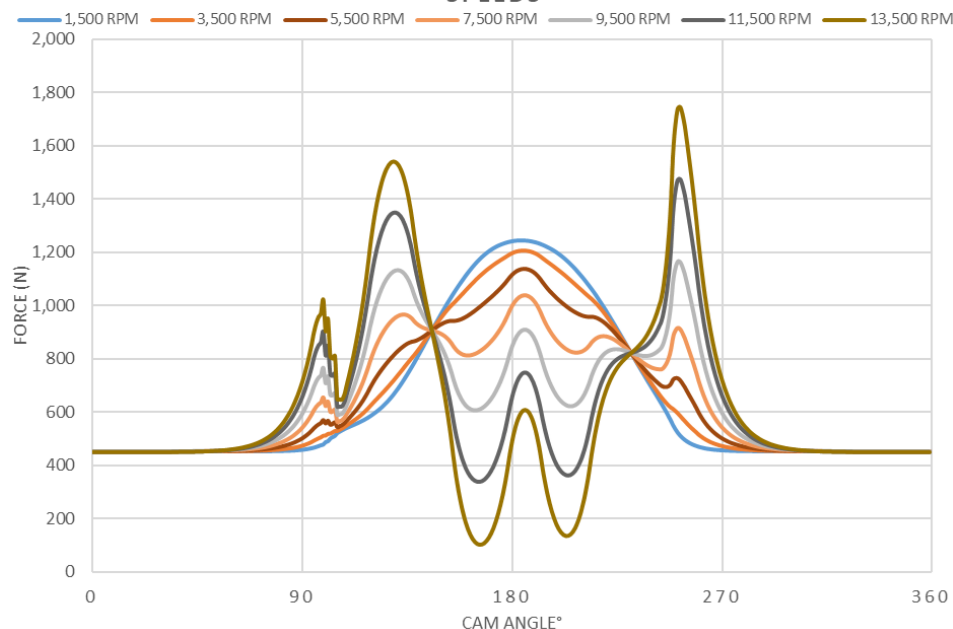
evidently clear that at each RPM the KTM is withstanding a lot more force than the YZ250F. The maximum contact force at maximum engine speed for the KTM was 1915.96N, which is an increase of 135.79% in comparison with the YZ250F which is only experiencing 812.56N at peak RPM.

Additionally, during the same interview as previously discussed during the literature review with the 2019 ZX-10R's project Leader Azuma-san, he states that in a valve-train using a bucket-tappet system, "*The bucket is always rotating*" which then increases durability as the cam lobes contact point is constantly changing. On the other hand, with a finger follower actuating system, the followers are fixed and that means "*you need to build a very strong follower with*

strong surface treatment, which is why we use DLC" (Azuma-san, 2018). Subsequently, DLC gets a value towards diamond in Rockwell Hardness testing, the significantly higher contact force experienced

Graph 15 and Graph 16 shows the contact force plotted against cam angle for the exhaust valve-trains of both engines, the same graphs regarding the intake valve trains can be found in Appendix 14.15. Each graph starts at 1,500RPM going up in increments of 2,000 until reaching the maximum engine speed; 13,500 RPM for the KTM and 12,900 RPM for the YZ250F. Furthermore, each graph shows that as engine speed increases so does the peaks and troughs. Additionally, upon further review it is

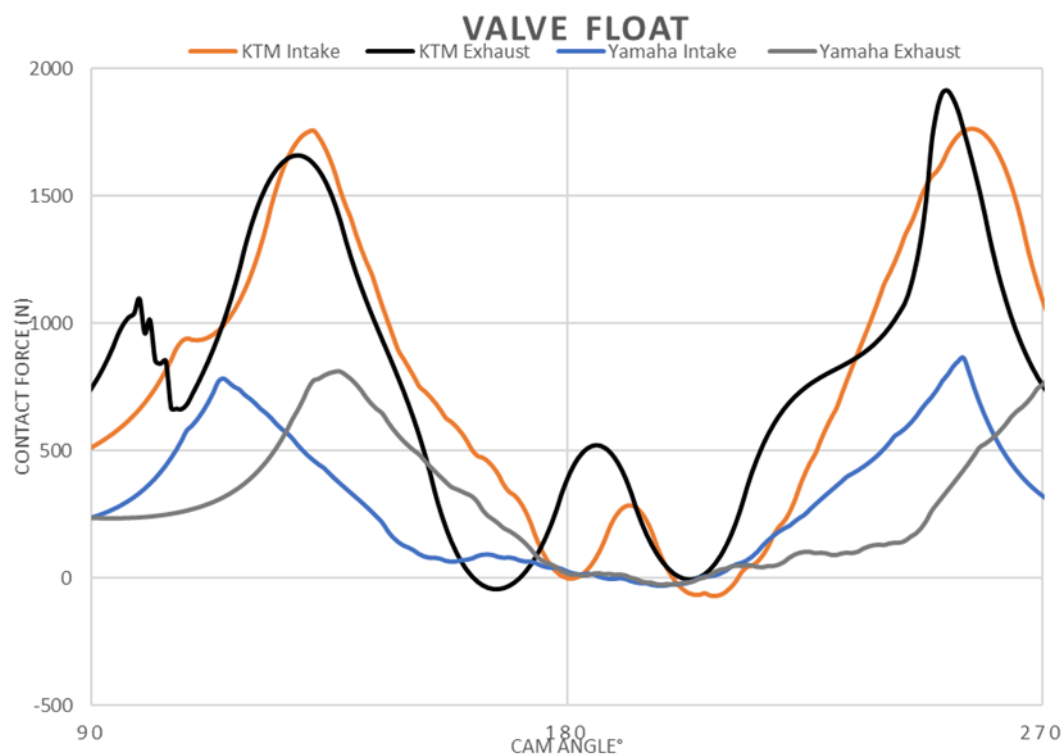
KTM EXHAUST CONTACT FORCE AT VARYING ENGINE SPEEDS



Graph 16 KTM Intake Contact Force at Varying Engine Speeds

throughout the rev range of the KTM finger-follower system as opposed to the bucket-tappet system, may further explain why KTM used a DLC coating in the Moto3 and Yamaha GYTR did not in the YZ250F.

As previously explained during the analysis of Graph 12, when the valves are at full lift, the valve acceleration is negative and as engine speed increases so does this negative acceleration. Furthermore, as RPM continues to increase the negative acceleration can be so large that ma (*moving mass x valve acceleration*) is equal and opposite to the spring force ($K(y_0 + y)$), causing the contact force to equal zero, this is then the engine speed in which the valve-train begins to experience valve float. Valve float is the term used for when the valves do not follow the closing ramp of the cam profile. Valve float can be broken down into two different types, Loft and Bounce; loft is where the lifter is thrown off the cam lobe. The more applicable in this study is bounce, where the valve bounces off its seat before settling. To the rider slight valve float will be felt as power quickly falling off at high engine speeds, whereas more severe valve float will be heard as a “*Sputtering Sound*” at peak engine speed, and in the most severe cases can cause the valve to hit the piston (Summit Racing, 2020).



Graph 17 Float Speed

As valve float speed can be expressed as; $FC = 0$, float speed for both the intake and exhaust valve-trains of both engines was determined by plotting the contact force against cam angle and then altering engine speed until it dropped below 0 Newtons as shown in Graph 17. Additionally, the float speed for each valve-train can be found in Table 9 on the subsequent page.

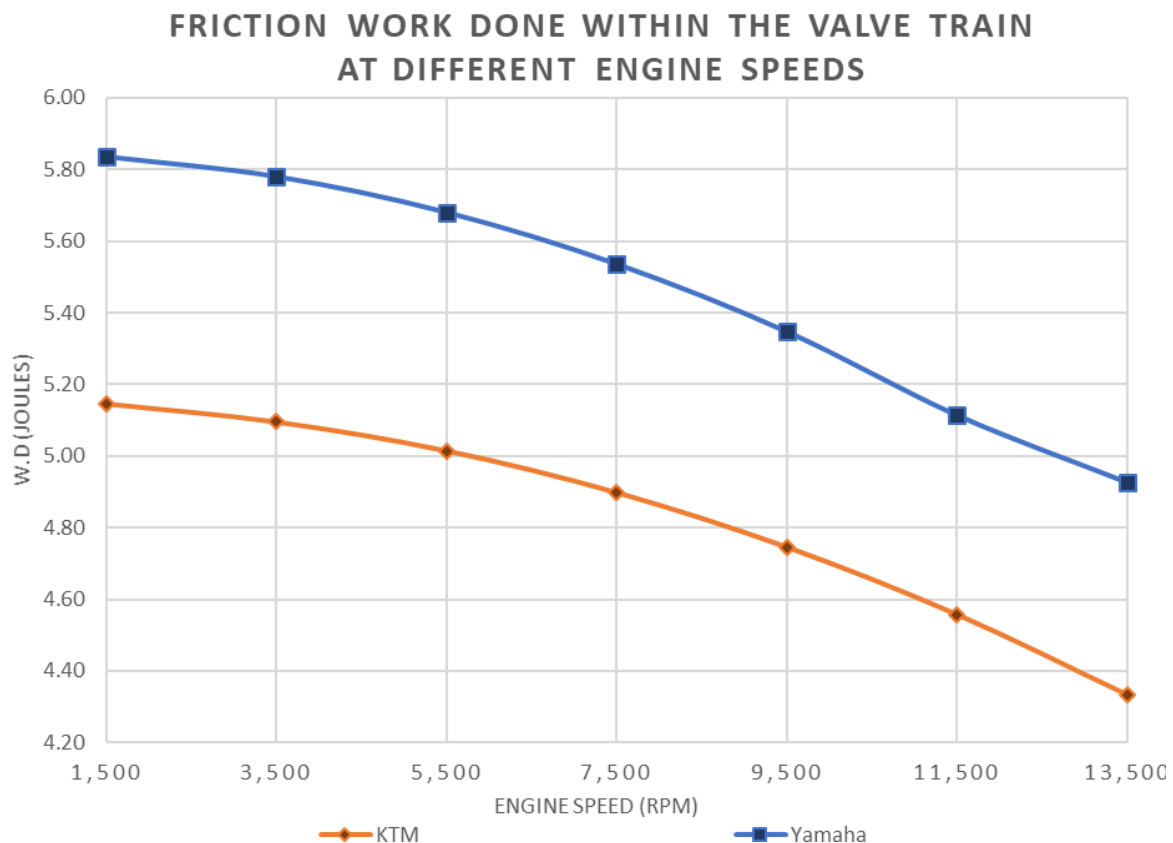
Valve float for all valve-trains are below their respective rev ranges which of course is an absolute necessity, especially for the KTM which is frequently running at higher engine speeds than the YZ250F.

	Valve Float		
	KTM	Yamaha	
Intake	14,500	13,750	RPM
Exhaust	13,850	13,600	RPM

Table 9 Valve Float RPM

10.2 Friction Model

The two following graphs show the work done and power losses through friction within the entire valve-train (*Intake + Exhaust*), of both the KTM and YZ250F, ranging from 1,500-13,500RPM increasing in increments of 2,000. Furthermore, as previously stated throughout the study the peak RPM for the YZ250F is 12,900RPM but for the purpose of the friction model, both engines were run to 13,500RPM to analyse the differences in power losses and work done through friction, that each engine experiences when running at the same engine speeds. The graphs regarding the intake and exhaust valve-trains of each engine individually for power losses and work done can be found in Appendix 14.19 & 14.20, these were not used in the main body of the report as the objective was to analyse the overall frictional losses within the valve-trains of both engines as opposed to the intake and exhaust separately.

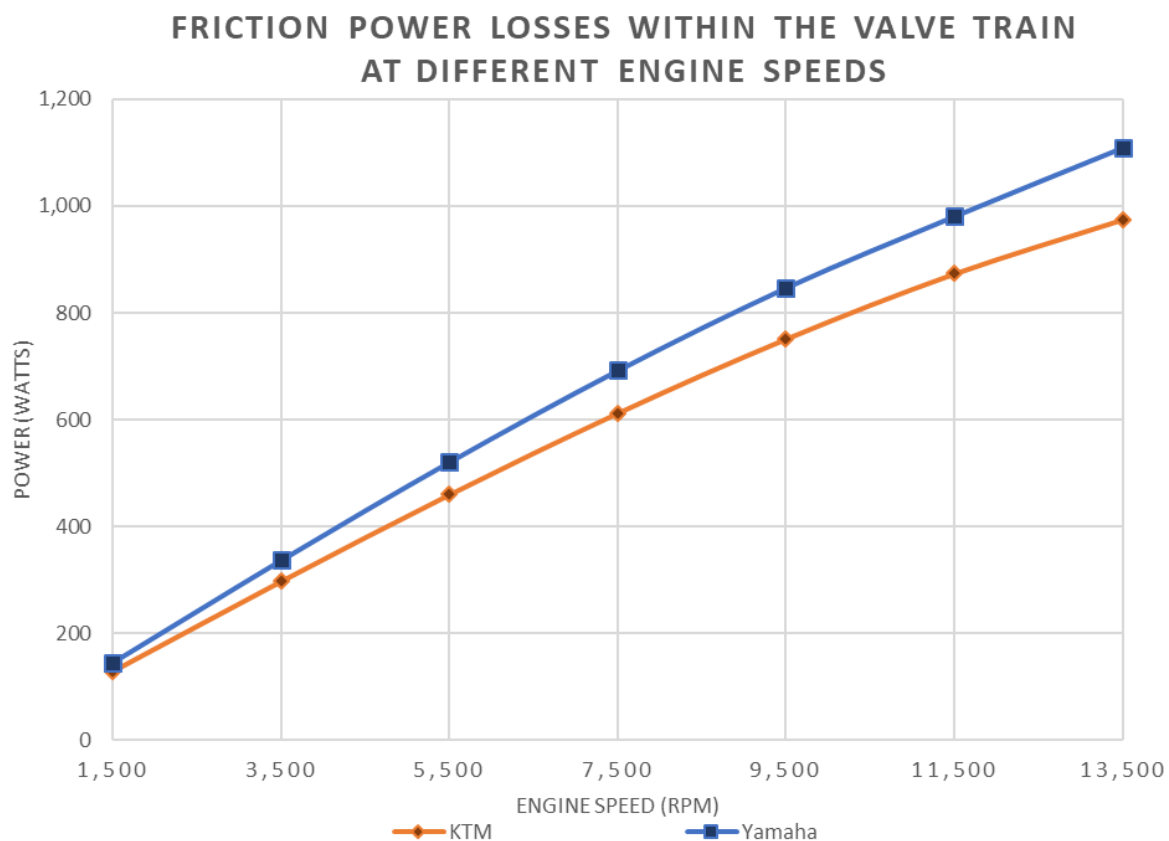


Graph 18 Friction Work Done

Engine speed is not a contributing factor when calculating the spring force; $k(y + y_o)$, thus the spring force values during the camshaft's revolution will remain the same throughout the rev range. On the

other hand, the inertia force; $\text{moving mass} \times \text{Valve Acceleration}$, is continuously changing as RPM increases. These two forces are added to one another to calculate the contact force of the YZ250F (Equation 18) and Valve Force of the KTM (Equation 19), both of which are contributing factors when calculating the Work Done of the valve-train. Additionally, the work done of each valve-train in Graph 18 is an absolute sum of the work done at every degree of cam angle (Equation 28). Also, as explained in section 10.1.3, when engine speed increases so does the negative acceleration seen at full lift, as a result of this, the sum of contact force per revolution will also decrease with the increase in RPM. In turn, this will cause the sum of work done by friction through the valve-train to decrease as engine speed increases, this is visually demonstrated in Graph 18.

Upon further review, it is clear in Graph 18 that the valve train of the YZ250F experiences more work done in comparison to the KTM, on average the YZ250F showed 13.1% more work done by friction throughout the rev range. Additionally, when considering the power losses of both valve-trains the YZ250F, once again had higher results than the KTM, Graph 19.



Graph 19 Valve-Train Power losses Through Friction

The reason that the KTM experiences less power loss and work done by friction is due to the significantly lower coefficient of friction of the DLC coating on the finger follower as opposed to the uncoated steel bucket-tappets used in the YZ250F. One study state that DLC coatings showed on

average a 40% lower coefficient of friction when compared to that of uncoated steel parts (Simko, 2012). In regard to this study the coefficient of friction for both valve-trains, was determined through the authors research, as previously discussed and referenced in section 9.3.1.3. The DLC coated finger followers have a friction coefficient of 0.05 which is only 37.59% of the bucket-tappets 0.16 friction coefficient. Therefore, as a result of using a DLC coating, a coefficient of friction reduction of 62.41% in was achieved by KTM in comparison with the YZ250F's uncoated steel bucket-tappets.

Figure 89 shows the DLC-Coated Finger Follower from the KTM's valve train which “shows a highly polished surface” (Arup Gangopadhyay, 2012). Additionally, as discussed in section 10.1.3 the finger followers experience significantly higher contact forces compared to the bucket-tappets. Additionally, the contact force is a contributing factor in the methodology of the friction model. Therefore, the fact that the KTM valve-train experiences significantly more contact force, yet the power losses through the entire valve train is noticeably lower than that of the YZ250F shows how effective the use of thin-film coatings are as a method of reducing friction.



Figure 89 KTM DLC Coated Finger Follower

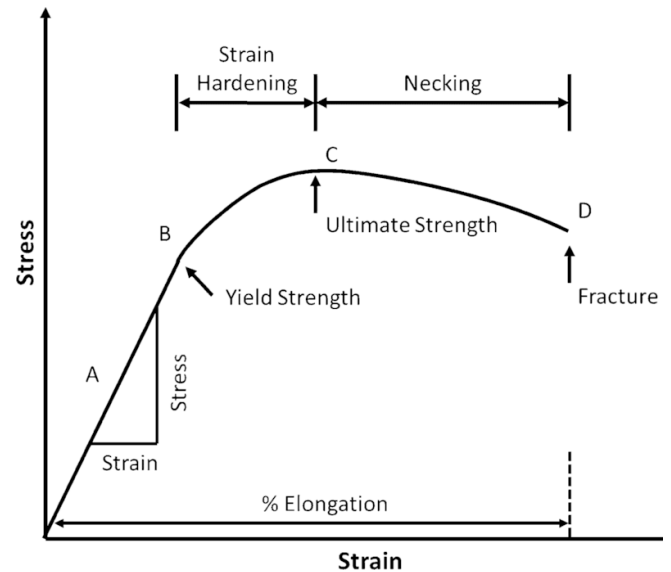
	Overall Valve train power loss in HP						
	1,500 RPM	3,500 RPM	5,500 RPM	7,500 RPM	9,500 RPM	11,500 RPM	13,500 RPM
KTM	0.1725	0.3986	0.6164	0.8210	1.0076	1.1712	1.2768
Yamaha	0.1956	0.4521	0.6982	0.9279	1.1353	1.3145	1.4422

Table 10 Power-Losses in HP at Different Engine Speeds

Table 10 shows the power losses in the valve-train in HP. Throughout the rev range on average the Yamaha is losing 12.54% more power compared to the KTM. Subsequently, this further proves that by applying a DLC thin-film coating, assists in reducing power losses through valve-train and in turn will enhance engine performance even if only slightly. At 13,500 RPM the KTM is only losing 0.1654 less HP than the YZ250F, even though this is a considerably insignificant amount, in the world of motorsport it is a case of every little helps.

10.3 FEA Study

Graph 20 Stress-Strain Curve Graph 20 shows a typical stress-strain curve. Point B on the curve is the point of yield, which is when the stress a component experiences surpasses the point of yield meaning the deformation it experiences is no longer elastic (*Non-Permanent*) and instead becomes plastic meaning the deformation becomes permanent. Furthermore, Point C “*Ultimate Strength*”, is the point on the stress-strain curve where a material reaches its UTS (*Ultimate Tensile Strength*) or simply tensile strength. UTS “*quantifies the force required to pull a material to the point where it breaks*” (Ashish, 2017).



Graph 20 Stress-Strain Curve (Hoag, 2013)

The yield and tensile stress of the YZ250F’s bucket-tappet were generated automatically within SolidWorks when the material was applied to the component. On the other hand, regarding the DLC coated steel used for the finger follower of the KTM, a custom material was created as previously discussed, where the mechanical properties were inputted. The tensile strength of ta-C form DLC coated steel was attained and added, but unfortunately the yield point was not found, so was set to the same value as uncoated steel. This was not a major issue, as the purpose of the FEA study was to compare the stresses being exerted upon the bucket-tappet/finger follower from the cam lobe and how close this was to their UTS values.

Figure 90 and Figure 91 below show the Von Mises stress results from the FEA study.

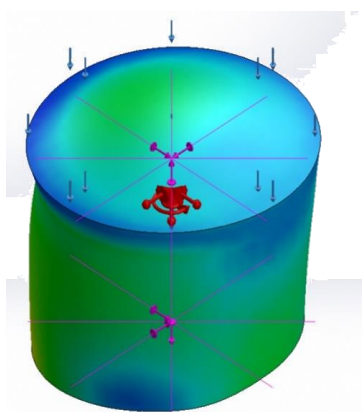


Figure 90 Bucket-Tappet; Von-Mises Stress Results

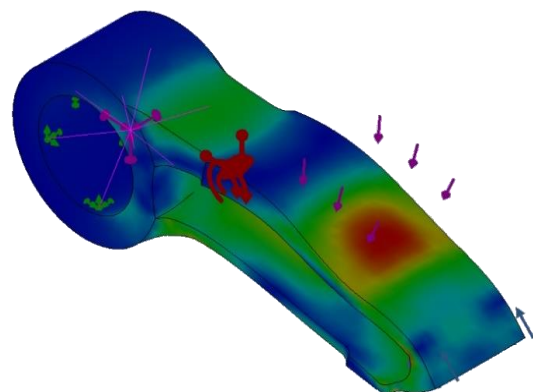


Figure 91 Finger-Follower; Von-Mises Stress Results

As previously explained the finger follower experiences significantly higher contact forces compared to the bucket-tappets. The maximum contact force seen at top engine speed on the finger follower is

94.8% higher compared to the maximum contact force exerted on the YZ250F's bucket-tappets at peak RPM.

The maximum stress being exerted onto the bucket-tappet was $2.866\text{e}+07 \text{ N/m}^2$ and was located at node 9224, which was near the centre of the top surface where the max force between the cam-lobe and bucket takes place. Subsequently, the maximum stress seen in regard to the KTM's finger follower was $1.795\text{e}+08 \text{ N/m}^2$ and was found at node 5259 situated around the area on top of the follower where the maximum force is being exerted upon it from the cam-lobe similarly to the bucket-tappet. Furthermore, a node in FEA is a location in space where D.O.F (*degrees of freedom*) are defined "*the results of a finite element analysis, (deflections and stresses), are usually given at the nodes*" (Nodes & Elements, 2014). The maximum stress experienced by the bucket-tappet is only 6.4% of its tensile strength ($4.48\text{e}+07 \text{ N/m}^2$) and the maximum stress the KTM's finger follower experienced was only 10.39% of its tensile strength ($1.73\text{e}+09 \text{ N/m}^2$).

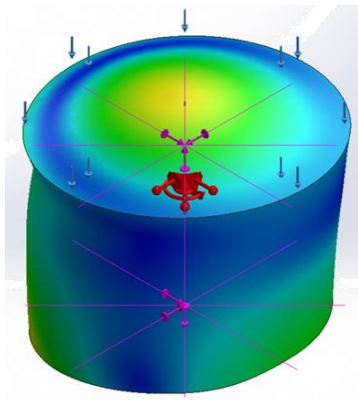


Figure 92 Bucket-Tappet; Resultant Displacement

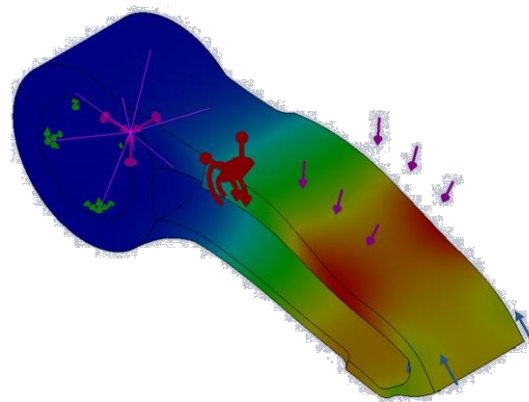


Figure 93 Finger Follower; Resultant Displacement

Within the FEA model of the finger-follower there was a maximum displacement of $4.724\text{e}-02\text{mm}$ and $2.123\text{e}-03\text{mm}$ for the YZ250F's bucket-tappet. Both of these values are so small they were regarded inconsequential. To see the same images regarding the equivalent strain of both components please refer to Appendix 14.21, these were not included in the main body of the study as the values were so minute they were disregarded.

11 Conclusion

To conclude, this study has outlined the importance of the valve-train within the internal combustion engine, and how it can be designed to reduce emissions and increase performance, by reducing mass and also by reducing power losses as a result of friction.

The comparative analysis of the two valve-trains demonstrated how vast the differences between the two engines are, ranging from the cam-profiles to the valve actuating systems. The KTM moto3 is a purpose-built road racing engine where *“money is no object”*. Whereas the GYTR cylinder head used in the YZ250F uses an aftermarket, high-performance valve-train, which is more affordable and produced in higher quantities. The results of this can be seen throughout the study, an example being the cheaper and more easily mass-produced symmetric camshaft used in the YZ250F. As opposed to the KTM’s more expensive asymmetric camshaft allowing a more aggressive cam-profile, in turn allowing a higher valve-lift, which is more favourable in road-racing where the rider will be at full power for longer durations than an Enduro rider.

Finally, this report has shown the relevance of using thin-film coatings on ICE components to increase durability, reduce wear, and reduce power losses as a result of friction, this was proven when analysing the DLC coated finger followers as opposed to the uncoated steel bucket-tappets.

12 Covid-19 Pandemic: Implications on Project

The majority of work that required the facilities only available on campus had already been completed prior the closure of the university, unfortunately there was a small number of limiting factors to this project as a result of the Covid-19 Pandemic, they are as follows;

- The cam-profile of the Yamaha YZ250F was to be re-measured as the closing phase of the valve acceleration graph showed a noticeably larger value than what was seen during the opening phase. This should not be the case as the camshaft is symmetrical, thus questioning the validity of results. Additionally, the Motion simulation was to be run at all the same engine speeds as was used when calculating the contact force.
- A forces graph was to be generated within the motion simulation of the contact forces between the cam-lobe and bucket-tappet/finger follower. Next, this information was to be exported out for each RPM to a Microsoft Excel Spreadsheet and compared to the calculated contact forces as a validation tool, this could not be completed as the author did not have the computational power at home to carry this out and solely relied on the computers on campus.
- The cam-profile and valve lift of the KTM Moto3 was to be remeasured, to ensure the validity of the graphs.

- The valve lift of the YZ250F was to be measured using the same method as the KTM and then compared to the valve lift calculated from the 360 x 360 matrix.
- The FEA study was carried out once, briefly before the closure of the university campus, the results used within this paper had already been put into a word document, and as previously explained the author did not have the computational power at home to re-run the study and collect more information to be used in the analysis. Finally, more FEA studies were also planned, but unfortunately could not be completed.

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14 Appendices

14.1 Appendix 1 Valve Timing Diagrams

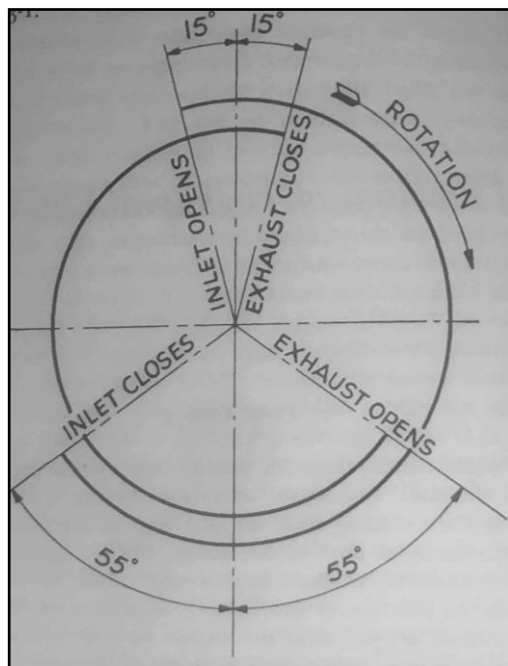


Figure 94 Typical Valve Timing for Showroom Stock Sports Car (Intake & Exhaust Strokes Only) (Smith P. H., 1977)

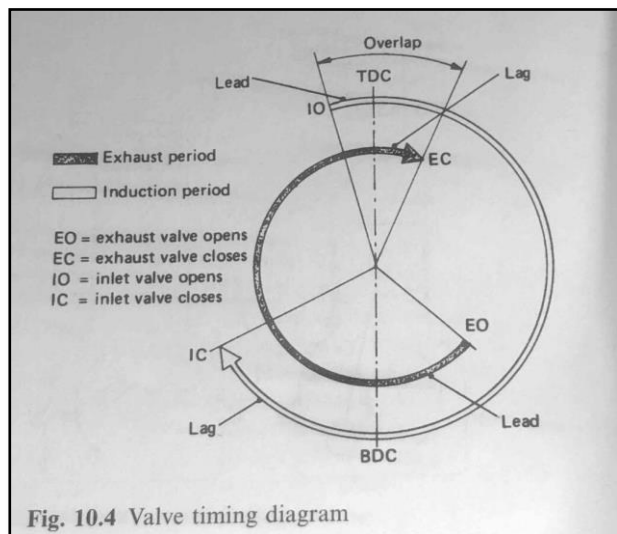


Fig. 10.4 Valve timing diagram

Figure 95 Valve Timing Heinz (Heisler, 1999)

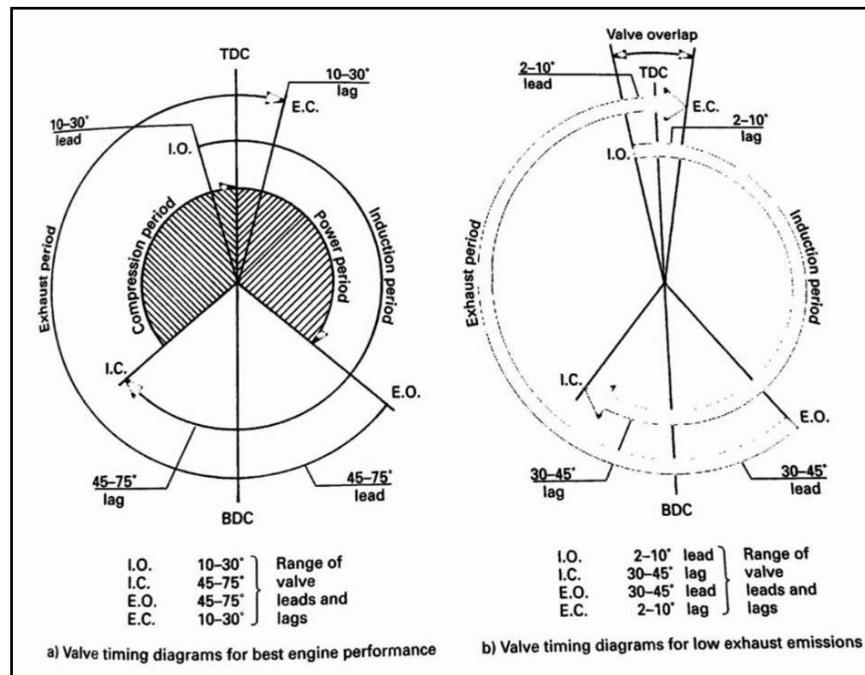


Figure 96 Valve Timing Diagram for Best Engine Performance & Low Exhaust Emissions (Heisler, 1999)

14.2 Appendix 2 Over Bucket Shim Pop

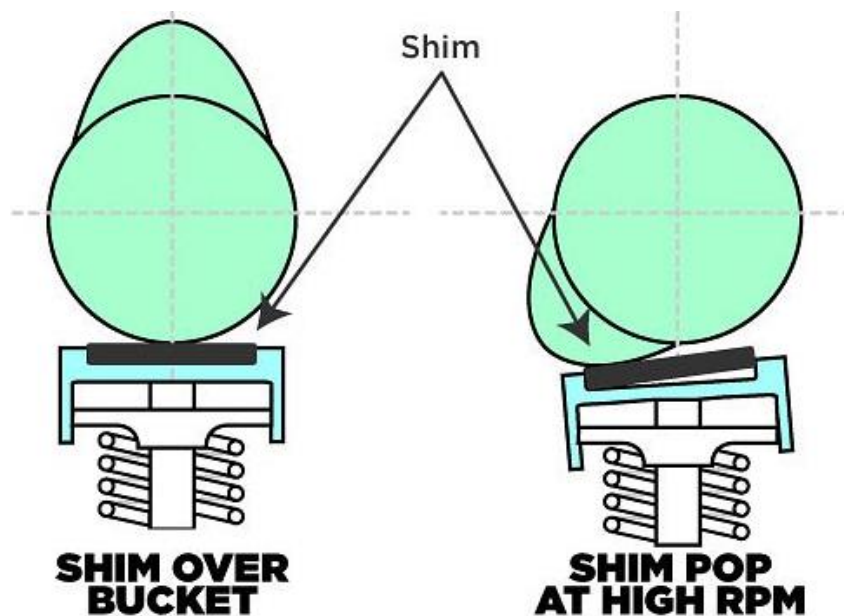


Figure 97 Shim Pop at High RPM (Mechanics Stack Exchange, 2017)

14.3 Appendix 3 Moto3 Rev-Limit Regulations

14.3.1 BSB HEL Performance MotoStar Moto3

5.3.14 Ignition/Engine Control System (ECU)
ECU is free with a maximum rev limit of 13,500rpm, this may be checked at any time during the event on a Dyno or be enforced by the compulsory fitment of approved logging device.

Figure 98 BSB HEL Performance MotoStar Moto3 Rev Limit (Motorsport Vision Racing, 2019)

14.3.2 FIM Moto3 Junior World Championship

4. Maximum engine RPM in acceleration is controlled by the official ECU. The RPM Limiter strategy target is set at 13,500 rpm.

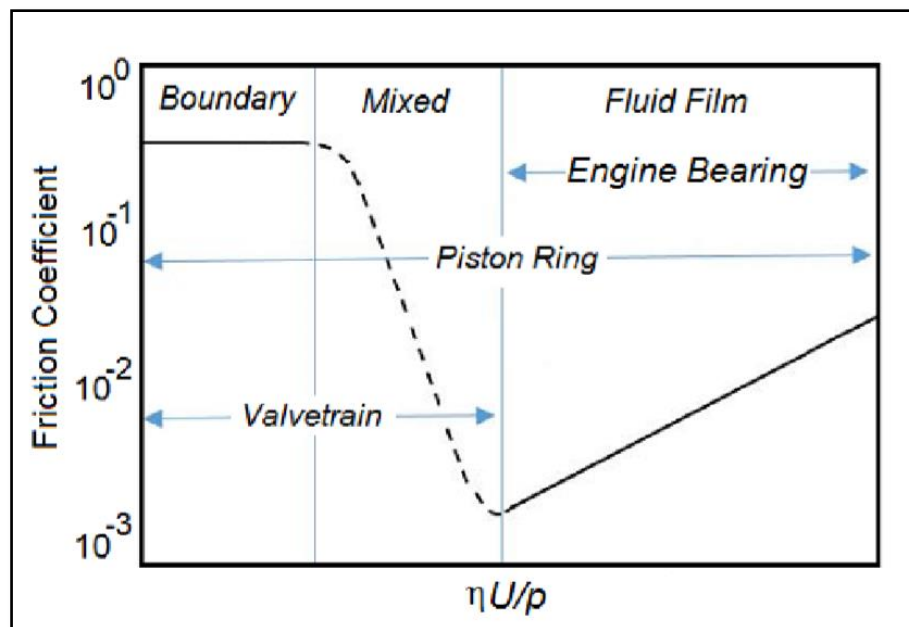
Figure 99 FIM Moto3 Junior World Championship Rev Limit (FIM, 2019)

14.3.3 World Moto3

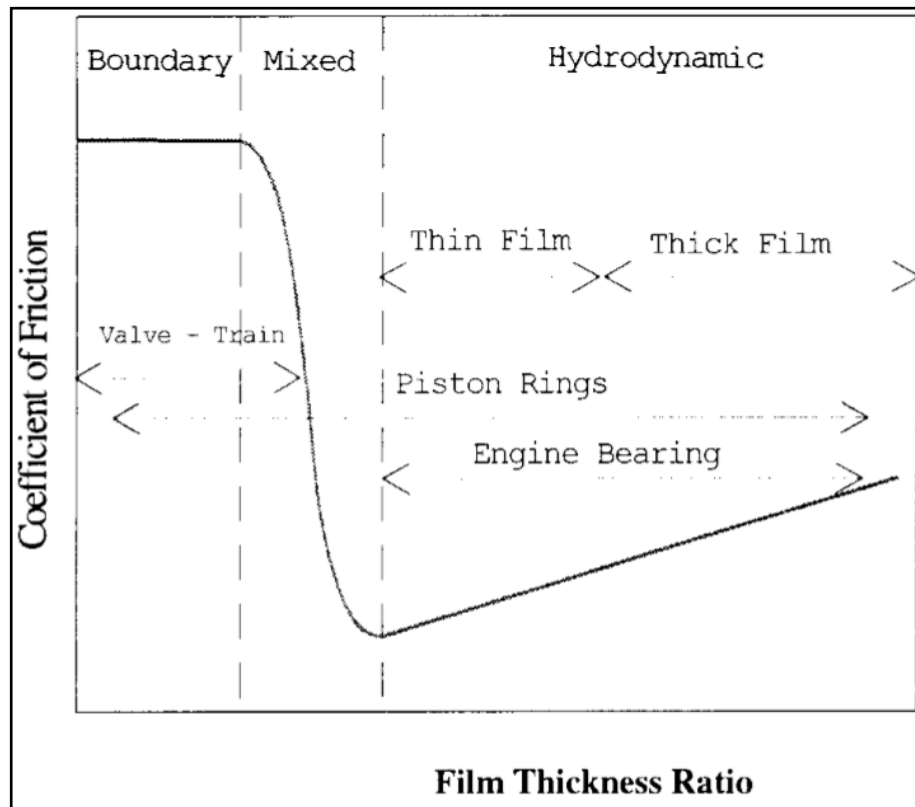
11. Maximum engine RPM in acceleration is controlled by the official ECU*. The RPM Limiter strategy target is set at 13,500 rpm.

Figure 100 World Moto3 Rev Limit (FIM, 2019)

14.4 Appendix 4 Stribeck Curve



Graph 21 Modified Stribeck curve and different regimes of lubrication of IC engine components under sliding motion (Raza, 2017)



Graph 22 Stribeck Diagram (Emes, 1998)

14.5 Appendix 5 Valve-Assembly photos

14.5.1 KTM Moto3 Exhaust



Figure 101 KTM Exhaust Valve



Figure 102 KTM Exhaust Valve-Spring

14.5.2 Yamaha YZ250F Exhaust



Figure 103 YZ250F Exhaust Valve Spring



Figure 104 YZ250F Exhaust Valve



Figure 105 YZ250F Exhaust Camshaft

14.6 Appendix 6 Optical Dividing-Head

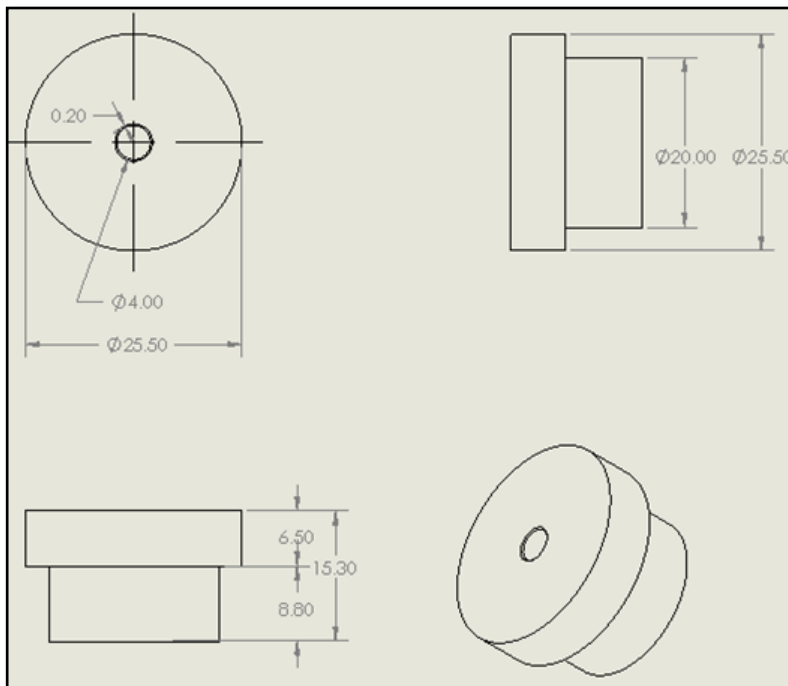
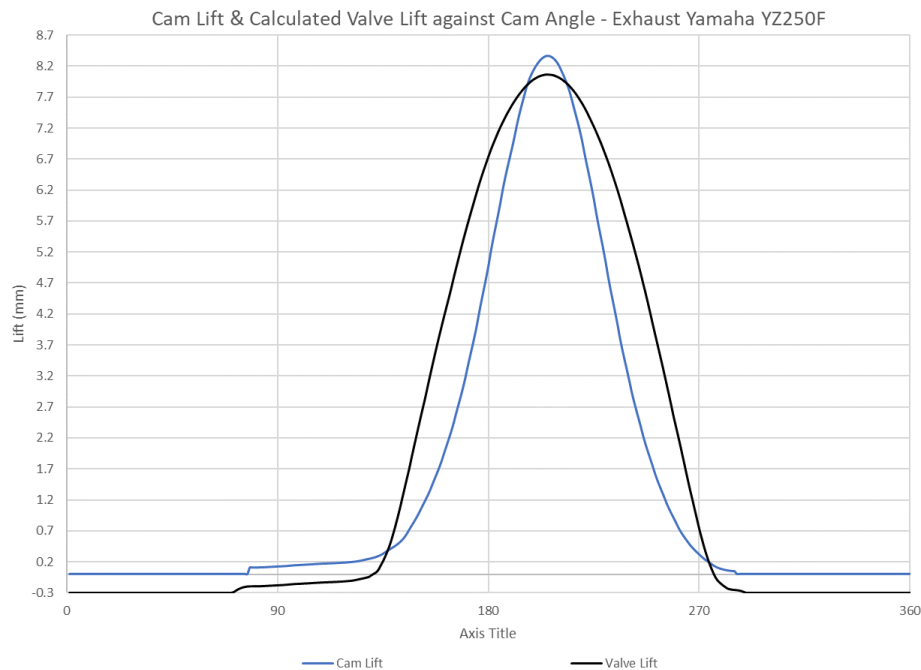


Figure 106 End Cap Technical Drawing



Figure 107 Aluminium Block

14.7 Appendix 7 Yamaha YZ250F Exhaust Cam lift & Valve Lift Against Cam Angle



Graph 23 YZ250F Exhaust Valve & Cam Lift against Cam angle

14.8 Appendix 8 Yamaha YZ250F Valve Spring Heights

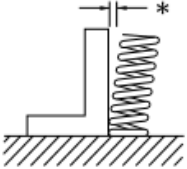
Valve spring:			
Free length	IN	37.81 mm (1.49 in)	36.81 mm (1.45 in)
	EX	37.54 mm (1.48 in)	36.54 mm (1.44 in)
Set length (valve closed)	IN	29.13 mm (1.15 in)	----
	EX	29.30 mm (1.15 in)	----
Compressed force (installed)	IN	99 ~ 114 N at 29.13 mm (9.9 ~ 11.4 kg at 29.13 mm, 22.27 ~ 25.57 lb at 1.15 in)	----
	EX	126 ~ 144 N at 29.30 mm (12.6 ~ 14.4 kg at 29.30 mm, 28.44 ~ 32.41 lb at 1.15 in)	----
Tilt limit *	IN	----	2.5°/1.7 mm (2.5°/0.067 in)
	EX	----	2.5°/1.6 mm (2.5°/0.063 in)
			
Direction of winding (top view)	IN	Clockwise	----
	EX	Clockwise	----

Figure 108 Yamaha YZ250F Valve Spring Heights (Yamaha, 2017)

14.9 Appendix 9 Moment Inertia from SolidWorks Table

Moments of inertia: (grams * square millimeters)
 Taken at the output coordinate system.

$I_{xx} = 95.17$	$I_{xy} = 117.99$
$I_{yx} = 117.99$	$I_{yy} = 1148.67$
$I_{zx} = 0.00$	$I_{zy} = 0.00$

Table 11 Finger Follower Moment of Inertia SolidWorks

14.10 Appendix 10 DLC Coating Properties

Strength. Based on the stretching tests, the tensile strength of the ta-C films is calculated and is equal to 1280 N/mm^2 , which is comparable to hardened steel.

Figure 109 ta-C Form DLC tensile strength (Davidson, 2000)

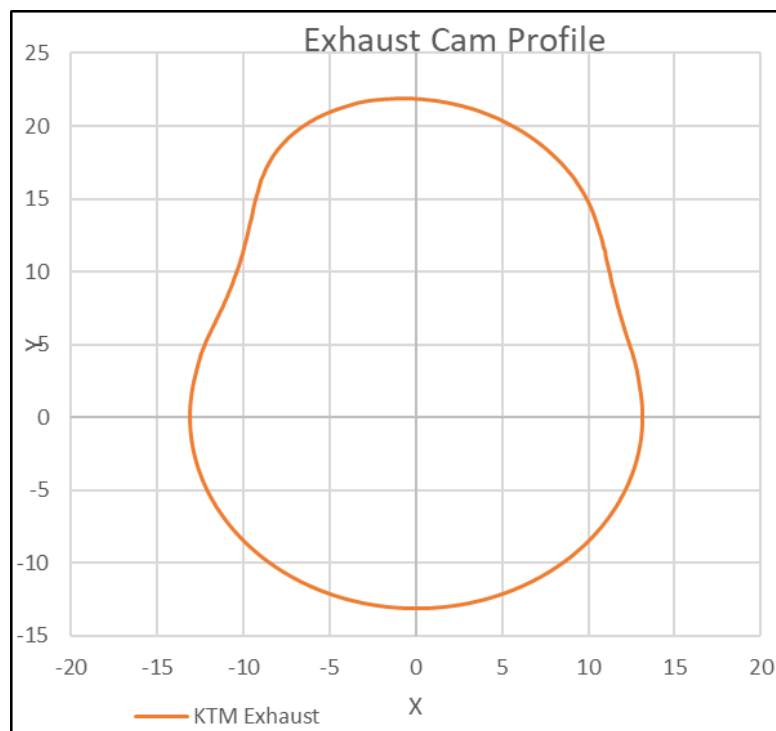
The elastic modulus and Poisson's ratio were $87 \pm 18 \text{ GPa}$ and 0.22 ± 0.33

Figure 110 Elastic Modulus & Poisson's Ratio (Sung-Jin Cho, 1999)

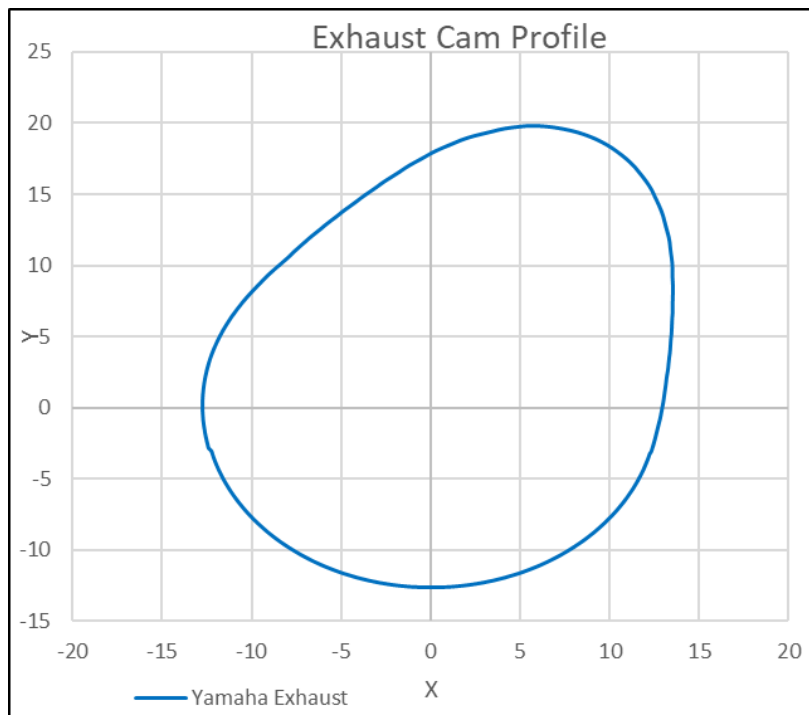
DLC film deposited on Si wafer. Typical DLC films have high residual compressive stress of up to 10 GPa. Although

Figure 111 Compressive Stress (Sung-Jin Cho, 1999)

14.11 Appendix 11 Exhaust Cam Profile Graphs

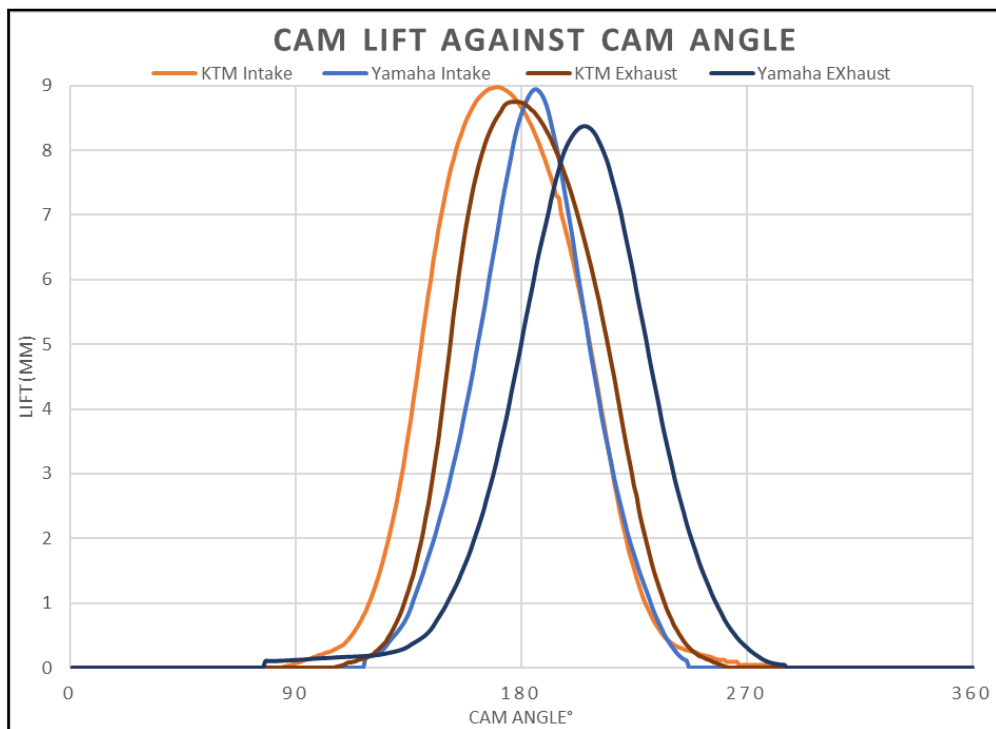


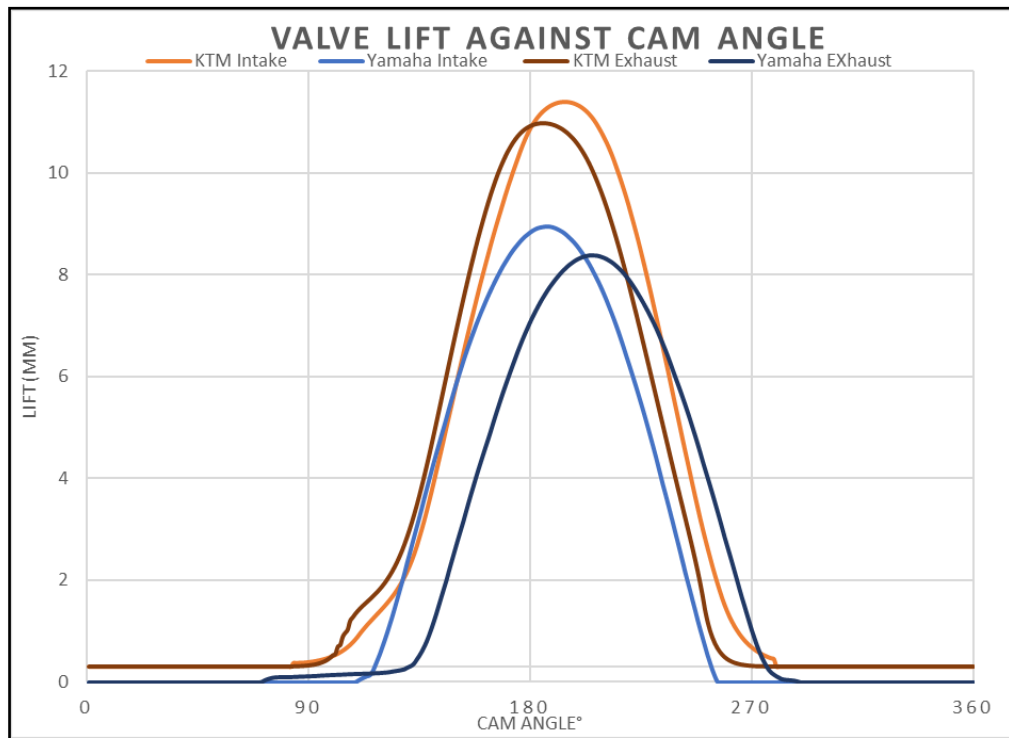
Graph 24 KTM Moto3 Exhaust Cam Profile



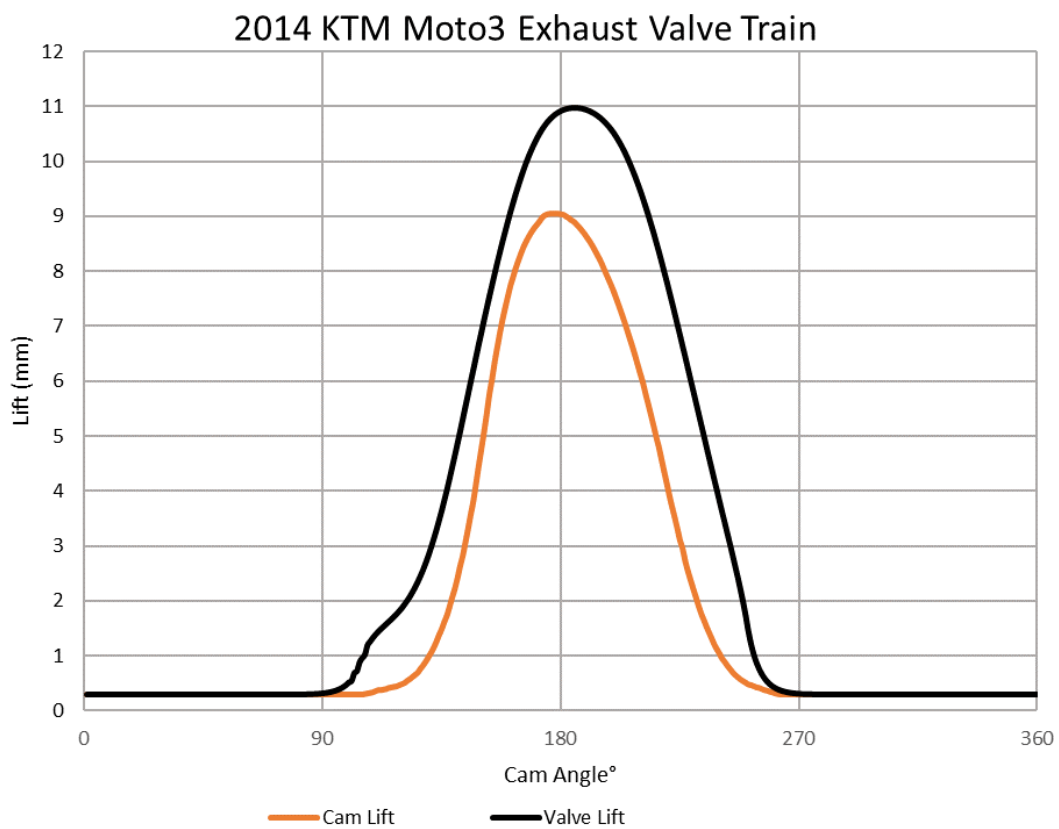
Graph 25 Yamaha YZ250F Exhaust Cam Profile

14.12 Appendix 12 Cam and Valve Lift



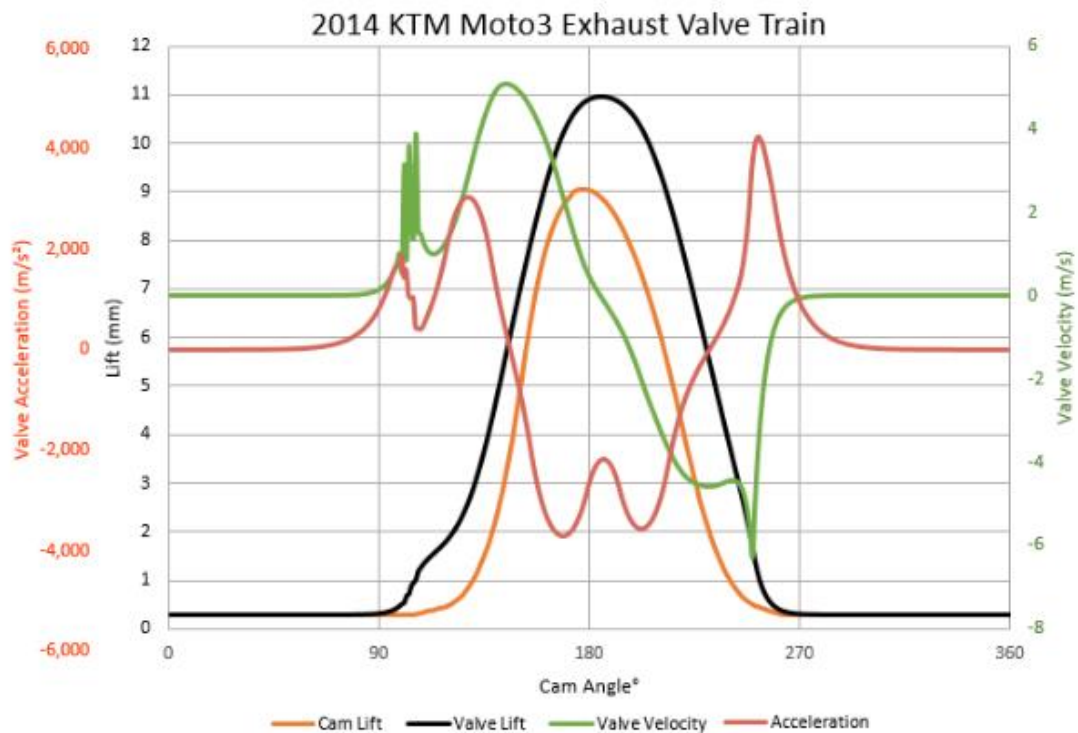


14.13 Appendix 13 KTM Exhaust Valve lift against Cam Angle



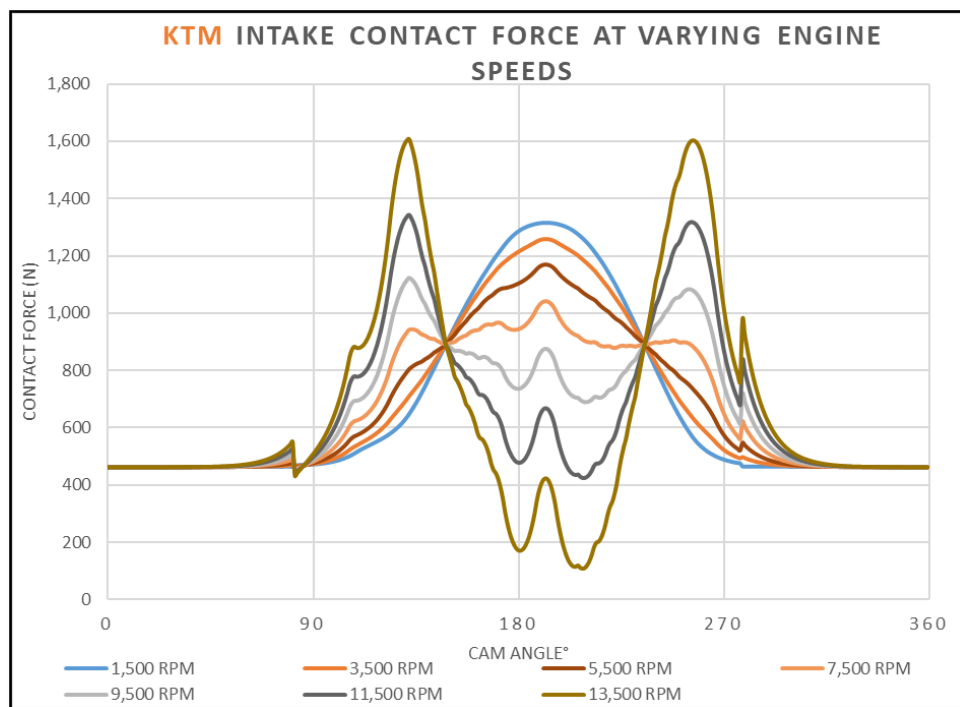
Graph 26 KTM Exhaust Valve Train

14.14 Appendix 14 KTM Cam Lift, Valve Lift, Velocity & Acceleration

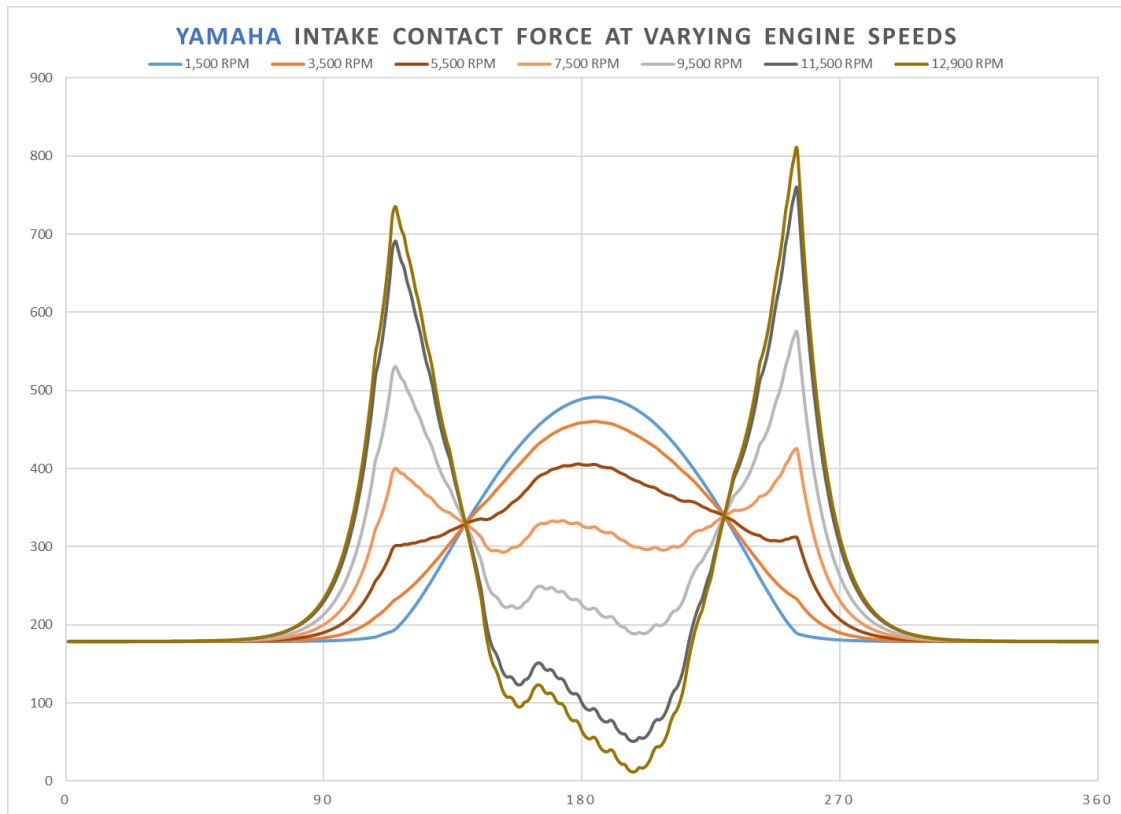


Graph 27 KTM Cam, Valve lift, Valve Velocity & Acceleration

14.15 Appendix 15 KTM and Yamaha Intake Contact Force Graphs

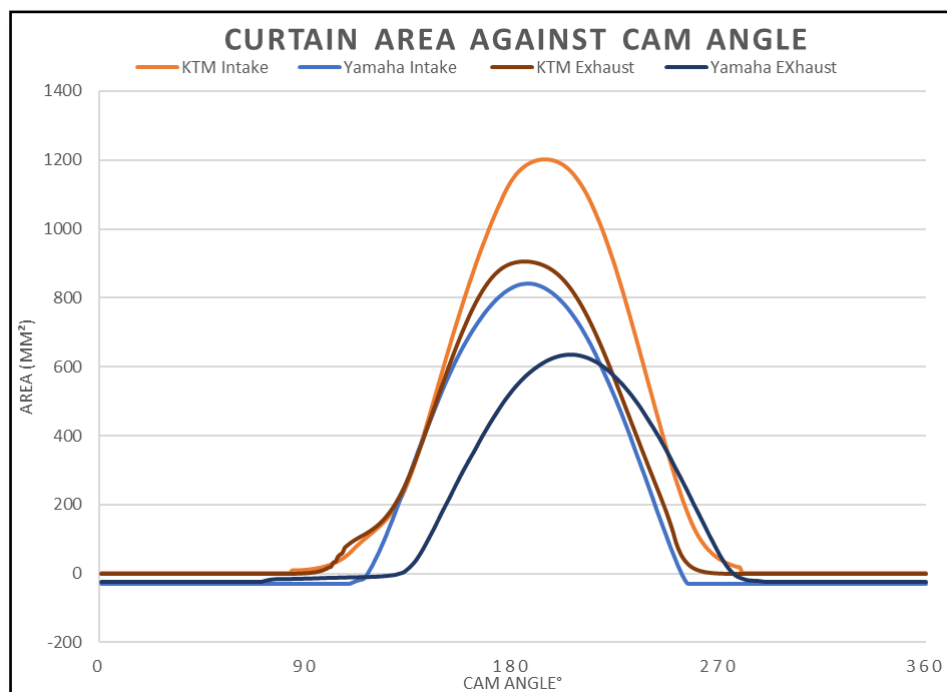


Graph 28 KTM Intake Contact Force at Varying Engine Speeds



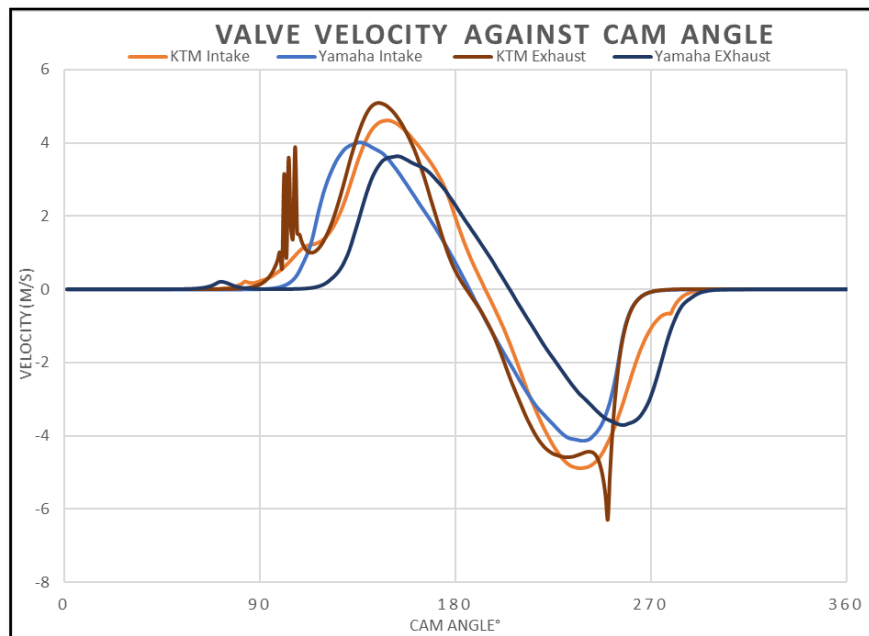
Graph 29 Yamaha Intake Contact Force

14.16 Appendix 13 Curtain Area

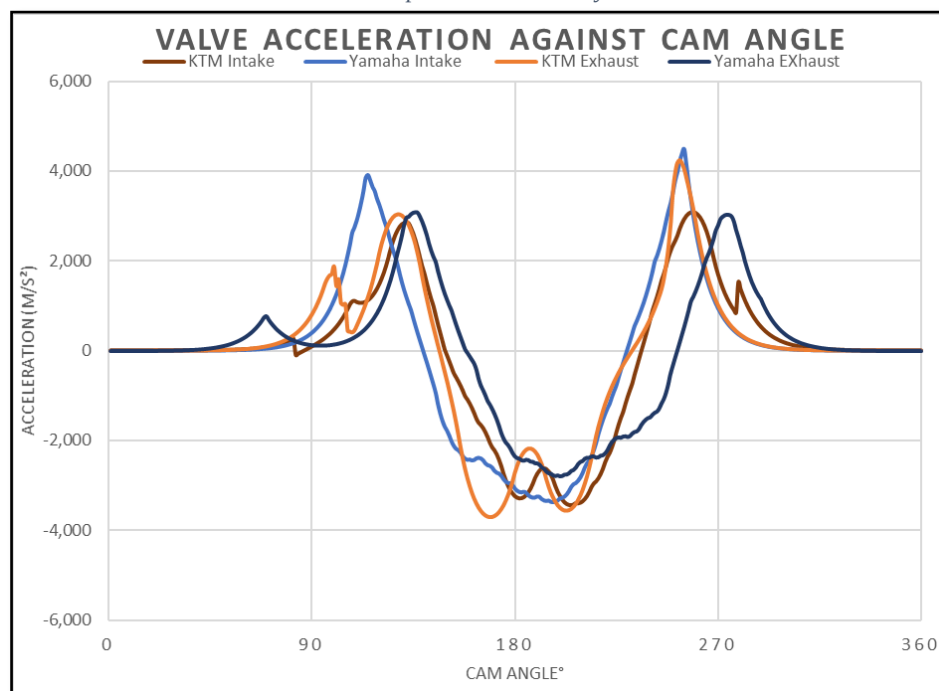


Graph 30 Curtain Area

14.17 Appendix 14 Velocity & Acceleration

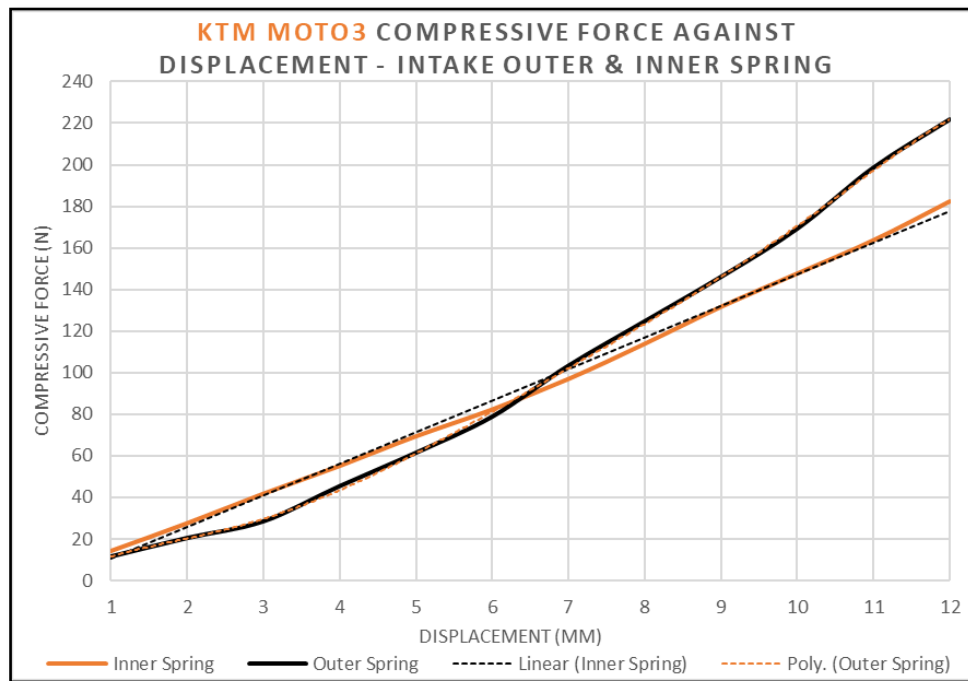


Graph 31 Valve Velocity



Graph 32 Valve Acceleration

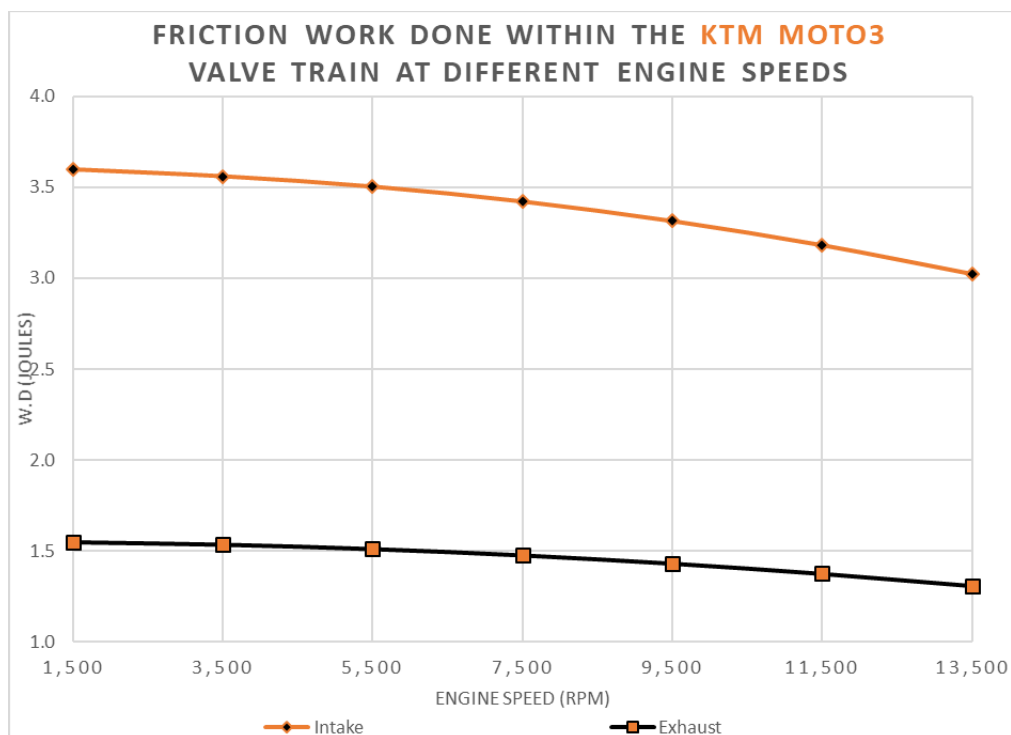
14.18 Appendix 16 KTM Moto3 Inner & Outer Coil Spring Rate



Graph 33 KTM Moto3 Compressive Force Against Displacement - Intake Outer & Inner Spring

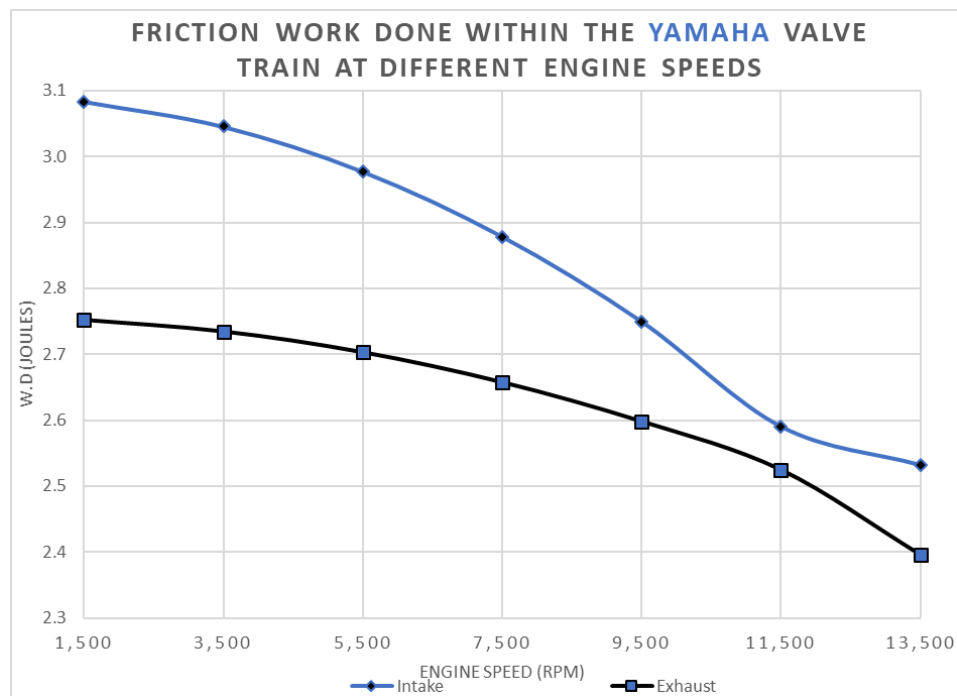
14.19 Appendix 17 Frictional Work Done

14.19.1 KTM



Graph 34 KTM Work Done by Friction

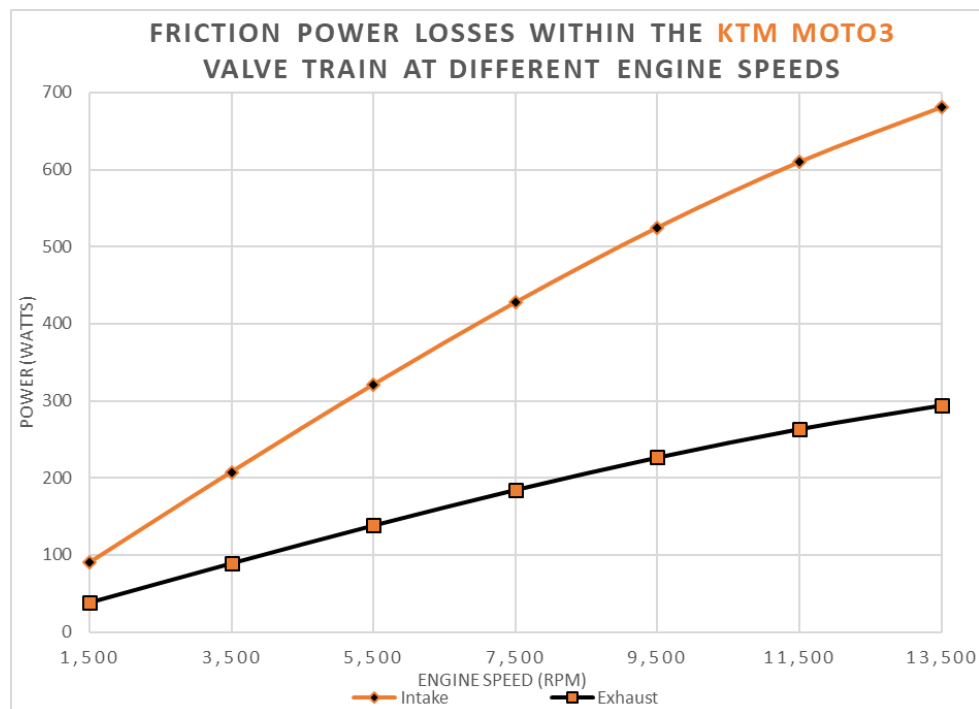
14.19.2 Yamaha



Graph 35 YZ250F Work Done by Friction

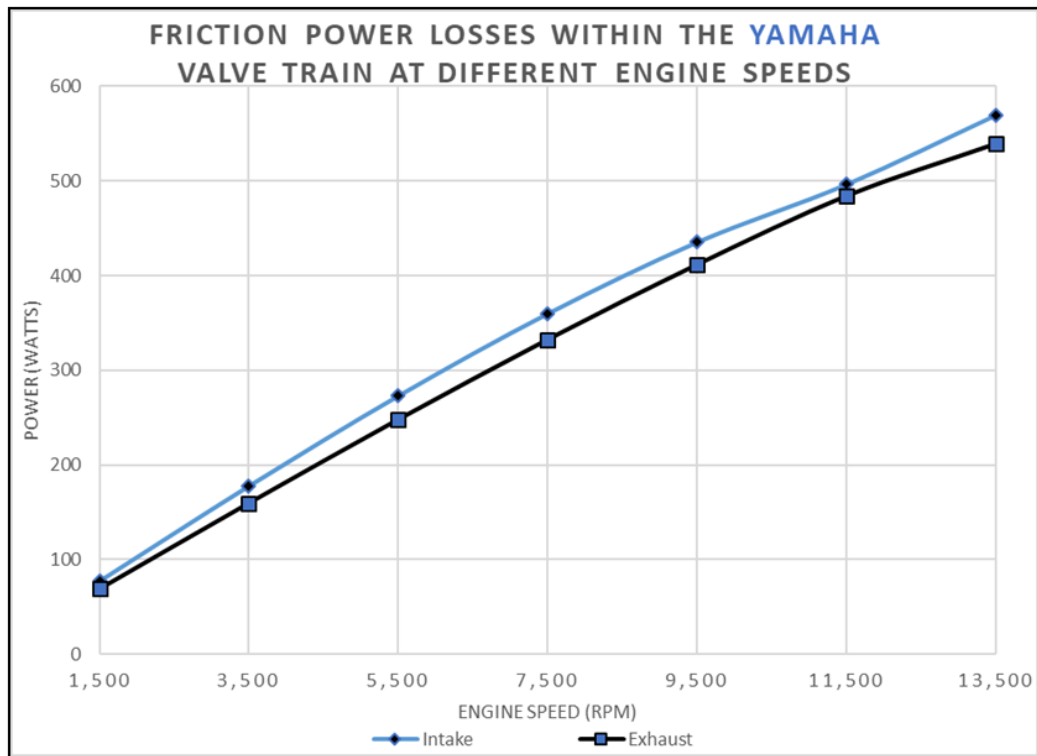
14.20 Appendix 18 Power Losses Through Friction

14.20.1 KTM



Graph 36 KTM Power Losses by Friction

14.20.2 Yamaha



Graph 37 YZ250F Power Loses Through Friction

14.21 Appendix 19 Equivalent Strain

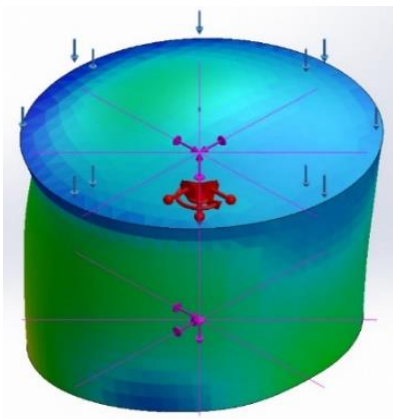
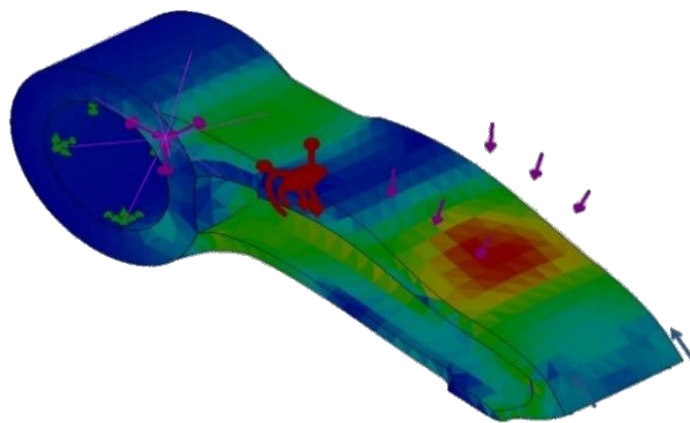
Figure 112 Bucket-Follower;
Equivalent Strain

Figure 113 Finger Follower; Equivalent Strain