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Master degree in Automotive Engineering
Propulsion Systems Development

Master Degree Thesis

Study of the lubrication system of a Wankel rotary engine to be applied as range extender

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Contents

1. Abstract 2
2. Introduction 4
   2.1. From the internal combustion engines to the hybrid powertrains 4
   2.2. Wankel engine 5
      2.2.1. Design and working principle 5
      2.2.2. Advantages 6
      2.2.3. Disadvantages 7
      2.2.4. Vehicle range extender 8
   2.3. Lubrication principles 11
      2.3.1. Aims of lubrication 11
      2.3.2. Wankel lubrication 12
      2.3.3. Lubrication regimes 14
      2.3.4. Lube oil classification 15
      2.3.5. SAE classification 16
      2.3.6. API classification 17
      2.3.7. ACEA classification 18
3. Methodology 20
   3.1. Engine data acquisition 20
   3.2. P-V diagram reconstruction 21
   3.3. Surface roughness and Stribeck's lubrication model 24
      3.3.1. Surface generation 25
      3.3.2. Stribeck's lubrication model 26
   3.4. Power losses computation 30
      3.4.1. Supports 31
      3.4.2. Apex rubbing friction 34
      3.4.3. Total losses 34
   3.5. Archard's model for Apex seals wear 35
      3.5.1. Wankel rotary engine 13B simulation 37
   3.6. Range extender simulation 40
4. Results and discussion 44
   4.1. Engine performance and combustion cycle 44
   4.2. Lubricant oil influence 47
   4.3. Supports power loss 50
      4.3.1. Rolling bearings power loss 50
      4.3.2. Bushings power loss 52
   4.4. Apex rubbing friction 55
   4.5. Total power loss 62
   4.6. Apex seals' wear 64
   4.7. Oil consumption 66
   4.8. Range extender application 69
5. Conclusion 73
6. References 75

1. Abstract

In the recent years, the main idea that leads the automotive field is to find a solution to the pollution generated by the internal combustion engines (ICE). Firstly, several after-treatment systems were designed but then, due to the recent CO_2 limitations imposed by the European Commission and by the EPA, the largest automotive companies were pushed toward a partial or total substitution of the conventional powertrain via the so called "vehicle electrification". This concept can be satisfied by means of a battery pure electric vehicle (BEV) or by means of a hybrid electric vehicle (HEV), in which a common ICE is combined with an electric powertrain. Since range autonomy is one of the main barriers for the commercial success of electric vehicles, parallel hybrids and series hybrids, such as the range extended electric vehicles, can be a good trade off between the driver requirements and the environmental ones.

The aim of this thesis is to investigate the application of a small Wankel rotary engine (Rotron RT300, 300 cm^3) as range extender, studying which are its main limits, understanding which aspects can be modified or optimized in order to understand if it is worth to use an engine with limited automotive diffusion for such an application. The small number of production cars using this engine is due to the fact that Wankel has several problems linked to fuel consumption, oil consumption and pollutant emissions. Part of these problems are strictly connected to the lubrication system: this project focuses on the study of the main components that are lubricated, to understand how and if it is possible to substitute the lubricant oil in order to reduce the friction losses and the seals wear, and to find out which is the best one in term of pollutant formation and emission according to the current regulations.

To achieve these results, the project starts from the acquisition of the curves, provided by the manufacturer, of torque, power and specific fuel consumption at full load, from 3000 to 8000 rpm: using analytical formulas, the brake mean effective pressure (BMEP) has been calculated and, combining it with the engine geometrical characteristics, it has been possible to reconstruct the ideal combustion cycle on a pressure-volume diagram. At this point, three different lubricant oils have been chosen, in order to understand how their chemical and physical characteristics can

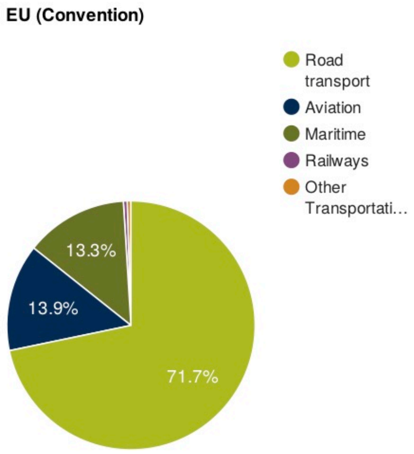
influence the lubrication of the seals and the performance of the support bearings: SAE 0W30, SAE 10W40 and a synthetic oil for 2 stroke applications.

The combination of the pressure previously obtained with the rotational speed, gave the possibility to compute the force exchanged between apex seals and engine housing: it was then used, in a lubricated contact model based on Stribeck's formula, to compute the power losses for rubbing friction and to compute the apex seals wear, according to the Archard's model. Next, an analytical model has been applied to support bearings and to shaft-rotor bearing in order to compute the total power losses, so the mechanical efficiency: the results have been compared to the ones obtained in the case in which the roller bearings are substituted by bushings, in the same operating conditions and with the same lubricant oil. The final step was to understand how all these aspects influence the fuel consumption and, since oil and fuel are injected in a blend at 2% in volume, the oil consumption.

Finally, the complete mathematical model, obtained for each lubricant oil, has been used to test the possible application of this rotary engine as range extender on both NEDC and WLTC, to understand what could be the increment of drivable range and what could be the duration of the apex seals after which a substitution is needed, compared to a common reciprocating engine used as range extender.

2. Introduction

2.1 From internal combustion engines to hybrid powertrains



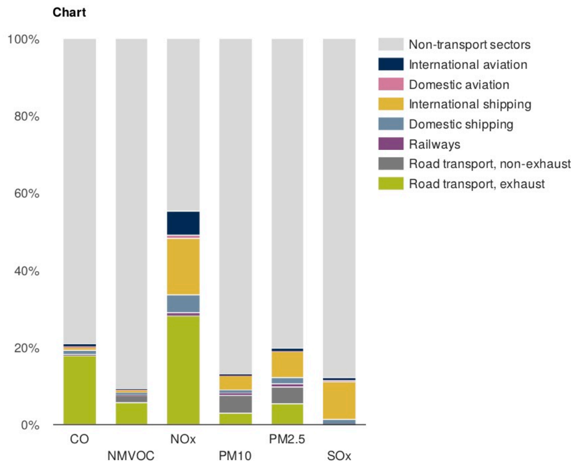


Figure 1: GHG emission transport share [1] **Figure 2**: main pollutant emission transport share [2]

Road transportation accounts for more than the 71% of the total emission of green house gases and between 5% and 28% of the four main pollutants (CO = 17,97%, NO_x = 28,12%, HC = 5,57% and PM = 17,5%): the vast majority of these pollutants are products of the combustion process of the internal combustion engines [1] [2]. This problem, combined with the decreasing availability of crude oil, is forcing the automotive industry to find new, sustainable and green ways to satisfy the increasing request of cars in the entire world, without worsening its pollution.

The easiest way to reduce the combustion-connected pollutants is to eliminate the combustion process itself: several automotive companies decided to substitute the internal combustion engine (ICE) with an electric motor (EM) endowed with a battery pack to provide the required energy. This electric system is nominally able to meet all the requirements previously met by any ICE but this is not always true during the daily usage: batteries' performance, and the drivable range in particular, of a battery pure electric vehicle (BEV) is strongly dependent on the environmental temperature, on the driving conditions, on the numbers of electric devices in use and on several other aspects. BEVs result to be more sensitive to these factors than any ICE based vehicle: the drivable range of a BEV can be drastically reduced (-41%) in extremely cold conditions (-6°C) and this is due to the fact that the devices needed to heat up the passenger compartment are electrically driven, while on a common car the same job is met by the heat produced by the engine [3] [4].

In order to satisfy the driver requirements, in term of range and performance, as well as the environmental ones, the automotive electrification found a less drastic approach: instead of substituting the ICE with a battery-electric motor pack, the two solutions have been combined together, in order to keep the benefits of both, in a so called hybrid vehicle (HEV).

The two most common architectures for HEV are "parallel hybrid" and "series hybrid": the main difference stands in the fact that in parallel hybrids both ICE and EM are connected to the transmission, so both can give traction to the wheels, while in series hybrid just the EM is connected to the transmission and the ICE is connected to an alternator used to generate energy for batteries and motor (range extender, RE).

The main drawbacks of combining the two elements on a single vehicle are the net increment of weight and the reduction of available space for the passenger compartment, since the installation of ICE and EM needs space: the solution is to find a hybrid powertrain characterized by a large power to weight ratio and that can ensure a long drivable range.

2.2 Wankel engine

In the following will be analysed the specifics that characterize a Wankel rotary engine, its main advantages and disadvantages, in order to describe why it could be suitable for a vehicle range extender application.

2.2.1 Design and working principle

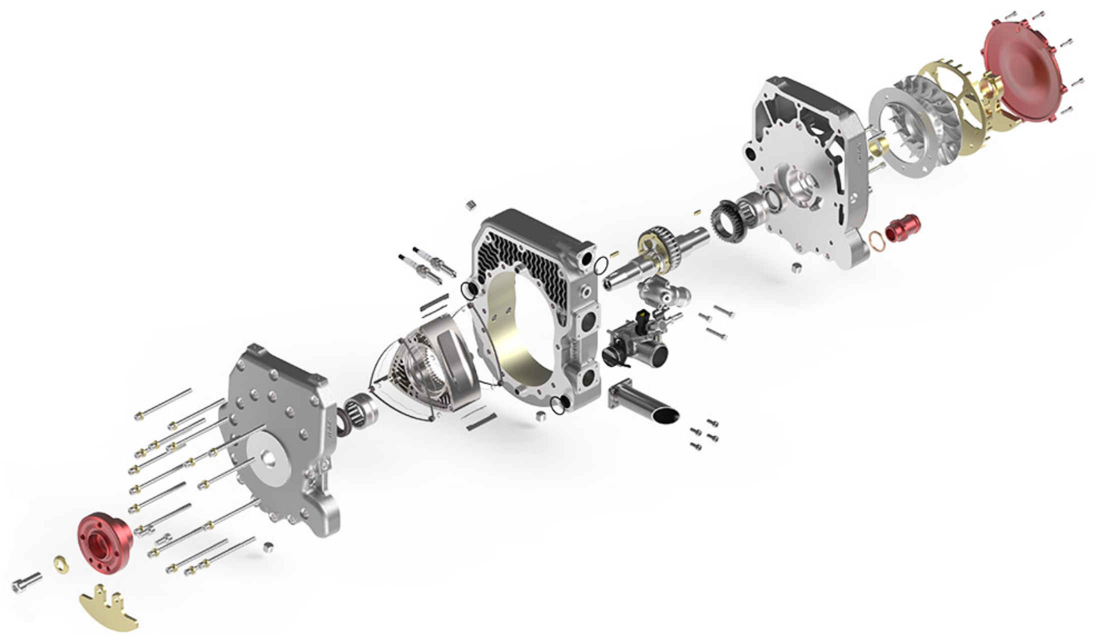


Figure 3: Exploded view of a Wankel rotary engine [68]

The Wankel engine is a type of internal combustion engine that uses an eccentric rotary design to convert pressure into rotating motion: it is mainly constituted by a epitrochoid-shaped housing, a rotor and an eccentric shaft, connected by a couple of toothed gears. With a shape similar to a Reuleaux triangle with the sides being somewhat flatter, the rotor moving inside the epitrochoidal housing is able to complete three Otto cycles per its revolution. The toothed gears allow the eccentric shaft to turn three times faster than the rotor, delivering one power pulse per shaft rotation: these geometrical characteristics, together with the fact that inside a Wankel there are just rotating parts, and not reciprocating ones, make this engine well suited for those applications in which simplicity, smoothness, compactness, high revolutions per minute, and a high power-to-weight ratio are required.

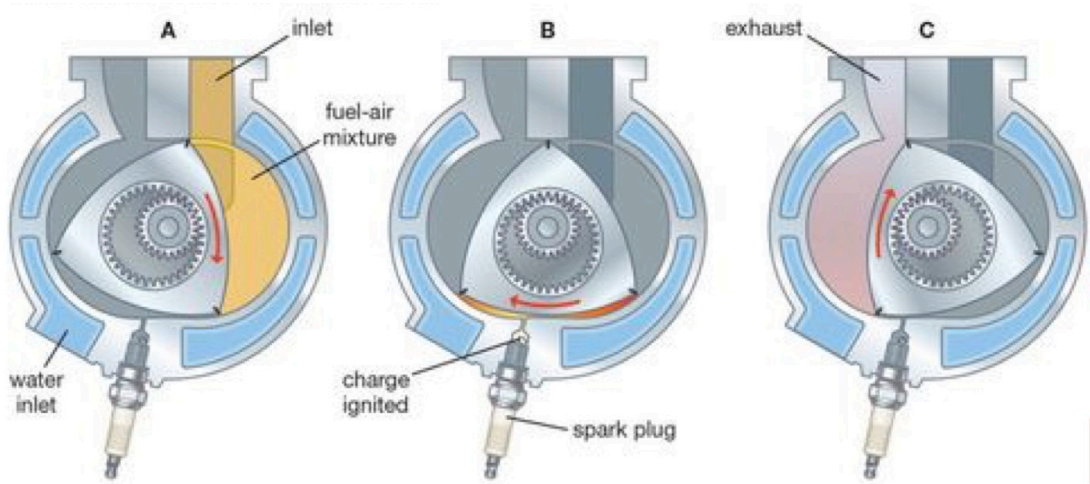


Figure 4: Wankel engine operating cycle

2.2.2 Advantages

The geometrical aspects and the operative specifics that characterize a Wankel rotary engine are the two main features that give to this engine several advantages.

The shape of housing and rotor ensures a quasi-overlap of the power strokes, which generate the typical smoothness and the quick reaction to power increases when the demand arises; moreover, thanks to the combustion chamber shape and features, the fuel octane requirements of Wankel engines are quite low (70-82) [6].

Then, for what concerns the fresh charge intake and the expulsion of the exhaust gases, it is possible to notice that the Wankel engine is characterized by the absence of valves or complex valve trains, reducing the weight, the number of moving parts and the pumping losses through the choking valves, substituting them with cuts into the walls of the rotor housing. Farther, the rotor rides directly on a large bearing on the output shaft, so there are no connecting rods and no crankshaft: the elimination of reciprocating masses, and the elimination of the most highly stressed and failure prone parts of piston engines, give the Wankel engine high reliability, a smoother flow of power, lower weight and less NVH connected problems.

So to recap, the main advantages are [5]:

- High power to weight ratio
- Easy to package in small engine space
- No reciprocating parts
- Able to reach high revolutions per minute
- Operating with almost no vibration
- Cheap to mass-produce, because the engine contains few parts [7] [8]
- Wide speed range giving great adaptability
- Can use fuels of wider octane ratings
- The oil sump remains uncontaminated by the combustion process, so no oil changes are required. The oil in the shaft is totally sealed from the combustion process so the oil for Apex seals and for eccentric shaft lubrication are separate.

2.2.3 Disadvantages

Although the triangular rotor, coupled with a epitrochoidal housing, give the possibility to overlap three Otto cycles, conferring to the Wankel engine several advantages, the same aspects make hard to achieve good results in term of lubrication and combustion, with consequent low efficiencies in term of fuel economy, oil consumption [9] [10].

The characteristic shape of the Wankel rotor and its working principle, require apex seals to separate the three different chambers, during the whole cycle: this induces that the two sides of the seals are exposed to different blend at different temperatures, making difficult to optimize the lubrication of the rotor. Moreover, being the apex seals pushed against the housing wall by a preloaded spring, their position is not fix, but it is influenced by the centrifugal forces: at light-load operation, imbalances in centrifugal force and gas pressure can cause the seal to lift off the surface, resulting in combustion gas leaking into the next chamber, due to the gaps developed between the apex seal and troichoid housing, and excessive wear due to the impacts with the housing. For these reasons, rotary engines tend to be over-lubricated at all engine speeds and loads, having high oil consumption, excessive carbon formation and emissions from burning oil.

The geometry of the rotor-housing system also leads to have a long, thin, and moving combustion chamber, which impairs a fast and complete combustion and leaving a high number of unburned hydrocarbons at high rpm.

Several disadvantages can be partially solved adding a second spark plug or using different fuels, but the ones mainly dependent on the geometry or on the operating principle are:

- Rotor sealing
- Apex seal lifting
- Apex wear
- High oil consumption
- Slow combustion
- Bad fuel economy
- High emissions

2.2.4 Vehicle range extender

Although Mazda of Japan, the only automotive Company producing Wankel rotary engines in series, left the motor industry with no production cars using this engine, ceasing the production of direct drive Wankel engines within their model range in 2012, many automotive companies worldwide apply this rotary engine as range extender in series-hybrid electric cars [11]. Audi proposed as prototype the A1 e-tron, that incorporated a small 250 cm^3 Wankel engine; FEV Inc promoted their electric version of the Fiat 500, in which a Wankel engine is used as a range extender; Valmet Automotive, in Finland, revealed a prototype car named EVA, incorporating a Wankel powered series-hybrid powertrain. Moreover, several engineering Companies design small-displacement Wankel engine based powertrains to be applied in HEV, such as Wankel SuperTec in Germany, Aixro Radial Engines in UK and AVL in Austria.

Mitsueo Hitomi, the global powertrain head of Mazda, stated [12]:

"A rotary engine is ideal as a range extender because it is compact and powerful, while generating low-vibration".

Due to its compact size and the high power to weight ratio, the Wankel engine has been proposed for electric vehicles as range extenders to provide supplementary power when electric battery levels are low. Such an engine, used only as a generator, has packaging, noise, vibration and weight distribution advantages when used in a vehicle, maximizing interior passenger and luggage space. The engine-generator may be at one end of the vehicle with the electric driving motors at the other, connected only by electric cables. There may be a combination between a large battery pack, charged from the grid to give pure electric run during the charge depleting (CD) mode, and the engine performing the dual function of range-extender and charger, when the battery charge is below a certain threshold, SOC-H in figure 4, in order to keep the state of charge of the battery around a predefined value, SOC-T, working in charge sustaining (CS) mode.

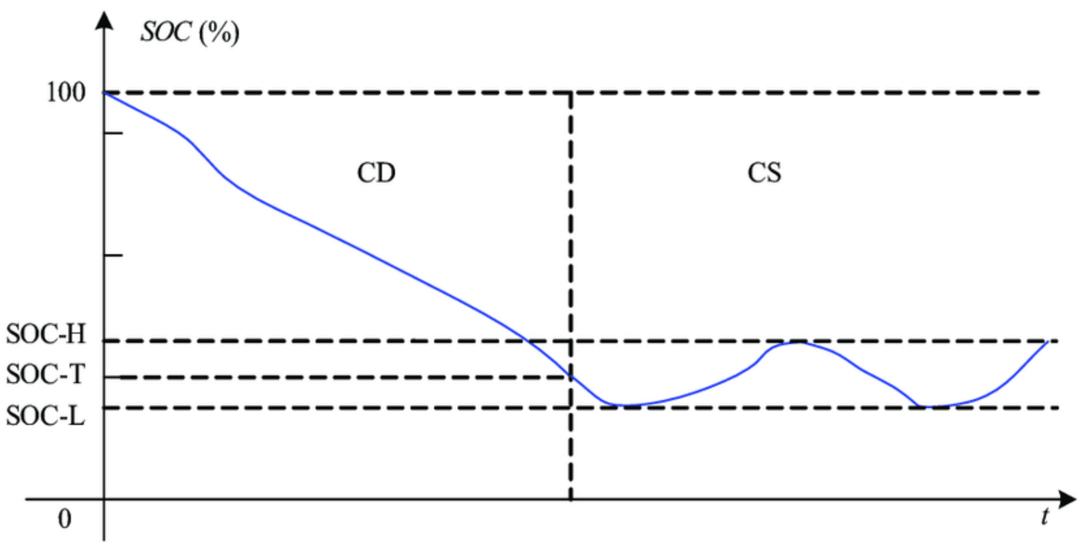


Figure 5: Charge depleting and charge sustaining mode

The rotary engine has very good dynamic performance, but it is not so good from a fuel economy point of view, when its rotational speed tends to vary continuously.

With a range extender application, it is possible to use a Wankel engine at fixed speed, in order to generate the power required to charge the batteries, while ensuring high efficiency and acceptable fuel consumption. Since the vehicle power demand varies accordingly to the driving conditions, batteries, electric motor and range extender have to work simultaneously to satisfy the power requested while keeping the batteries around a certain charge level. For this reason, the internal combustion engine generally works at two fixed and different operating conditions:

- ECO: lower rotational speed, lower power produced, higher efficiency, high fuel economy
- BOOST: higher rotational speed, higher power produced, lower efficiency, low fuel economy

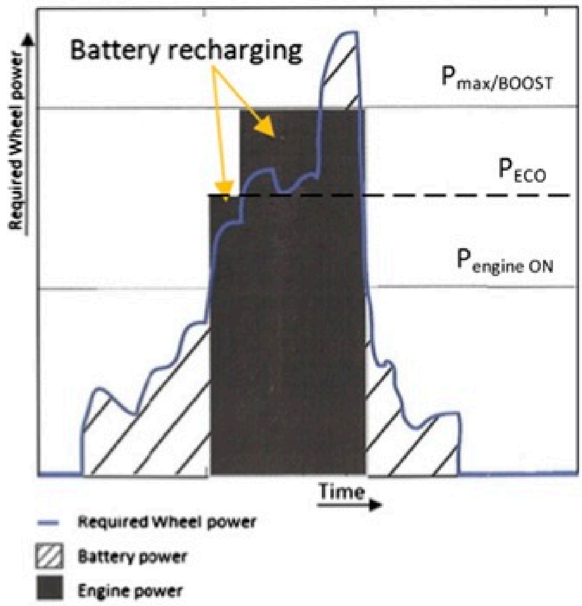


Figure 6: Example of energy management strategy in CS mode [13]

The working principle is based on the idea that in CD mode the internal combustion engine is OFF and the batteries discharge supplying energy to the electric motor; during CS mode, if the power required is below a given threshold ($P_{engine\ ON}$), the engine is still OFF, while if it is over that threshold, the engine switches ON and its rotational speed has to be set according to the power demand, working in ECO or BOOST condition [13].

While in CS mode, in order to maximize the fuel economy and to minimize the charging-discharging inefficiencies of the batteries, the power produced by the range extender is used to meet the required wheel power, supplying directly the electric motor, and just the surplus is used to recharge the batteries, both in ECO and BOOST mode.

The batteries discharge, instead, occurs if the power demand is lower than the power level needed to switch ON the range extender ($P_{engine\ ON}$), or over the power produced in BOOST mode, when ICE and EM have to work together (dashed are in figure 6).

2.3 Lubrication principles

Lubrication plays a fundamental role in the life expectancy of an engine. Without it, an engine would overheat and seize very quickly. Lubricants help in reducing these problems extending the life of the engine and of its components. A proper lubrication level is required wherever moving or sliding parts are present: apex sliding along the troichoid housing, rotor motion on engine bearing and shaft rotation in support bearings [17].

2.3.1 Aims of lubrication

Delivering the right amount of lubricant to moving parts is fundamental to meet several requirements:

- Generation of a stable oil film between sliding surfaces: a continuous oil film between the two surfaces is required to ensure a low coefficient of friction and to distribute the applied load. Bearings work in hydrodynamic lubrication regime.
- Provision of reliable engine operation in a wide temperature range: internal combustion engines are generally lubricated by means of oil. It results more suitable than grease, because it tends to have a better behaviour at high relative speed between the parts to be lubricated, it has good cooling effects, it applies a low resistant torque, it has good cleaning capabilities and results quite easy to be substituted. Oil viscosity strongly depends on its temperature. When an engine starts at low temperature the oil is viscous, ensuring a thick lubricant film. If the viscosity is too high the oil will not be able to flow to the sliding parts and the risk of seizure increases. On the other hand oil viscosity in a heated engine is low. The oil flows easily, but the lubricant film is thin, increasing the risk of metal-to-metal contact. This can cause excessive wear, overheating and fatigue failure of the components [18].
- Cleaning the engine and protecting its parts: lubricant oil protects metal parts from combustion gases, that containing water vapours and other chemically active gases may cause corrosion. Moreover, some combustion products dissolve in the oil increasing its acidity. Such oil may become aggressive to the metal components in contact with it. Combustion gases past through the apex seals, from one chamber to another, contain some amount of unburned

carbon which may deposit on the seals, forming a sludge. The sludge clogs oil passages and clearances decreasing lubrication of the engine parts.

- Cooling the engine parts: the combustion process, together with the friction generated, heat up the metal components. Heat must be removed from the engine in order to prevent its overheating: since not all the parts can be reached by the cooling fluid, part of heat energy is taken by the engine oil.

2.3.2 Wankel lubrication

Playing the lubrication a fundamental role, the separation of relative moving parts must be ensured at each operating condition. When the relative speed between the surfaces tends to raise the oil temperature, making vary its viscosity and density, a forced lubrication system is required. For this reason, all the internal combustion engines, as well as the Wankel, are endowed with a proper oil circuit.

Being the Wankel rotary engine characterized by the absence of intake valves, connecting rods and camshafts, its lubrication circuit appears quite simple.

A pump sucks the oil from the sump, pressurizes it and sends it firstly to a filter and then to the support and rotor bearings, by means of suitable channels in the eccentric shaft [23]. For what concerns the lubrication of the housing and of the apex seals, the Wankel engine has a working principle more similar to the one of a 2 stroke engine: the oil can be injected or inside the intake duct, so mixed with fuel before entering the chamber, or directly inside the housing to lubricate the walls and the rotor. A dosed oil spray is injected at each shaft rotation, to ensure a low-friction sliding between apex seals and housing walls, to reduce the Apex wear and to limit as much as possible the gas leakage from one chamber to another.

Thanks to this design, the oil sump remains uncontaminated by the combustion process, so no oil changes are required. The oil in the shaft is totally sealed from the combustion process so the oil for Apex seals and for eccentric shaft lubrication are separate. Theoretically, it is possible to use two different kinds of oil, one for the housing and one for the bearings. This solution would ensure several benefits in term of pollutant formation and efficiency but it is not used in practice because it would require the application and the installation of two oil circuits, frustrating the advantages of compactness and low weight of the Wankel.

The main disadvantage of this system is that, being the oil injected and completely burned together with the fuel, the Wankel is characterized by a very high oil consumption and consequent high pollutant emission.

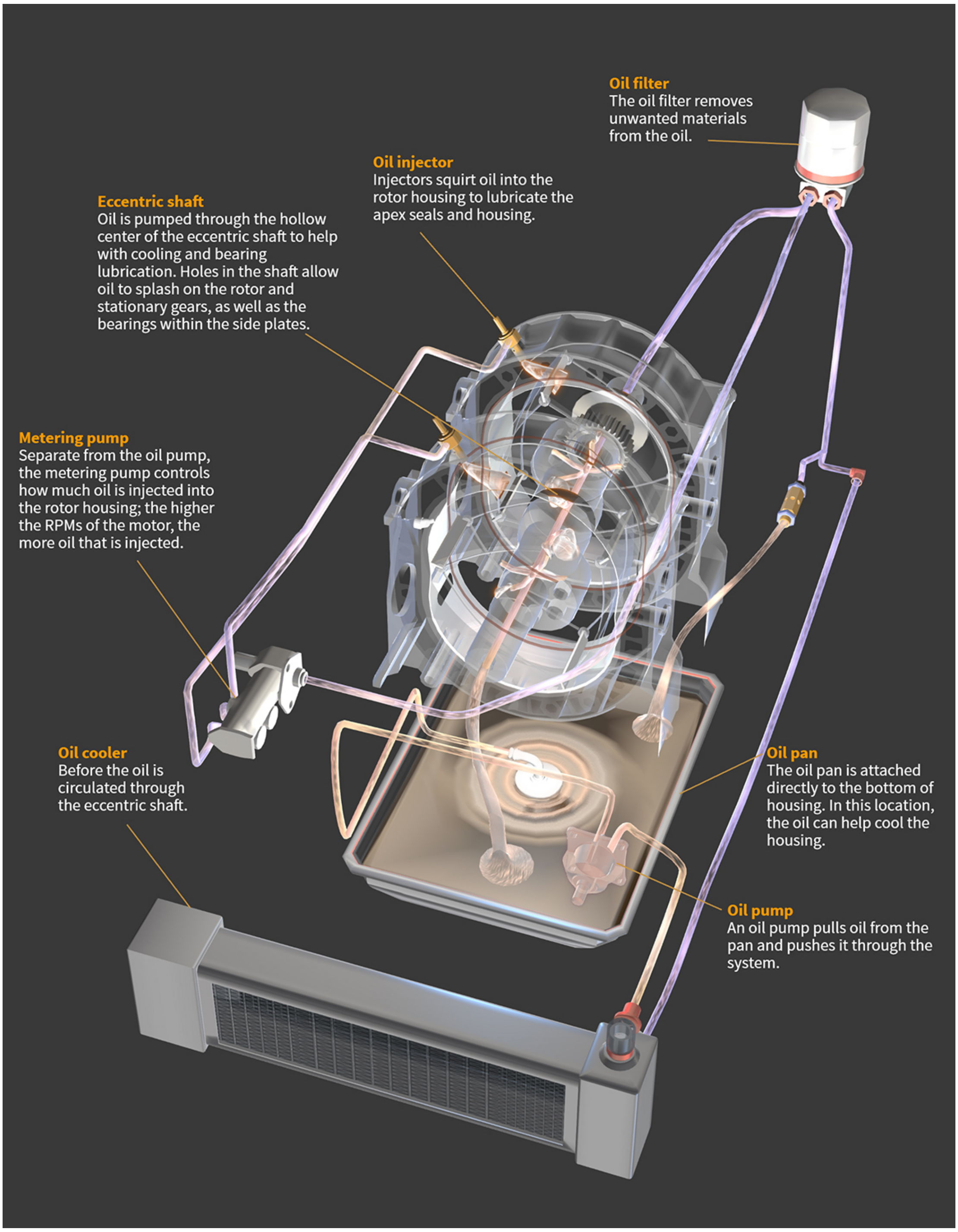


Figure 7: Wankel engine lubrication system [24]

2.3.3 Lubrication regimes

For each engine component, the required lubrication is obtained by means of two actions: physico-chemical and physical.

The physico-chemical action is found when the surfaces to be lubricated have a low relative speed and just a thin film of oil compensates the asperities of the surfaces. This condition is called “boundary lubrication regime” and it is strongly influenced by lubricant characteristics such as the unctuousness and the molecular cohesion. This regime is the most undesirable since it is characterized by high coefficient of friction, high energy losses, increased wear, possibility of bearing seizure and non-uniform distribution of load. Conditions for boundary lubrication are realized mainly at low speed and high loads, such as engine start and shutdown: in this regime the oil film thickness is lower than the surface roughness ($h < Ra$) as shown in figure 8.

Additives in the lubricant prevent seizure caused by direct metal-to-metal contact between the parts in the boundary lubrication regime.

When the relative speed between the parts increases, the lubrication becomes hydrodynamic. The separation of the surfaces is obtained by means of the physical action: thanks to the velocity, a pressure increment inside the lubricant separates the surfaces, favouring the relative sliding. The presence of a pressurized thick film of oil ($h >> Ra$) strongly reduces metal-to-metal contacts and the consequent losses, wear and seizure possibility; this lubrication regime, in increasing relative speed, tends to raise the hydrodynamic friction. In this condition, the two parts to be lubricated must have similar dimensions in order to avoid a lateral escape of the pressurized oil.

An intermediate lubrication condition ($h \approx Ra$) may cause intermittent metal-to-metal contacts between the friction surfaces at few high surface points (microasperities); this condition is called mixed lubrication regime [25].

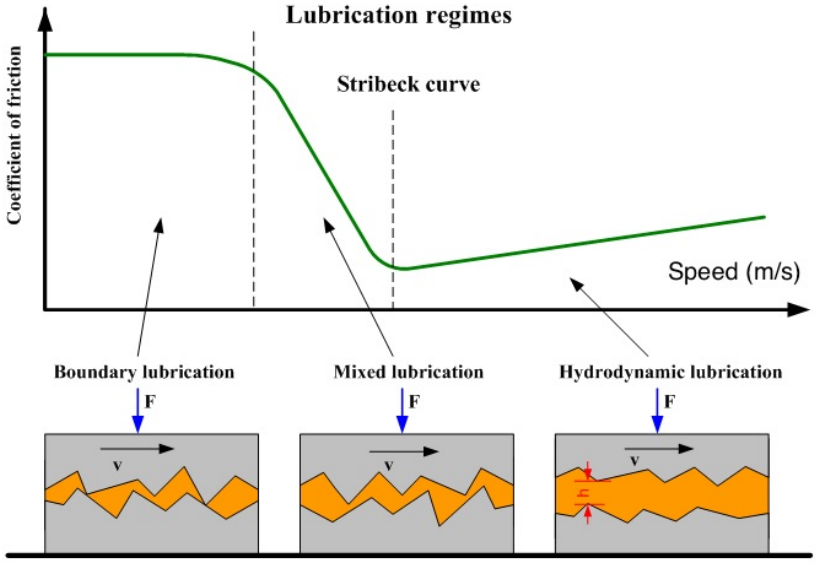


Figure 8: Lubrication regimes [25]

2.3.4 Lube oil classification

Since lubricants properties strongly influence the efficiency of the lubrication system, the performance of the engine, the emissions at the exhaust and the wear of the parts, it is good practice to understand how oils are classified.

Lubricants classification can be based on different parameters, such as their performance or their physico-chemical properties. According to this, it is possible to identify the Society of Automotive Engineering (SAE) viscosity grading system, which classifies oils according to their viscosity at different temperatures, the American Petroleum Institute (API) and the Associazione dei Costruttori Europei di Automobili (ACEA) classifications, which define oils according to the results obtained from bench tests [20][21].

2.3.5 SAE classification

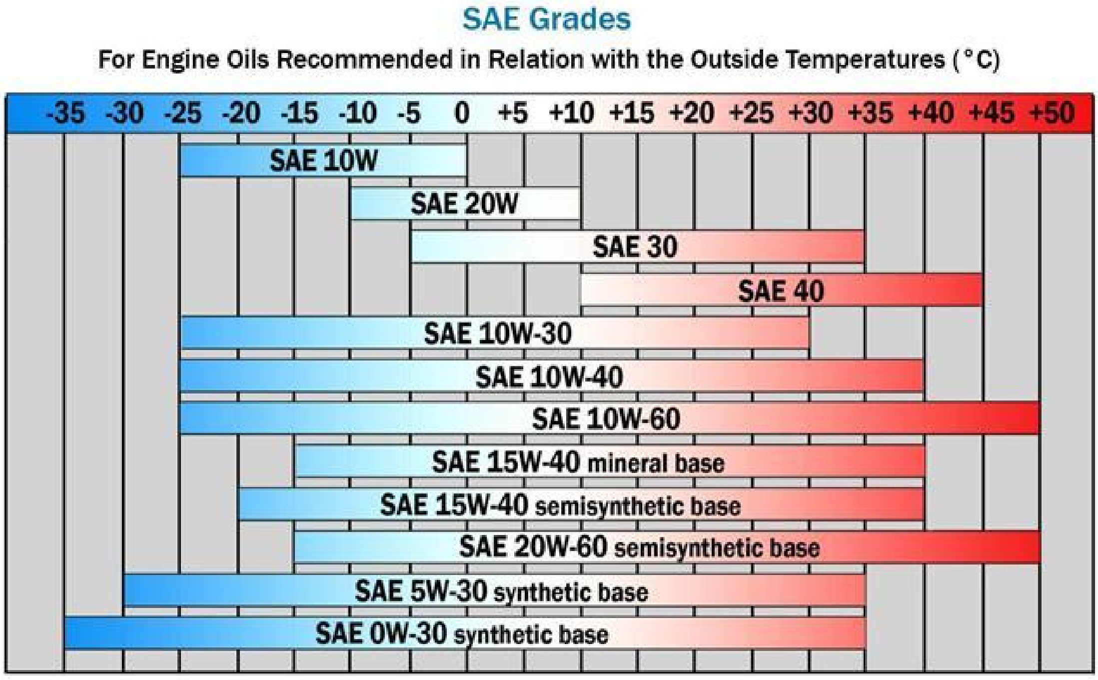


Figure 9: SAE grades for oil classification [64]

SAE establishes a viscosity grading system for engine oils. According to its viscosity grading system, all engine oils are divided into two classes: monograde and multigrade [19] [20] [21].

- Monograde engine oils are designated by one number that represents a level of the oil viscosity at a particular temperature. The higher the grade number, the higher the oil viscosity. Viscosity of lubricants designated with a number only, without the letter "W" (SAE 20, SAE 30, etc.), was specified at the temperature 212°F (100°C). These engine oils are suitable for use at high ambient temperatures. Viscosity of engine oils designated with a number followed by the letter "W" (SAE 20W, SAE 30W, etc.) was specified at the temperature 0°F (-18°C). The letter "W" means "winter" so these lubricants are used at low ambient temperatures.
- Multigrade engine oils viscosities are specified at both high and low temperature. They are designated by two numbers and the letter "W" (SAE 5W30, SAE 15W30, SAE 20W50, etc.): the first number specifies the oil viscosity at cold temperature while the second number specifies the oil viscosity at high temperature. For example: SAE 10W40 oil has a low

temperature viscosity similar to that of SAE 10W, but it has a high temperature viscosity similar to that of SAE 40. Multigrade oils can be used in a wide temperature range.

2.3.6 API classification

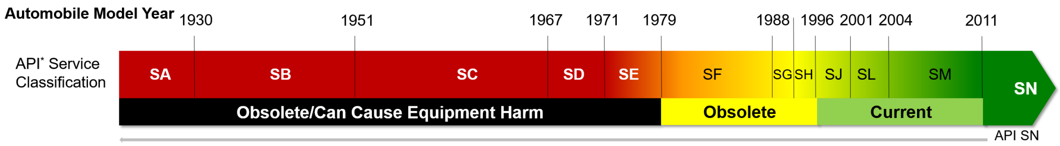


Figure 10: API oil classification [65]

API designates for engine oils two categories identified by a couple of letters.
The first letter defines the kind of engine for which the oil is designed: "S" means Service, so spark ignition engines, and "C" means Commercial, so diesel engines.
The second letter defines the severity of the test at which the oil is subjected, so the expected performance level: further is this letter in the alphabet, higher will be the performance. According to this, it is possible to say that a more recent API specification, being more severe, will satisfy the older ones as well.

Table 1: API oil classification

Classification	Type of engine	Performance
S	Spark ignition	SA, SB, SC, SD, SE, SF, SG, SH, SJ, SL, SM, SN
C	Compression ignition	CA, CB, CC, CD, CE, CF-4, CG-4, CH-4, CI-4, CJ-4

API SN has been introduced in 2010. It is based on tests to ensure a higher oxidation resistance, higher protection from deposits, higher protection from wear, higher performance at low temperature and the attitude to fuel economy.
API CJ-4 has been introduced in 2010. It is based on tests to verify the compatibility of the lubricant with the exhaust gas recirculation (EGR) system and with other exhaust gas management systems. Engine oils classified in this category, ensure high wear protection, high soot control and good viscosity maintenance [20] [21]. API SN and API CJ-4 are suitable for 4 stroke applications. For what concerns 2 stroke applications, different are the requirements and the specifications. These engines are generally characterized by the absence of oil recovery, so the lubricant takes part to the combustion process. For this reason, oils for 2 stroke applications, classified as

TC or CF-2, must ensure a stable blending with fuel, high lubrication capabilities, low soot formation and low ash content [22].

2.3.7 ACEA classification

ACEA designates oils by means of four different standards, based on the kind of engine and vehicle application. This classification is based on a letter followed by a number, which define respectively the engine typology and the different usages and applications in a given class, linked to performance levels.

Categories “A”, for spark ignition engines, and “B”, for compression ignition engines, are designated for low duty vehicles. Category "C" (Catalyst Compatible) is referred to low duty vehicles too, but more it requires the compatibility of the lubricant with after-treatment systems. Category "E" is dedicated to heavy duty and commercial vehicles.

Differently from API, a higher number in the classification doesn’t necessarily mean a higher performance level. ACEA classification is based on the results obtained from engine tests [20] [21]:

Table 2: ACEA oil classification

Classification	Application
A/B	*Oil for spark ignition and compression ignition engines*
A1/B1	Low viscosity oils, with low HT/HS (>2,6 <3,5 mPa.s), with fuel economy characteristics
A3/B3	Oils suitable for high performance engines, severe usage and long maintenance interval
A3/B4	Oils suitable for direct injection engines; satisfy requirements of A3/B3
A5/B5	Low viscosity oils, with low HT/HS (>2,9 <3,5 mPa.s), with fuel economy characteristics. Suitable for severe usage and long maintenance interval
C	*Oils compatible with catalytic systems*
C1	Stable oils for engine with after-treatment systems, with extra fuel economy characteristics, low saps (ash < 0,5%) and HTHS min = 2,9 mPa.s.

C2	Stable oils for engine with after-treatment systems, with fuel economy characteristics, mid saps (ash < 0,8%) and HTHS min = 2,9 mPa.s.
C3	Stable oils for engine with after-treatment systems, with low fuel economy characteristics, mid saps (ash < 0,8%) and HTHS min = 3,5 mPa.s.
C4	Stable oils for engine with after-treatment systems, with extra fuel economy characteristics, low saps (ash < 0,5%) and HTHS min = 3,5 mPa.s.

In order to achieve the highest performance level and to avoid a premature oil breakdown, several additives are combined with the base oils [19]:

- Corrosion inhibitors: added to engine oils in order to provide protection of metallic parts.
- Detergents: added to the engine oils to remove the sludge from the surfaces.
- Dispersant: added to the engine oils, help to maintain the removed sludge and other contaminants in form of fine suspension permitting engine functioning between the oil changes.
- Anti-foaming agents: added to the engine oils to prevent foaming. Engine oil may entrap air forming foam and loosing their effectiveness.

3. Methodology

3.1 Engine data acquisition

The acquisition of the Rotron RT300 full load brake power, torque and specific fuel consumption is the base of following models, computations and results.

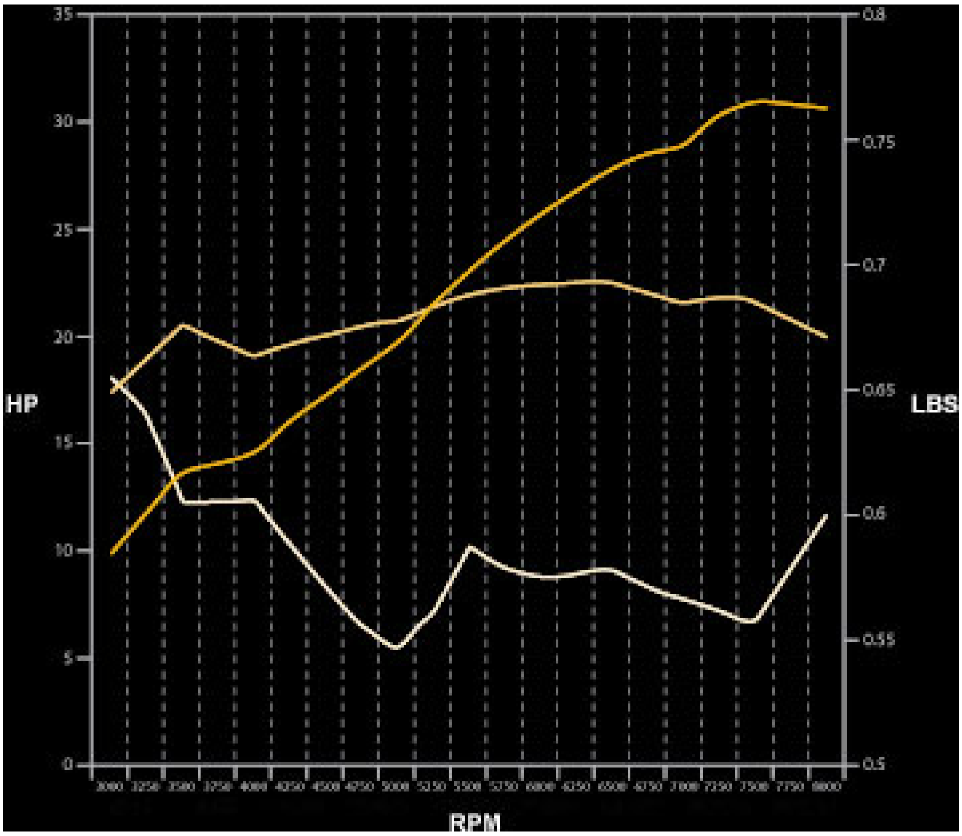


Figure 11: Rotron RT300 data [26]

The bench test results, provided by the manufacturer, are sampled as function of the engine rotational speed, from 3000 to 8000 rpm. These data must then be converted into the metric reference system:

$$Power\ [Hp \rightarrow kW] : power \bullet 0{,}7457 \tag{3.1}$$

$$Torque\ [Lbs\ ft \rightarrow Nm] : torque \bullet 1{,}355818 \tag{3.2}$$

$$Specific\ fuel\ consumption \left[\frac{Lbs}{Hp\ Hr} \rightarrow \frac{g}{kW\ Hr}\right] : BSFC \bullet 728{,}96 \tag{3.3}$$

Since the oil is premixed with fuel in a 2% blend [34], once obtained the fuel consumption, it is possible to compute the oil consumption as follows:

$$Specific\ oil\ consumption\left[\frac{g}{kW\ Hr}\right] : Specific\ fuel\ consumption \bullet 0{,}02 \quad (3.4)$$

The obtained data are taken as reference to develop the whole mathematical model and this implies [36]:

- The usage of two roller bearings and one ball bearing as support between shaft and housing
- The usage of one needles bearing as rotor bearing
- The usage of the synthetic oil for two stoke application
- The usage of the following geometrical characteristics [35]:

Table 3: Engine geometrical data

Rotron RT300 geometrical data		
Eccentricity [mm]	e	11,6
Generating radius [mm]	R	73,0
Housing Depth [mm]	b	68,2

The obtained model was then modified using:

- Lube oil SAE 0W30 instead of the synthetic oil for two stoke application
- Lube oil SAE 10W40 instead of the synthetic oil for two stoke application
- Bushings instead of rolling bearings

3.2 P-V diagram reconstruction

The Pressure-Volume diagram is used to represent the ideal combustion thermodynamic cycle followed by the Rotron RT300 Wankel rotary engine.

Firstly the mean effective pressure (MEP) is computed using the formulas 3.5, 3.6 and 3.7 [38], referring to the full load curve for each rotational speed:

$$Power = \frac{displacement \bullet MEP \bullet rpm}{60 \bullet 100}\ [kW] \quad (3.5)$$

$$MEP = \frac{60 \bullet 100 \bullet Power}{displacement \bullet rpm} \bullet 10^\wedge 2\ [BAR] \quad (3.6)$$

$$Displacement = 3 \bullet sqrt(3) \bullet R^2 \bullet b \bullet \left(\frac{e}{R}\right) = 300\ [cm^3] \quad (3.7)$$

Then the pressure versus volume trend, for each rotational speed at full load, is obtained imposing the parameters in table 4 in formulas 3.8 to 3.12:

- *Intake*

$$P = p_{min} \tag{3.8}$$
$$V = V_{min} \rightarrow V_{max}$$

- *Compression*

$$P = p_{min} \bullet \left(\frac{V_{max}}{V_{max} \rightarrow V_{min}}\right)^{k_{compression}} \tag{3.9}$$
$$V = V_{max} \rightarrow V_{min}$$

- *Combustion*

$$P = p_{at\ end\ of\ compression} \rightarrow p_{max} \tag{3.10}$$
$$V = V_{min}$$

- *Expansion*

$$P = p_{max} \bullet \left(\frac{V_{min}}{V_{min} \rightarrow V_{max}}\right)^{k_{expansion}} \tag{3.11}$$
$$V = V_{min} \rightarrow V_{max}$$

- *Exhaust*

$$P = p_{at\ end\ of\ expansion} \rightarrow p_{min} \tag{3.12}$$
$$V = V_{max} \rightarrow V_{min}$$

Table 4: Cycle thermodynamic data

Parameter	**Meaning**
P_min [bar]	Intake phase pressure
P_max [bar]	Peak combustion pressure
$k_{compression}$	Polytropic compression exponent [37]
$k_{expansion}$	Polytropic expansion exponent [37]
V_min [cm^3]	Volume at the end of the compression stroke
V_max [cm^3]	Volume at the end of expansion stroke

The minimum and maximum volume are computed making reference to table 3 and using the formulas 3.13 and 3.14:

$$Vol_{max} = A_{min} \bullet b = \pi \bullet \left[\left(\frac{e}{R}\right)^2 + \frac{1}{3}\right] - \frac{sqrt(3)}{4} \bullet [1 - 6 \bullet \left(\frac{e}{R}\right)] \quad (3.13)$$

$$Vol_{min} = A_{min} \bullet b = \pi \bullet \left[\left(\frac{e}{R}\right)^2 + \frac{1}{3}\right] - \frac{sqrt(3)}{4} \bullet [1 + 6 \bullet \left(\frac{e}{R}\right)] \quad (3.14)$$

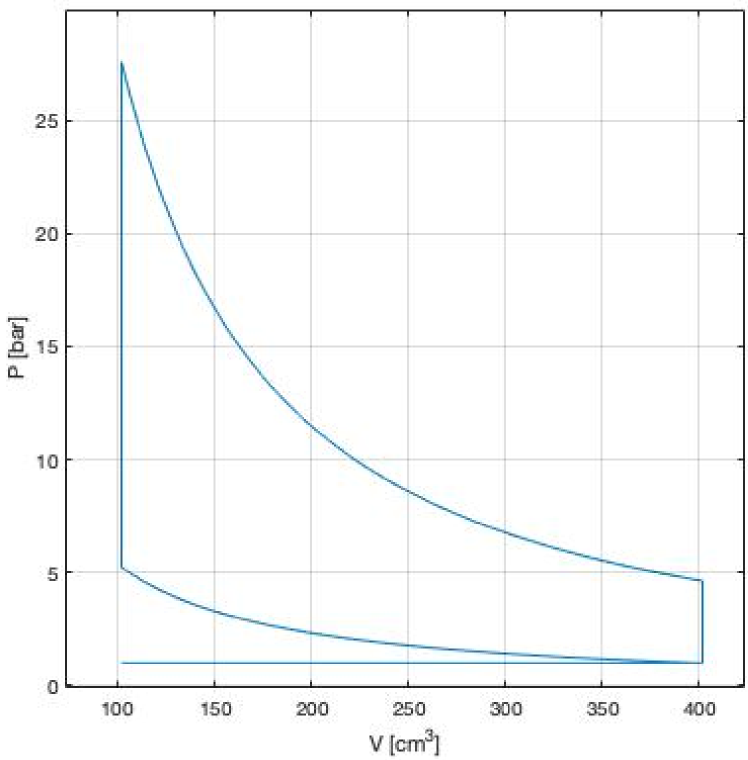


Figure 12: Ideal combustion cycle on P-V diagram

The used formulas represent the fuel-air combustion cycle followed by a common four stroke engine: since the Wankel rotary engines complete three Otto cycles for each rotor movement, the resultant thermodynamic cycle is the superposition of three ideal combustion cycles in 1020°.

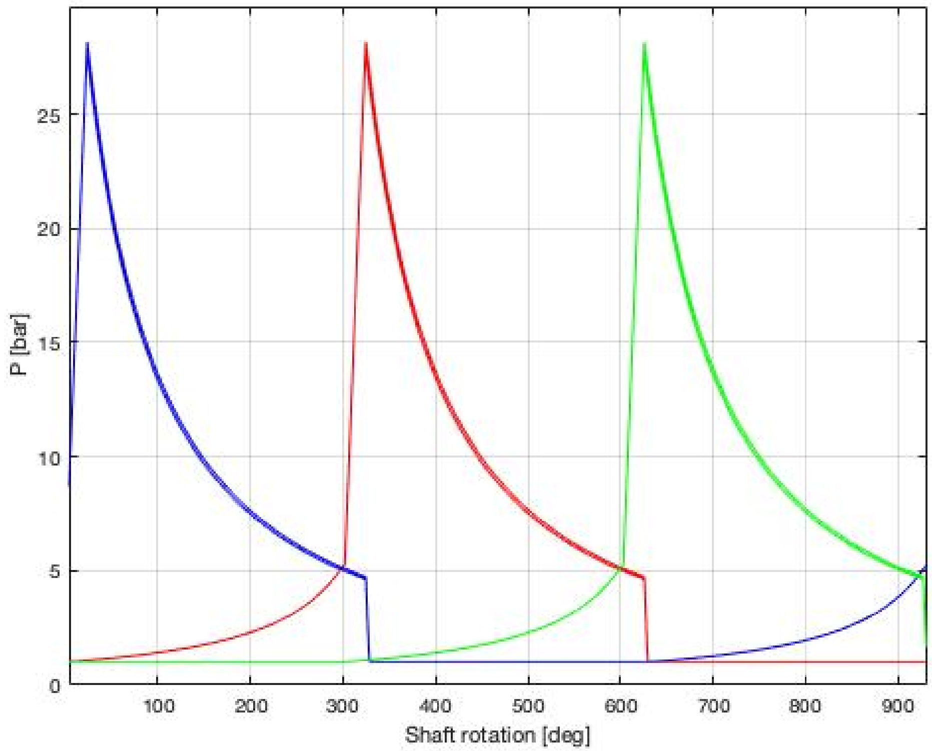


Figure 13: Ideal combustion cycle for Wankel rotary engines

3.3 Surface roughness and Stribeck's lubrication model

This part is dedicated to the description of the mathematical model used to represent the sliding contact between apex seals and the internal surface of the housing. The development of this model starts from the assumption that the two materials, of Apex and housing, are characterized by different surface roughness and the lubrication is in charge of interposing between the two a film of oil thick enough to compensate for the asperities. The idea is to use the Stribeck's model to compute the minimum oil film thickness and to compute the composite friction coefficient. Since the two surfaces have relative speed, a power loss due to friction occurs, but it occurs differently according to the fact that there is metal-to-metal contact or full film lubrication. Starting from the forces exchanged between the parts and the generation of the roughness profile, the expected output are the coefficient of friction (COF) in each point of the contact and the computation of the friction forces and friction power losses.

3.3.1 Surface generation

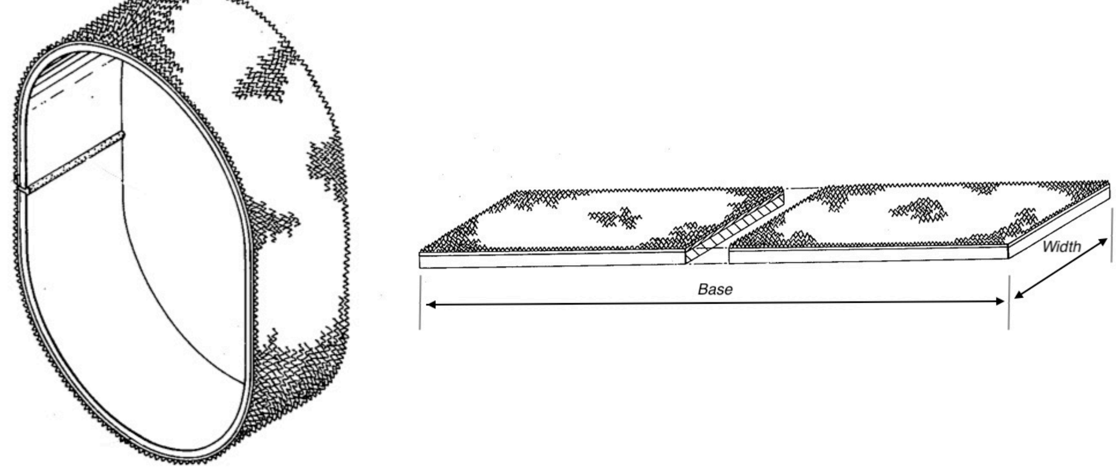


Figure 14: Housing surface representation [39]

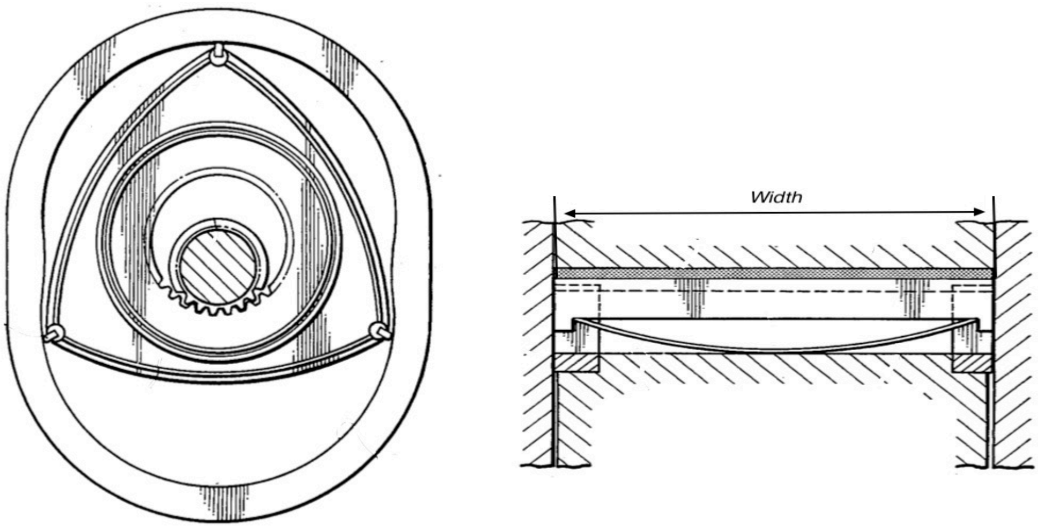


Figure 15: Apex representation [40]

The apex seals slide on the internal surface of the housing. For sake of simplicity the curved area of the housing is represented by an equivalent rectangle with the same dimensions and roughness. Since, in reality, the forces exchanged by Apex and housing are function of the angular position and this is not visible on a rectangle, the dependency on the angle is made explicit in the computation of the forces themselves. The apex seals are represented by an equivalent rectangle as well, in order to generate a constant contact path between its upper surface and the housing one.

The rough surfaces are generated by means of a MATLAB code, using the simple formulas 3.15 and 3.16, for both housing and Apex:

```
for x = 1:Per_S
    for y = 1:Width_S
        rugosity_S(x,y) = 2*rand*(R_S); %[micron]
    end
end
```
(3.15)

```
for x = 1:Per_A
    for y = 1:Width_A
        rugosity_A(x,y) = 2*rand*(R_A); %[micron]
    end
end
```
(3.16)

In script, the abbreviations "Per_A" and "Per_S" represent respectively the base dimensions of the Apex and stator, while "Width_A" and "Width_S" represent the width of the two parts. Basically, iterating the process in the whole area of the objects, it is possible to generate two surfaces with average roughness values of "R_A" and "R_S". These two values have been taken from literature [40] [41] and the mean value between the proposed ones has been chosen in order to develop a representative model.

The formulas 3.15 and 3.16 derive from the definition of surface roughness [42]: *"The measure of roughness Ra, expressed in microns, is the arithmetic average value of the deviations (taken as an absolute value) of the real profile of the surface with respect to the average line"*. Moreover, the surface roughness has a random trend with the average value of Ra: since on MATLAB the function "rand" has a mean value of 0.5, in formulas appears a multiplication by 2.

3.3.2 Stribeck's lubrication model

This analytical model is used to represent the lubricated contact between Apex and housing during the movement of the rotor. It starts acquiring the data relative to the roughness of the surfaces and computing the oil film thickness needed to compensate for the asperities. Combining this with the forces exchanged by apex seals and housing, it is possible to understand where metal-to-metal contacts occur and to compute the COF in each point of the contact path, according to the Stribeck's curve. The model gives as output the friction force and the friction power losses, as function of the lubrication regimes and of the roughness.

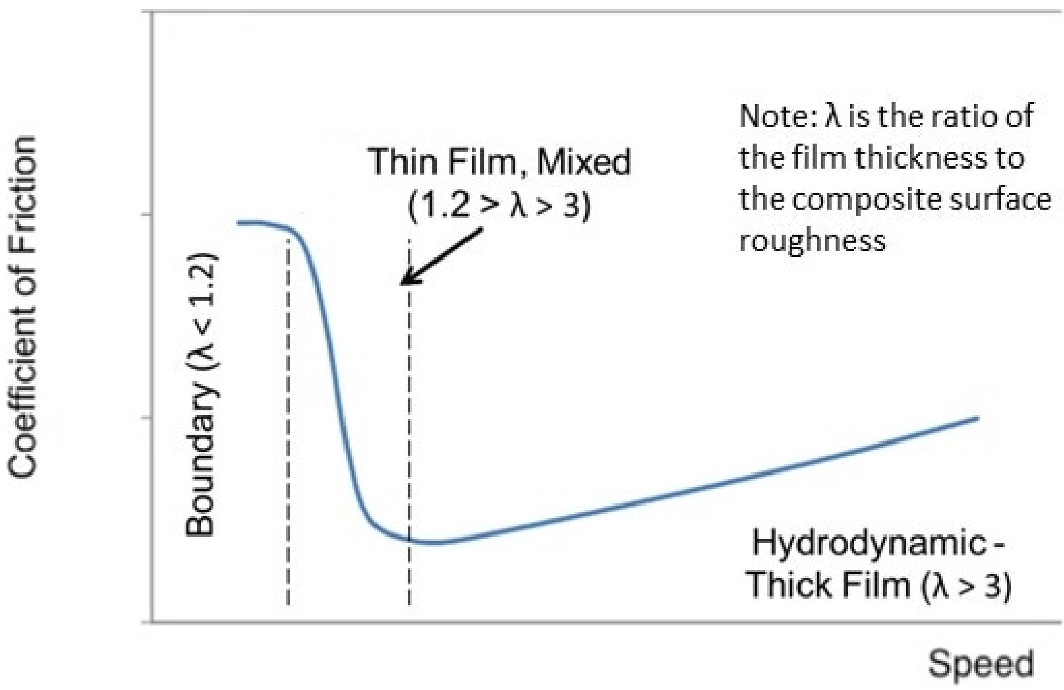


Figure 16: Lubrication regimes as function of the composite roughness

The model focuses on the optimization of the lubrication. In order to minimize the oil consumption, the power losses and the material wear, the following computations are based on a mixed lubrication regime (λ = 1,5). This ensures a low composite friction coefficient with consequent low power losses and wear.

The computation of the oil film thickness (OFT) is obtained by means of 3.17 and 3.18:

$$Composite\ roughness = \sigma = \sqrt{R_A^2 + R_S^2}\ [\mu m] \tag{3.17}$$

$$OFT = h = \sigma * 1.5\ [\mu m] \tag{3.18}$$

The formula takes into account the root mean square (RMS) value and it is set for a mixed lubrication regime: it is possible that in some points of the surface the OFT is enough to impair the metal-to-metal contacts, while in some others they can occur.

The forces exchanged between Apex and housing are function of several different aspects, physical and geometrical: pressure in adjacent chambers, preload of the Apex spring, rotational speed regime, width and base dimensions of the seal.

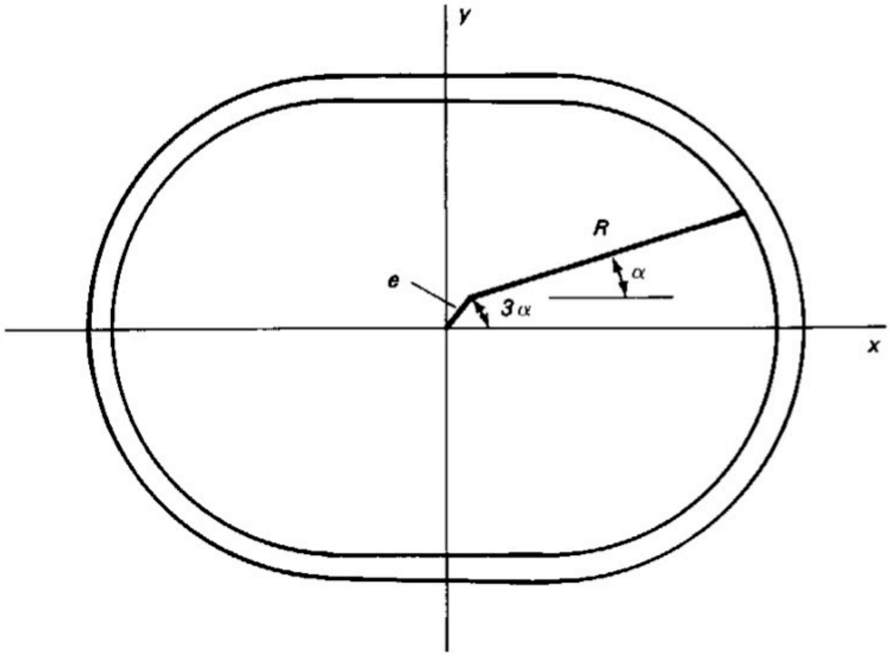


Figure 17: Nomenclature for epitrochoid parametric equations [38]

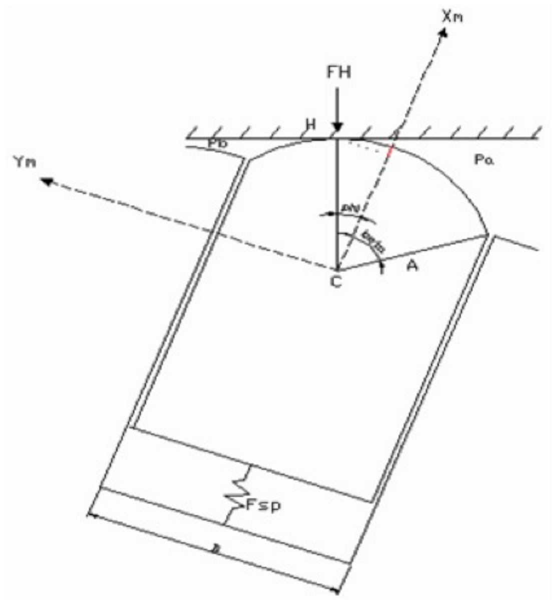


Figure 18: Schematic drawing of apex seal and exerted housing force [43]

$$Apex\ speed = v = \omega_{shaft} \bullet \frac{R}{3} \bullet \left(1 + 3 \bullet \frac{\left(\frac{e}{R}\right)\cos(2\alpha)}{\cos(\varphi)}\right) \tag{3.19}$$

$$\varphi = \arccos \frac{\left(1 + 3 \bullet \left(\frac{e}{R}\right)\cos(2\alpha)\right)}{\sqrt{\left(1 + 6 \bullet \left(\frac{e}{R}\right)\cos(2\alpha) + 9 \bullet \left(\frac{e}{R}\right)^2\right)}} \tag{3.20}$$

$$Apex\ radial\ acc = a_{rad} = \omega^2 \bullet R \bullet \left(\frac{1}{9} + \frac{e}{R}\cos(2\alpha)\right) \quad (3.21)$$

$$Apex\ tangential\ acc = a_{tang} = \omega^2 \bullet e \bullet \sin(2\alpha) \quad (3.22)$$

$$F_{py} = -P_a \bullet W \bullet [A \bullet cos(\varphi) - A \bullet \cos(\beta)] + P_b \bullet W \bullet [A \bullet co\,s(\varphi) - A \bullet \cos(\beta)] \quad (3.23)$$

$$F_{px} = -P_a \bullet W \bullet \left[\frac{B}{2} + A \bullet \sin(\varphi)\right] - P_b \bullet W \bullet [A \bullet \sin(\varphi) - \frac{B}{2}] \quad (3.24)$$

In which $2\beta = \sin^{-1}(B/2A)$ and W is the width of the Apex.

Once computed all these parameters, it has been possible to compute the force exchanged by seals and housing:

$$F_H = m_s a_{rad} \bullet \cos(\varphi) - F_{sp} \bullet \cos(\varphi) - F_{py} \bullet \sin(\varphi) + F_{px} \bullet \cos(\varphi) \quad (3.25)$$

In which m_s is the mass of the Apex and F_{sp} is the preload of the Apex spring.

Since the force exchanged between the two parts is function of the engine speed, it is recomputed for each value from 3000 to 8000 rpm. At each rotational regime a different equilibrium position, between apex seal and housing, is found considering also the properties of the oil interposed.

The computed values of F_h are used in the Stribeck's formula to compute the $COF_{lubricant}$, as follows [44]:

$$COF_{lubricant} = \mu = a_4 D^4 + a_3 D^3 + a_2 D^2 + a_1 D + a_0 \quad (3.26)$$

$$Duty\ parameter = D = \frac{\eta \bullet v}{\frac{F_h}{L}} \quad (3.27)$$

In which η is the oil kinematic viscosity, v is the speed of the Apex, F_h is the normal force, L is the characteristic length of the seal and the parameters a_0, a_1, a_2, a_3, a_4 derive from figure 19.

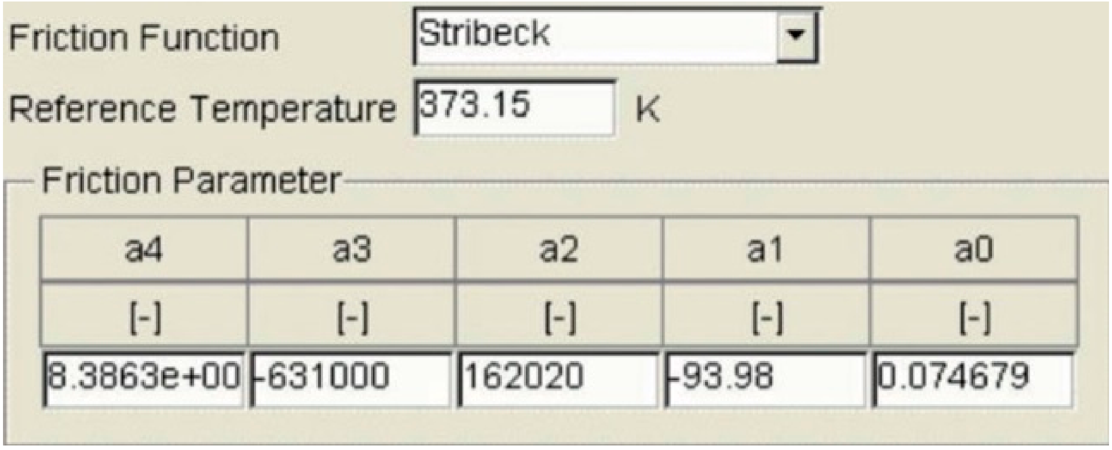


Figure 19: Experimental measure of Stribeck's coefficients [44]

When metal-to-metal contact occurs, the COF_{metal} is considered as constant and function of the materials of Apex and housing.
What has been obtained up to this point is the coefficient of friction in each point of the contact path, between seals and epitrochoid housing, for each rotational regime and according to a mixed lubrication regime.
The obtained composite coefficients of friction can be used to compute the friction forces, the friction torques and the friction power losses.

3.4 Power losses computation

One of the aims of the lubrication is to reduce the friction between two surfaces in relative motion. For this reason, several elements inside a Wankel rotary engine need to be lubricated:

- Housing-shaft supports
- Rotor-shaft support
- Apex seals

For each of these parts, a suitable mathematical model is developed, in order to understand which are the main aspects that influence the power losses. These computations have a direct dependency on the chemico-physical characteristics of the lubricant oil and on the geometrical parameters of the lubricated elements.
According to the applications and to the number of rotors, Wankel rotary engines can be endowed with rolling bearings or simple bushings. These solutions would have

different requirements in term of oil delivering, oil consumption and, consequently, power losses. For sake of completeness, two different models have been developed.

For what concerns the power losses due to rubbing friction caused by the apex seals, the model makes reference to chapter 3.3.2.

3.4.1 Supports

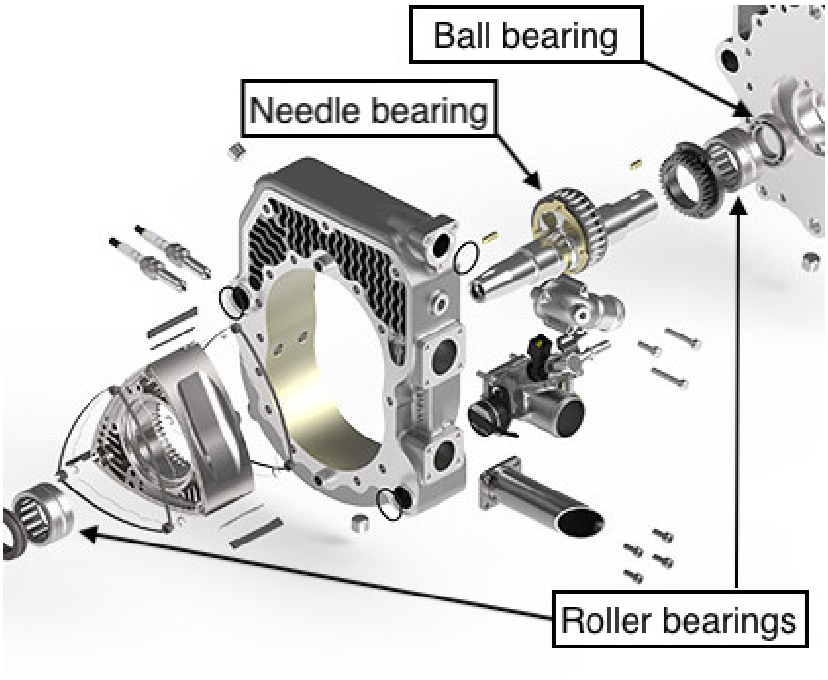


Figure 20: Detail view of the supports [68]

In the case in which the supports are rolling bearings, the formulas to calculate the friction power losses are provided by SKF [45].

$$Total\ friction\ torque = M = M_{rr} + M_{sl} + M_{seal} + M_{drag}\ [Nmm] \tag{3.28}$$

$$Rolling\ friction\ torque = M_{rr} = \phi_{ftag} \bullet \phi_{rs} \bullet G_{rr} \bullet (\upsilon \bullet \nu)^{0,6}\ [Nmm] \tag{3.29}$$

In which ϕ_{ftag} is the reduction factor due to heating for cutting phenomena, ϕ_{rs} is the kinematic corrective factor for filling, G_{rr} is a variable function of kind of bearing, mean diameter of the bearing and load, n is the rotational speed and υ is the oil kinematic viscosity at 100°C.

$$Sliding\ friction\ torque = M_{sl} = G_{sl} \bullet (\phi_{bl} \bullet \mu_{bl} + (1 - \phi_{bl}) \bullet \mu_{EHL})\ [Nmm] \tag{3.30}$$

In which G_{sl} is a variable function of kind of bearing, mean diameter of the bearing and load, ϕ_{bl} is a sliding friction factor, μ_{bl} is a constant function of the movement and μ_{bl} is a constant dependent on the kind of bearing lubrication.

$$Seals\ friction\ torque = M_{seals} = K_{s1} \bullet d_s^{\beta} + K_{s2}\ [Nmm] \tag{3.31}$$

In which K_{s1} is a constant function of the kind and of the dimension of the bearing, d_s is the diameter of the seals, β is a constant function of the kind of bearing and K_{s2} is a constant function of the kind and of the dimension of the bearing.

$$Drag\ torque = M_{drag_{ball}} =$$

$$4 \bullet V_m \bullet K_{ball} \bullet d_m^5 \bullet n^2 + 1{,}093 \bullet 10^{-7} \bullet n^2 \bullet d_m^3 \bullet \left(\frac{n \bullet d_m^2 \bullet f_t}{\nu}\right)^{-1{,}379} \bullet R_s\ [Nmm] \tag{3.32}$$

$$Drag\ torque = M_{drag_{roll}} =$$

$$4 \bullet V_m \bullet K_{roll} \bullet C_w \bullet B \bullet d_m^4 \bullet n^2 + 1{,}093 \bullet 10^{-7} \bullet n^2 \bullet d_m^3 \bullet \left(\frac{n \bullet d_m^2 \bullet f_t}{\nu}\right)^{-1{,}379} \bullet R_s\ [Nmm] \tag{3.33}$$

In which V_m is a sliding loss actor, B is the width of the bearing, d_m is the mean diameter of the bearing, K_{ball} and K_{roll} are two constants dependent on the rolling body, C_w is a constant function of the bearing dimension, R_s is function of the bearing dimensions, n is the rotational speed, υ is the oil kinematic viscosity at 100°C and f_t is a function the lubrication regime.
To obtain the power loss, the total resistant torque has to be multiplied by the rotational speed.

$$Power\ loss = M \bullet \omega\ [W] \tag{3.34}$$

In the case in which, instead, the supports are not rolling bearings but bushings, it is possible to compute the power losses making reference to Petroff, Sommerfeld and Ocvirk analytical models. Sommerfeld and Ocvirk models make possible to compute the oil pressure distribution between journal and bearing [46]. This distribution is function of the eccentricity, rotational speed, load and chemico-physical properties of the oil. Petroff equation makes possible to compute the power loss as function of the rotational speed, oil properties, dimensions of the bushings and height of the oil film [47].

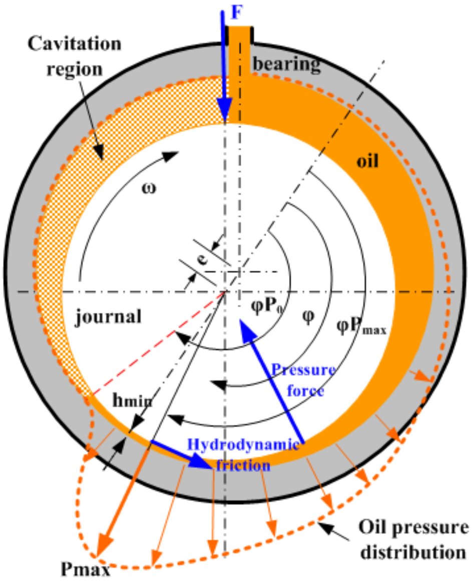


Figure 21: Bushing oil pressure distribution [47]

$$\textit{Pressure distribution for infinite long bearing (Sommerfeld)}$$

$$P_S = \frac{\eta \bullet U \bullet r}{C_r^2} \bullet \left(\frac{6 \bullet \varepsilon \bullet (2 + \varepsilon \bullet \cos(\phi)) \bullet \sin(\phi)}{(2 + \varepsilon^2) \bullet (1 + \varepsilon \bullet \cos(\phi))^2} \right) + p_0 \; [BAR] \tag{3.35}$$

$$\textit{Pressure distribution for infinite short bearing (Ocvirk)}$$

$$P_O = \frac{\eta \bullet U}{r \bullet C_r^2} \bullet \left(\frac{B^2}{4} - z^2 \right) \bullet \frac{3 \bullet \varepsilon \bullet \sin(\phi)}{(1 + \varepsilon \bullet \cos(\phi))^3} + p_0 \; [BAR] \tag{3.36}$$

In which η is the oil dynamic viscosity at 100°C, U is linear velocity of journal, Cr is the radial clearance, r is the bearing radius, Dj is the journal diameter, ε is the eccentricity ratio (ε = e/Cr), e is the absolute bearing eccentricity, B is the bearing length, z is the longitudinal direction coordinate and p_0 is the cavitation pressure,

The hydrodynamic friction that develops in the region of the oil in which the pressure is higher can be computed by means of Petroff equation 3.37.

$$Losses = \frac{2 \bullet \pi \bullet \eta \bullet \omega^2 \bullet L \bullet R^3}{h_{min}} \; [W] \tag{3.37}$$

In which η is the lubricant dynamic viscosity, ω is the engine's angular speed, L is the bearing width, R is the bearing radius and h_{min} is the minimum oil film thickness. It may be observed from the Petroff equation that friction power loss varies linearly with lubricant viscosity and square of angular speed.

3.4.2 Apex rubbing friction

According to the results obtained in chapter 3.3.2, it is possible to compute the losses due to Apex rubbing fiction. The composite coefficients of friction, in lubricated regions ($COF_{lubricant}$) and in metal-to-metal contacts (COF_{metal}), are now used to compute friction forces, friction torques and friction power losses.

$$Friction\ force = F_f = F_h \bullet COF\ [N] \tag{3.38}$$

$$Friction\ torque = T_f = F_f \bullet R \bullet \cos(\varphi)\ [Nm] \tag{3.39}$$

$$Friction\ power\ losses = P_f = T_f \bullet rotational\ speed\ [W] \tag{3.40}$$

The values obtained in formulas 3.38, 3.39 and 3.40 must be intended as mean values for each rotational speed, from 3000 to 8000 rpm.

3.4.3 Total losses

In order to obtain the total power losses for each engine speed, it is enough to sum all the contributions given by supports and apex seals.

In case of usage of bushing, the total power loss for supports is:

$$Bushing\ power\ loss =$$
$$Bush_{PL} = 2 \bullet shaft\ bushings + 1 \bullet rotor\ bushing\ [W] \tag{3.41}$$

In case of usage of rolling bearings, the total power loss for supports is:

$$Bearing\ power\ loss =$$
$$Bear_{PL} = 2 \bullet roller\ bearing + 1 \bullet needle\ bearing + 1 \bullet ball\ bearing\ [W] \tag{3.42}$$

For what concerns the apex seals connected losses, they can be calculated as:

$$Apex\ rubbing\ friction\ power\ loss\ = P_A = 3 \bullet P_f\ [W] \tag{3.43}$$

Since the total power loss due to rubbing friction is the sum of the power losses generated by each Apex seal, in formula 3.43 appears a multiplication by 3.

The sum of each contribution gives the total power loss for each engine speed:

$$Total\ power\ loss\ bushings = Bush_{PL} + P_A\ [W] \tag{3.44}$$

$$Total\ power\ loss\ bearings = Bear_{PL} + P_A\ [W] \tag{3.45}$$

3.5 Archard's model for Apex seals wear

The Archard's equation is a simple analytical model, used to describe the abrasive wear. It is based on the asperities contact theory, which means that the load exchanged by the bodies is not distributed on the whole area but just on the peaks of the rough surface.

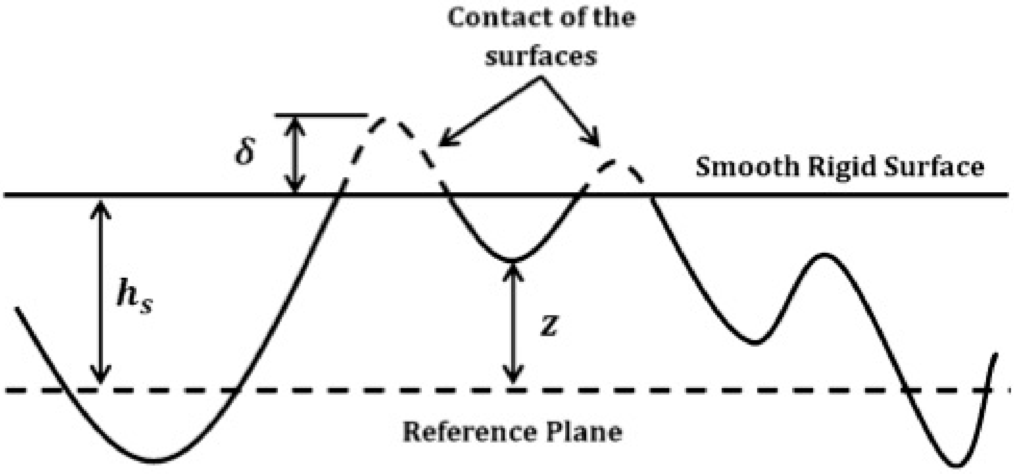


Figure 22: Schematic representation of the asperities contact

This load sharing, combined with the relative motion of the parts, make the harder material to remove those peaks from the softer material. This condition is called "two-body abrasive wear".

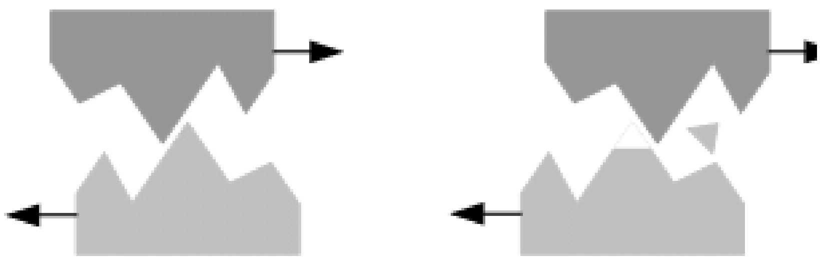


Figure 23: Two-body abrasive wear

The abrasion of the softer material makes it to loose part of its volume at any time a metal-to-metal contact occurs. Basically this is what happens during the relative sliding between apex seals and housing. In the study of the Apex wear, lubrication plays a fundamental role. An oil film thick enough can strongly reduce the amount of asperity-contacts, reducing together with the coefficient of friction the amount of material worn.

$$Worn\ volume\ per\ shaft\ rotation = V_{worn} = K \bullet \frac{L \bullet F}{H}\ [m^3] \quad (3.46)$$

Where L is the sliding distance per shaft rotation, F is the exchanged normal force, H is the hardness of the softer material and K is the Archard's wear coefficient [48]. To understand which could be the duration of an apex seal in these conditions, it is possible to use formula 3.47:

$$Seal\ duration = \frac{Sealing\ volume}{Worn\ volume\ per\ shaft\ rotation}\ [Hours\ or\ km] \quad (3.47)$$

In this thesis project, the parameters have been derived as follows:

- L, sliding distance: characteristic dimension of the housing derived from CAD
- F, exchanged normal force: obtained by the computations done in chapter 3.3.2
- H, hardness of the softer material: found in literature [49]
- Sealing volume: characteristic dimension of the Apex derived from CAD
- K, Archard's wear coefficient: determined a posteriori using a study case on a different engine (Mazda RX8, 13B Wankel rotary engine)

The Archard's wear coefficient is generally determined experimentally by means of bench tests. Since it is strongly influenced by the lubrication conditions, it is worth to understand how it could vary passing from a boundary to a hydrodynamic lubrication regime.

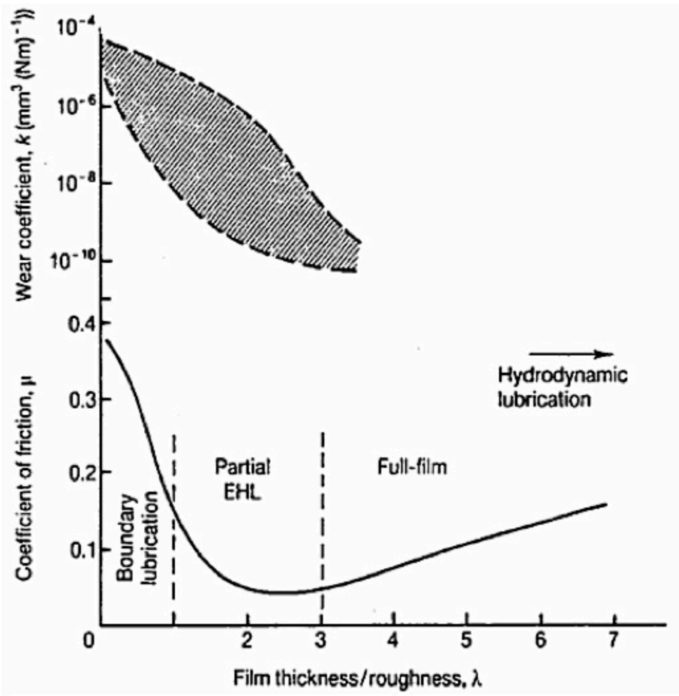


Figure 24: Lubrication regime VS Wear coefficient [50]

3.5.1 Wankel rotary engine 13B simulation

The determination of the coefficient K for this project is done studying a known case. Applying the same procedure and hypothesis described in chapters 3.1, 3.2, 3.3 to a Mazda RX8 13B Wankel engine, it is possible to simulate two driving cycles in order to understand what could be the apex seals lifespan in real life conditions.

The simulation begins with the acquisition of the 13B engine's operative map and of the RX8's gear ratios [52]. Then, starting from the NEDC and WLTC speed maps, it is possible to compute the gear shifting profile and the relative engine speed.

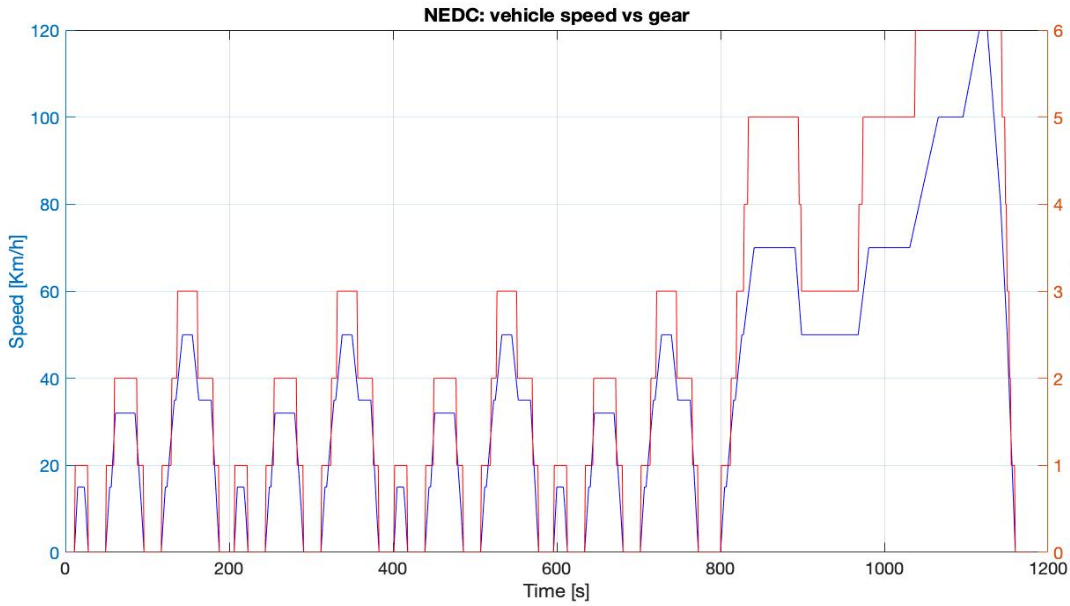


Figure 25: NEDC gear shifting profile

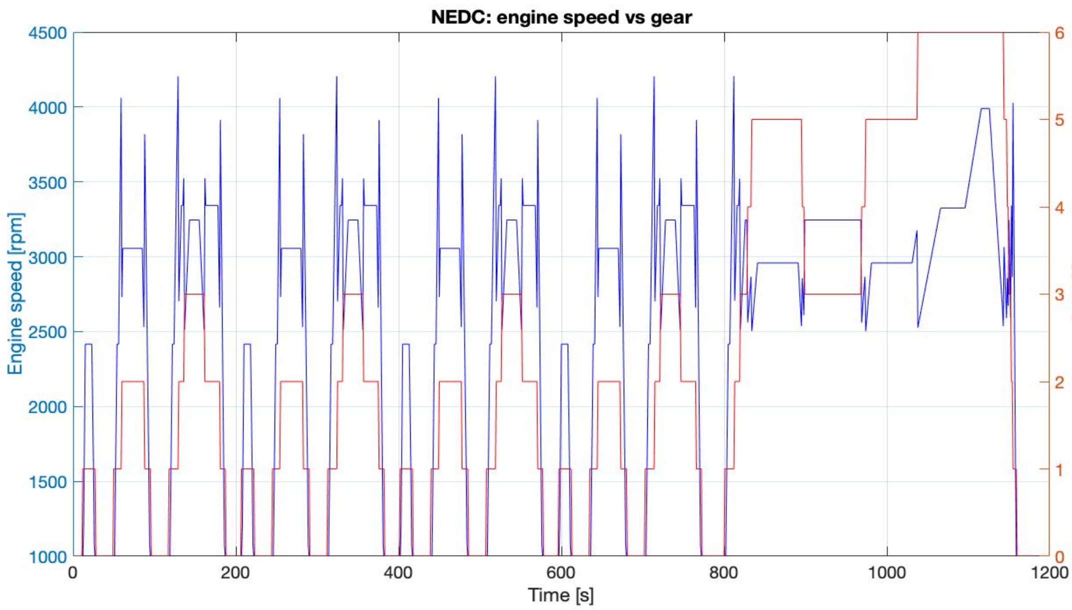


Figure 26: NEDC engine speed profile

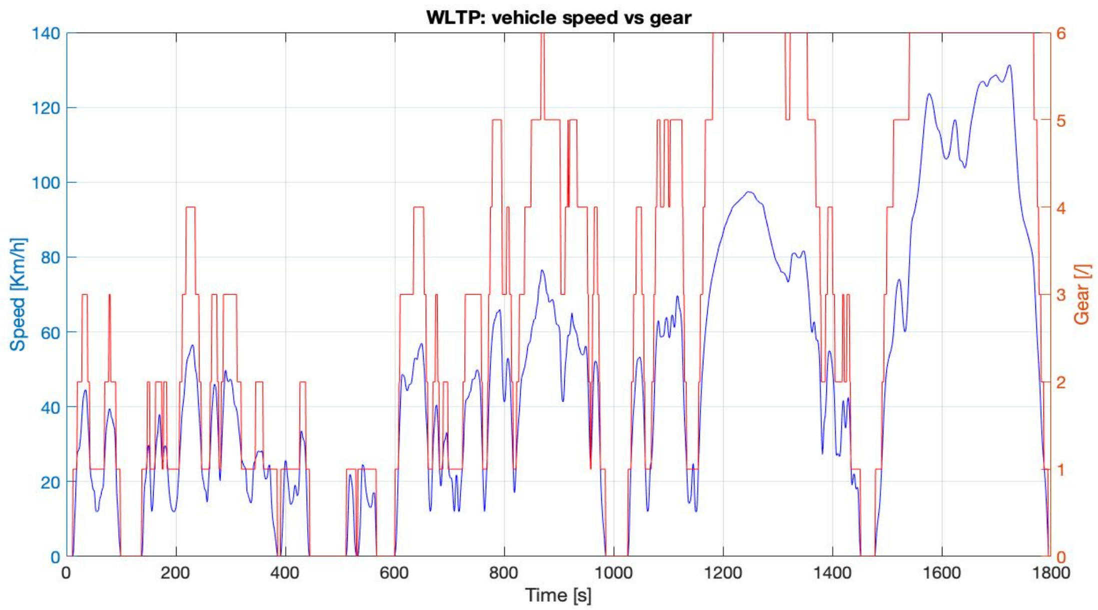


Figure 27: WLTP gear shifting profile

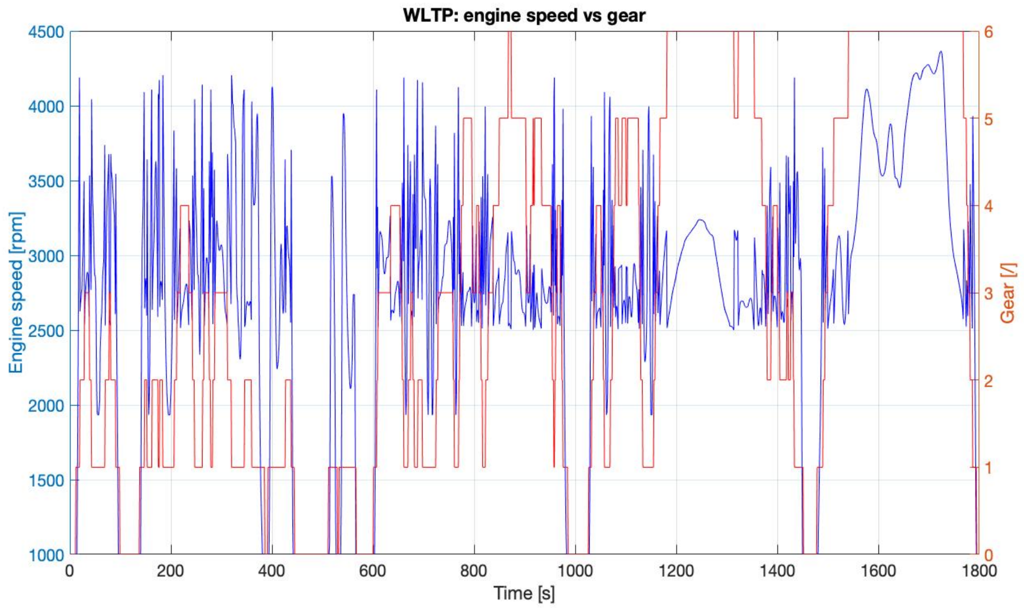
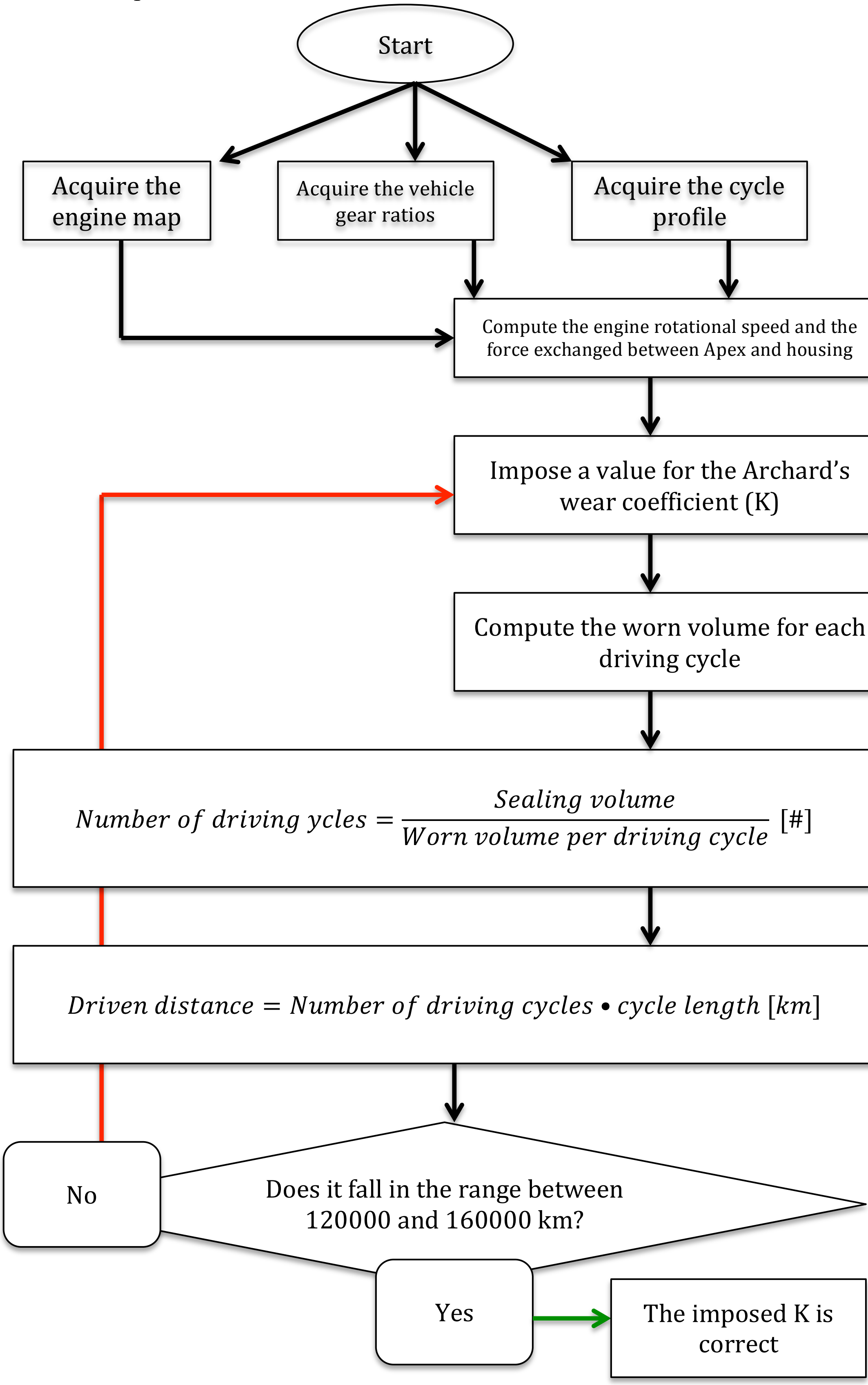


Figure 28: WLTP engine speed profile

For each rotational speed, it is now possible to compute the normal forces exchanged by apex seals and housing, as described in chapter 3.3.2. These forces are then used in the Archard's wear equation to compute the amount of worn volume per driving cycle.

In order to obtain the wanted wear coefficient (K) a hypothesis must be done. Imposing an apex lifespan in a range between 120.000 and 160.000 km [51] it is possible to compute, by means of an iterative process, the coefficient that brings as a result such apex duration.



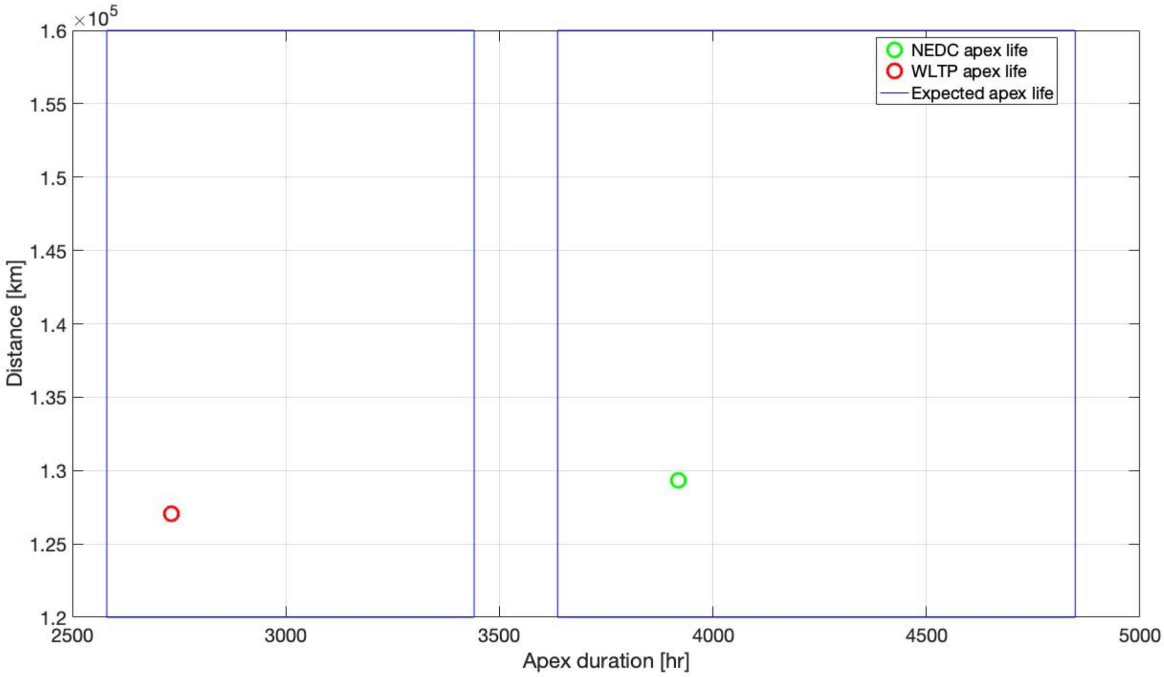


Figure 29: Apex seals life for NEDC and WLTC

The results obtained by this simulation derive from an imposition of the Archard's wear coefficient equal to $10^{-7.9}$.

The output of this model is used as input in the computation of the apex seals life for the Rotron RT300, applied as range extender.

3.6 Range extender simulation

This simulation is used to understand if a small displacement Wankel rotary engine can be a valuable alternative to a common piston engine, applied as range extender in hybrid electric vehicles.

To understand what could be the performance of a Wankel engine in this application, it is combined with a 42.2kWh battery pack [53] and together are tested through NEDC and WLTC. The simulation starts computing, per each vehicle speed prescribed by the test cycle, the power demand according to rolling resistance and aerodynamic resistance.

$$Rolling\ resistance = R_r = m \bullet g \bullet k_r \bullet speed\ [W] \tag{3.48}$$

$$Aerodynamic\ resistance = R_a = \frac{1}{2} \bullet c_x \bullet \rho \bullet S \bullet speed^3 [W] \tag{3.49}$$

$$Total\ resistance = Power\ demand = R_a + R_r\ [W] \tag{3.50}$$

Where m is the mass of the vehicle, k_r is the rolling coefficient, c_x is the aerodynamic drag coefficient, ρ is the density of the air and S is the resistant surface of the vehicle.

The power supply of REHEV works as follows:

- If the battery charge is above SOC_{min}, the range extender is OFF and the power is supplied by the battery pack
- If the battery charge is below SOC_{min} and the power demand is below P_{ON}, the range extender is OFF and the power is supplied by the battery pack
- If the battery charge is below SOC_{min} and the power demand is between P_{ON} and P_{ECO}, the range extender is ON and works in ECO mode. The power produced is sent directly to the electric motor and just the surplus (produced-demand) is used to recharge the batteries
- If the battery charge is below SOC_{min} and the power demand is between P_{ECO} and P_{BOOST}, the range extender is ON and works in BOOST mode. The power produced is sent directly to the electric motor and just the surplus (produced-demand) is used to recharge the batteries
- If the battery charge is below SOC_{min} and the power demand is above P_{BOOST}, the range extender is ON and works together with the electric motor supplied by the battery pack to meet the power demand.

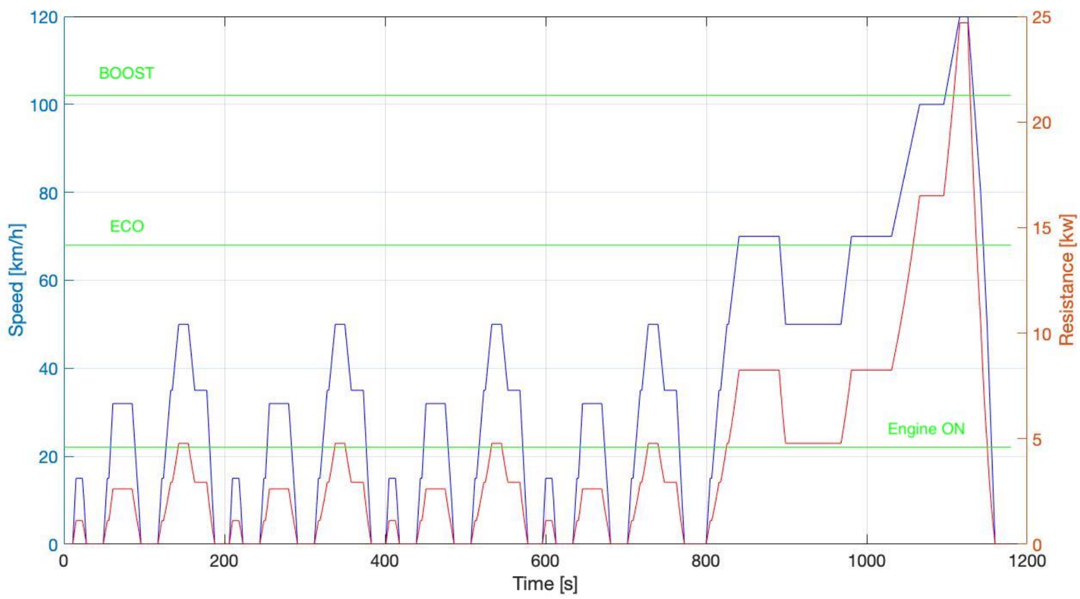


Figure 30: RE simulation: NEDC

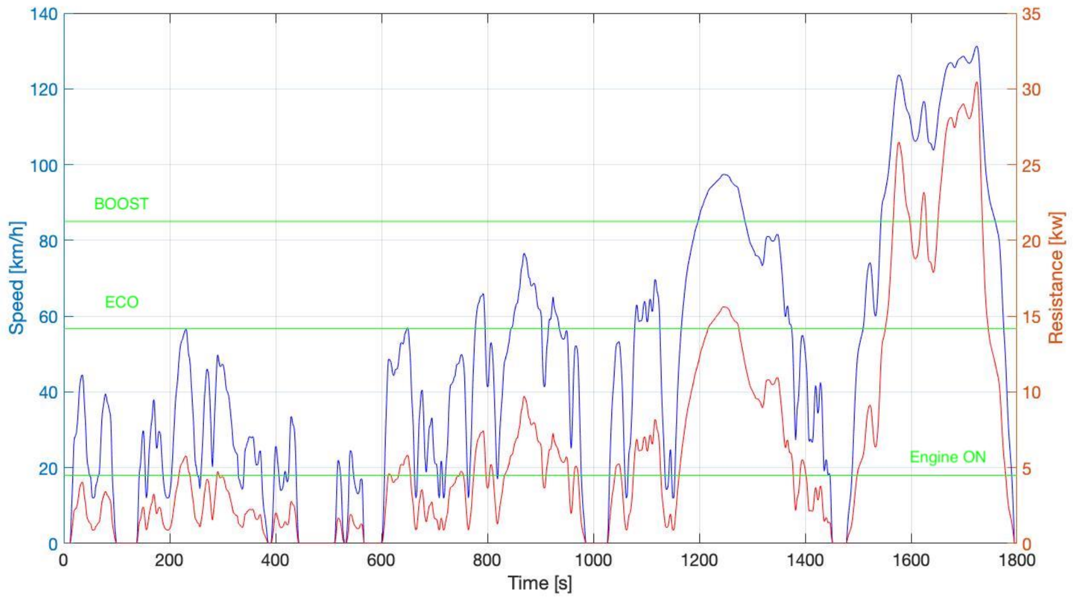


Figure 31: RE simulation: WLTC

Since the Wankel for range extender applications works at two fixed points, it results easy to compute the fuel consumption for a driving cycle. Imposing a hypothetic fuel tank dimension, it is possible to compute the actual range extension.

$$Fuel\ consumption\ per\ driving\ cycle =$$
$$= \frac{FC(5000) \bullet time(ECO)}{BP(5000)} + \frac{FC(7000) \bullet time(BOOST)}{BP(7000)} \quad \left[\frac{g}{cycle}\right] \tag{3.50}$$

In which FC is the brake specific fuel consumption at a given engine speed (5000 or 7000), BP is the brake power at a given engine speed (5000 or 7000), and time(ECO) and time(BOOST) are the times spent respectively in ECO mode and in BOOST mode.

Using the results obtained in chapter 4.5, it is also possible to evaluate the apex seals life in these operative conditions.

$$Worn\ volume\ per\ driving\ cycle = WV$$
$$\frac{5000}{60} \bullet time(ECO) \bullet wv(5000) + \frac{7000}{60} \bullet time(BOOST) \bullet wv(7000) \left[\frac{m^3}{cycle}\right] \tag{3.51}$$
$$Apex\ life = \frac{Sealing\ volume}{WV} \bullet cycle\ length\ [km] \tag{3.52}$$

In which WV is the worn volume at 5000 or 7000 rpm, and time(ECO) and time(BOOST) are the times spent respectively in ECO mode and in BOOST mode.
For sake of completeness, the obtained results are paragoned with the data declared by BMW for the i3 [55]: the simulation is conduced with the same vehicle and powertrain parameters.

4. Results and discussion

The methods described in previous chapters are now applied to a Rotron RT300 Wankel rotary engine, in order to perform a complete numerical investigation.

4.1 Engine performance and combustion cycle

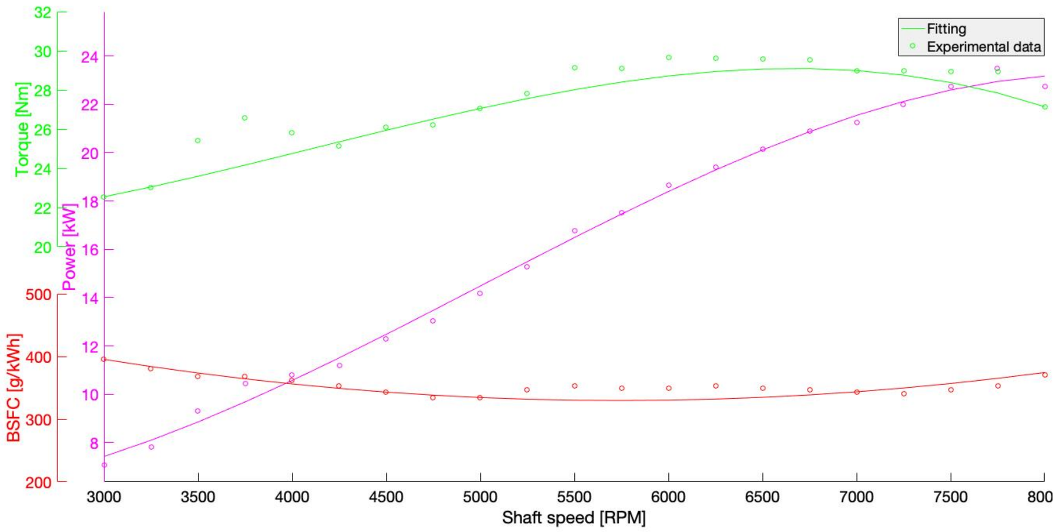


Figure 32: Rotron RT300 performance map

In this study, the fitted curves of power, torque and BSFC are used to investigate the combustion cycle. Applying the inverse formulas of thermodynamic laws, the brake mean effective pressure (BMEP) per each engine speed is obtained.

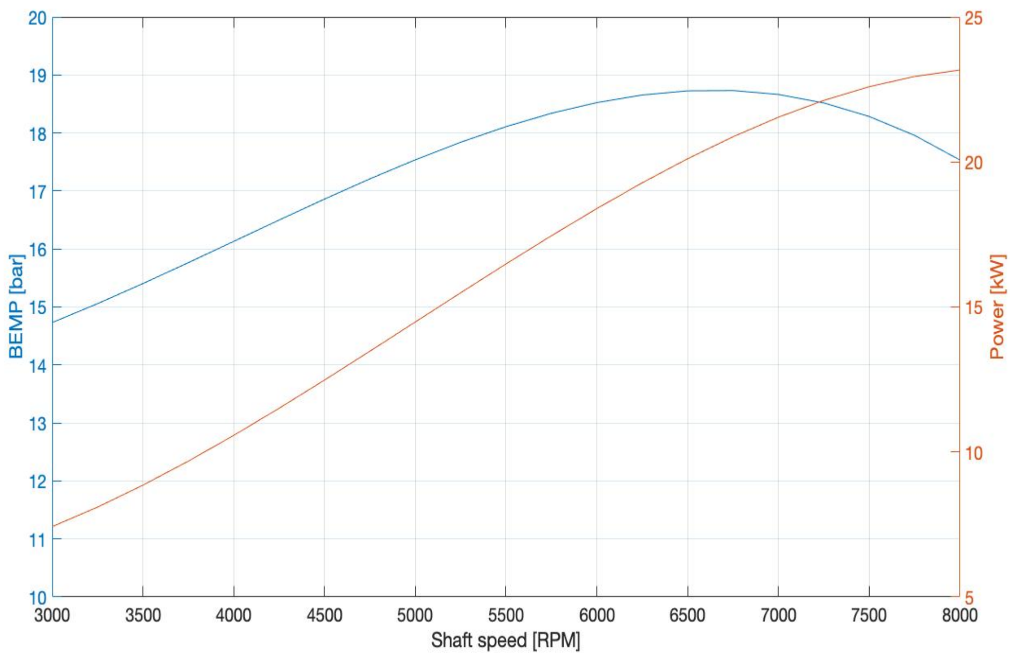


Figure 33: BEMP vs Engine speed: full load curve

BMEP is computed as a function of the power produced at a given engine speed, in particular:

$$BMEP \propto \frac{Power}{Engine\ speed} \tag{4.1}$$

As it is possible to see in figure 33, the brake mean effective pressure tends to follow an increasing trend with the power, up to a point in which, due to several aspects relative to inefficiencies at high rotational speed, the maximum load decreases.

The values of BMEP are used to reconstruct the ideal combustion cycle, followed by each Wankel chamber. The analysis starts from the computation of the peak firing pressure (PFP), imposing some thermodynamic limits, such as the intake pressure, the exponent of the polytropic compression curve and the exponent of the polytropic expansion curve.

Since the Wankel is operated as a naturally aspirated engine, the simulation starts from a value of the intake pressure equal to the environmental one, so 1 bar. The upper limit of the pressure diagram is the PFP, which is function of the engine speed. The values of the two polytropic exponents, in compression and expansion, are taken from literature [37] and respectively equal to 1,2 and 1,3.

For what concerns the limits of the volume diagram, the Rotron RT300 has a total displacement of 300 cm^3, with a volume at the end of the compression stroke equal to 102 cm^3 and a volume at the end of the expansion stroke of 402 cm^3.

This leads to the definition of the ideal combustion cycle, per each engine speed.

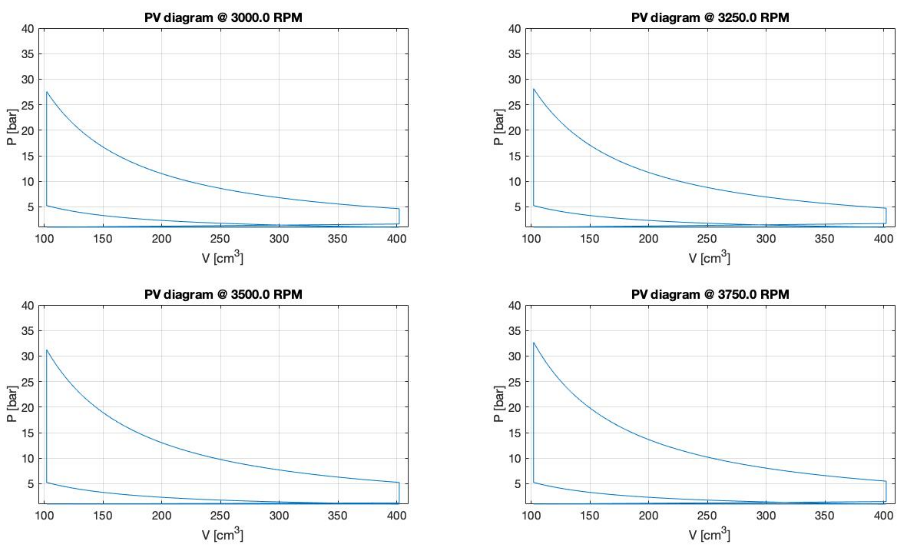


Figure 34: PV diagram: from 3000 to 3750 rpm

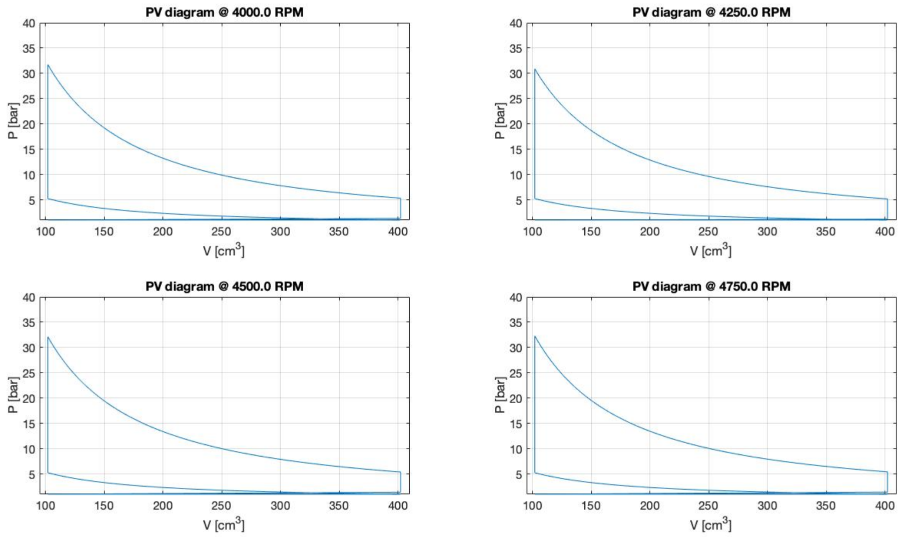


Figure 35: PV diagram: from 4000 to 4750 rpm

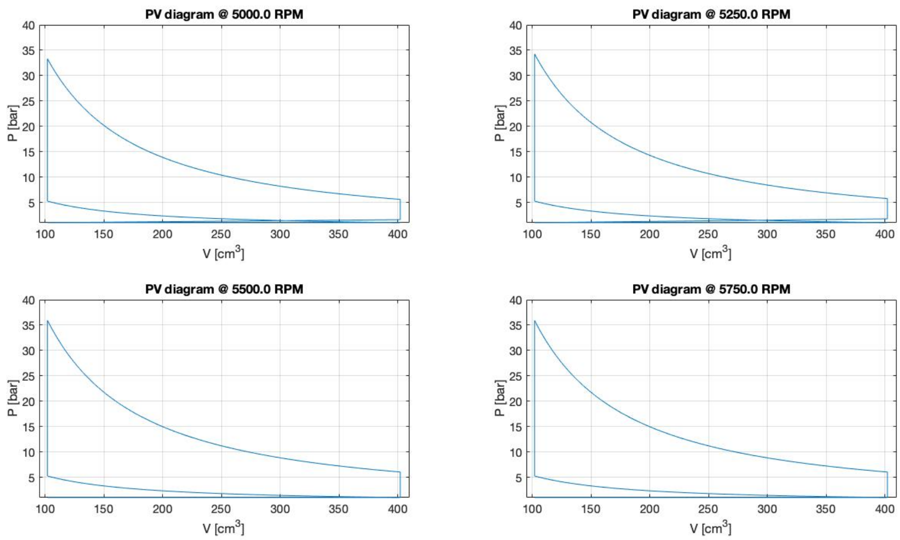


Figure 36: PV diagram: from 5000 to 5750 rpm

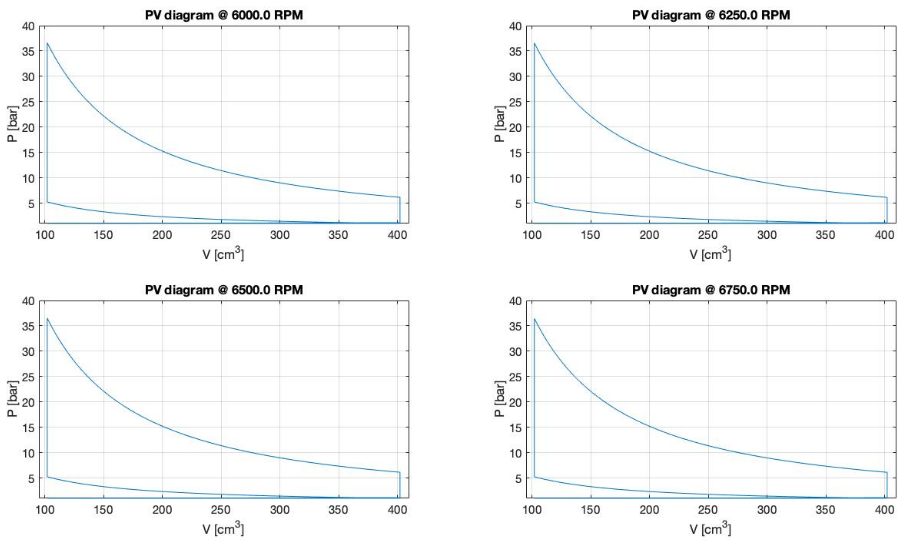


Figure 37: PV diagram: from 6000 to 6750 rpm

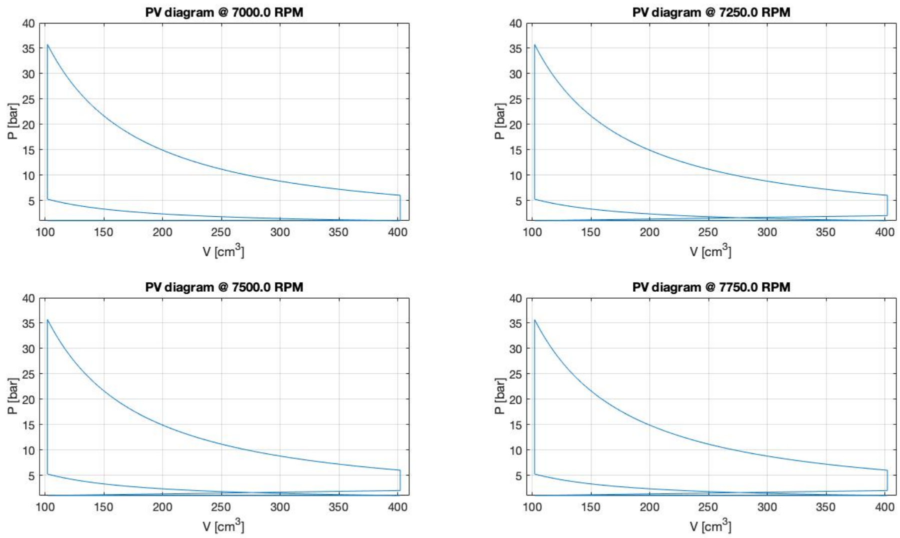


Figure 38: PV diagram: from 7000 to 7750 rpm

The definition of the pressure trend, per each rotational speed, is fundamental to compute the normal force exchanged by apex seals and housing. Moreover, the pressure per each chamber is translated into a force acting on the rotor: these three forces lead to a time varying load acting on bearings used as supports.
According to the results obtained, it is possible to define the contact path between seals and housing, so the coefficients of friction, the Apex wear, the power losses due to rubbing friction and the power losses inside the bearings.
In the following these steps are repeated for the three lubricant oils chosen and for the different supports to be compared: rolling bearings and bushings.

4.2 Lubricant oil influence

The core of the project is to understand how the physico-chemical characteristics of the lubricant oil influence the performance of the engine. For this reason, three different lube oil have been chosen:

Table 5: Lubricant oil characteristics

Property	OIL		
Grade	**SAE 10W-40**	**SAE 0W-30**	**Motul Gp 2T**
Density @ 15.6 C, g/ml, ASTM D4052	0.87	0.841	0.916
Flash Point, Cleveland Open Cup, °C, ASTM D92	230	226	256
Kinematic Viscosity @ 100 C, mm2/s, ASTM D445	13.2	11.5	16.9
Pour Point, °C, ASTM D97	-42	-42	-33

In table 5 are reported the main aspects that influence the material wear and the power losses. In particular, what is to be expected is:

- The oil with the lowest viscosity (SAE 0W30) will have the lowest power losses
- The oil with the highest density (Motul Gp 2T) will ensure the lowest Apex wear
- The oil with the lowest volatility and the highest viscosity (Motul Gp 2T) will ensure the lowest oil consumption

These considerations can be explained as follows.

For what concerns the connection between viscosity and power loss, it is clearly visible in the formulas used to compute the losses:

$$Rolling\ friction\ power\ in\ bearings = P_{rr} = \phi_{ftag} \bullet \phi_{rs} \bullet G_{rr} \bullet (\upsilon \bullet \nu)^{0,6} \bullet \omega\ [W] \tag{4.2}$$

$$Power\ losses\ in\ bushings = \frac{2 \bullet \pi \bullet \eta \bullet \omega^2 \bullet L \bullet R^3}{h_{min}}\ [W] \tag{4.3}$$

$$Apex\ rubbing\ friction = P_f = F_h \bullet COF \bullet R \bullet \cos(\varphi) \bullet \omega\ [W] \tag{4.4}$$

In which

$$COF = \mu = a_4\left(\frac{\eta \bullet v}{\frac{F_h}{L}}\right)^4 + a_3\left(\frac{\eta \bullet v}{\frac{F_h}{L}}\right)^3 + a_2\left(\frac{\eta \bullet v}{\frac{F_h}{L}}\right)^2 + a_1\frac{\eta \bullet v}{\frac{F_h}{L}} + a_0 \tag{4.5}$$

So, as it is possible to see, all the formulas used to compute the friction losses have a direct dependency on the oil viscosity. The higher is the oil viscosity the higher will be the power loss.

About the material's wear, the main oil characteristic that influences the Apex abrasion is the density. According to the Archard's model, the amount of worn volume is directly proportional to the force exchanged between the elements. The force exchanged between seals and housing is just function of the spring preload, of the rotational speed and of the in-chamber pressure, so it is independent on the oil chars. What matters is how the force is distributed on the rough surface of the softer material: the oil, which is interposed between Apex and housing, makes distribute the exchanged force and, moreover, the higher the oil density, the lower its compressibility and the lower the metal-to-metal contacts.

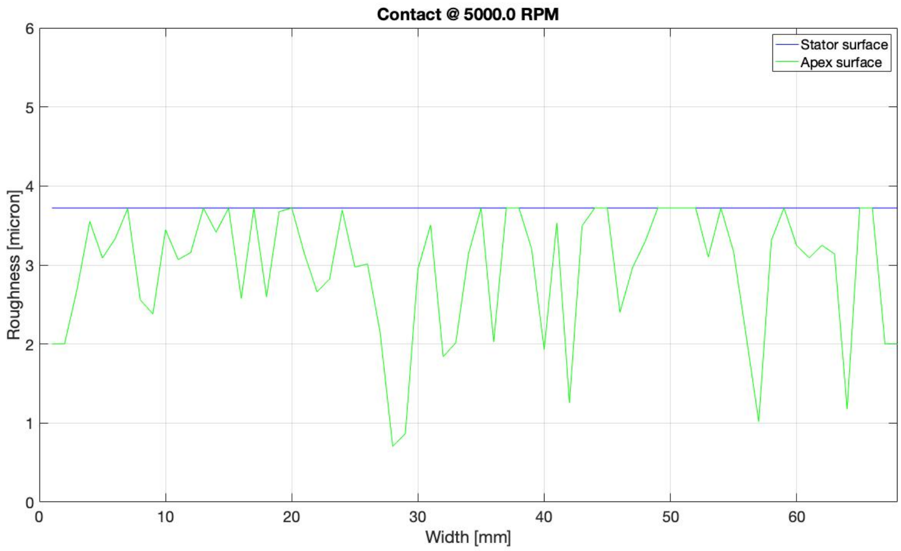


Figure 39: Contact path @ 5000 RPM: SAE 0W30

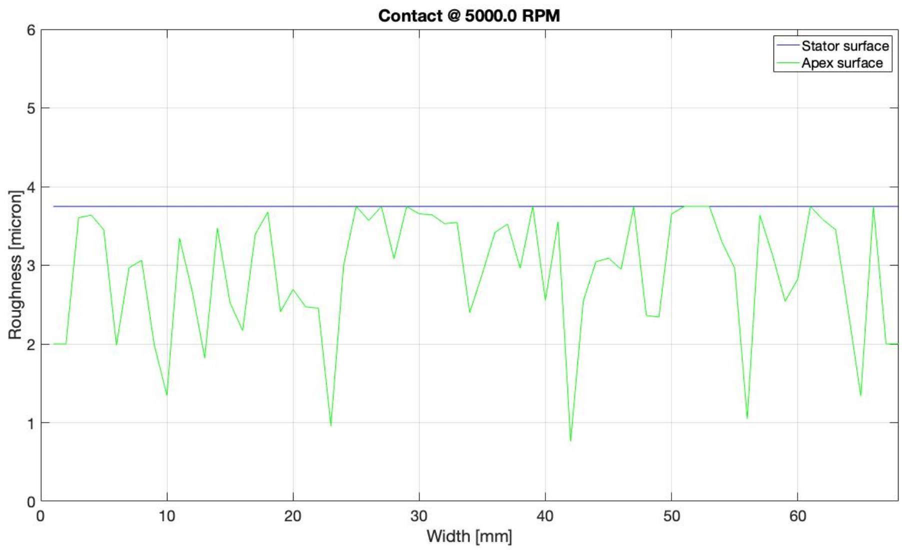


Figure 40: Contact path @ 5000 RPM: SAE 10W40

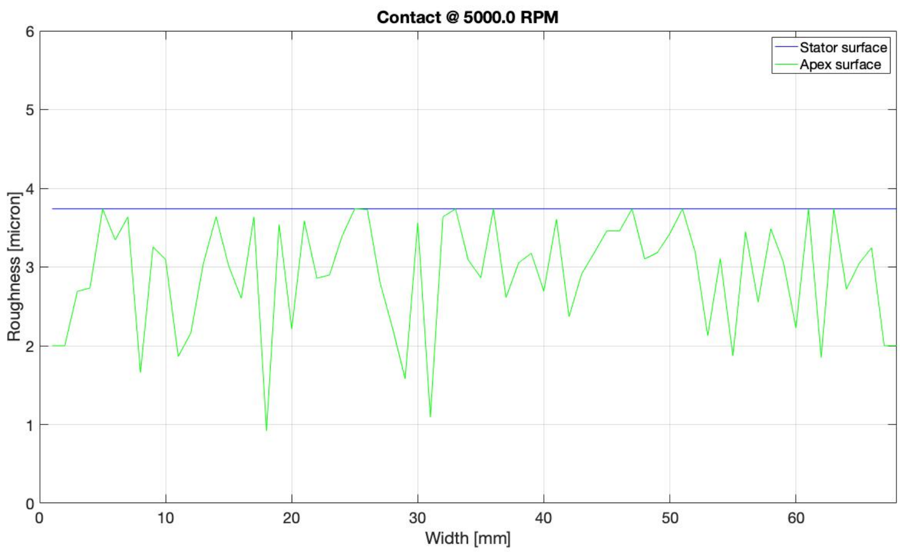


Figure 41: Contact path @ 5000 RPM: Motul GP 2T

As it is possible to see in figures 39, 40 and 41, at a given rotational speed so for a given force, the number of metal-to-metal contacts tends to decrease as the oil density increases. According to this and to the Archard's model, the Apex's wear is expected to be the lowest when the Motul GP 2T oil is used.

Finally, regarding the oil consumption, the main properties that influence this aspect are the oil volatility and the viscosity. A SAE experimental research predicts for every centistoke (cSt) less of oil viscosity, a 3% increment in oil consumption and for every 1% more of oil volatility a 1% more of oil consumption [56].

4.3 Supports power loss

In the following will be discussed the results obtained using the Petroff's equation for bushings and using the analytical model provided by SKF for rolling bearings. Since rolling bearings are designed to stand rotating bodies with the lowest friction possible, the expected result is a total power loss higher for bushings and lower rolling bearings.

4.3.1 Rolling bearings power loss

The rolling bearings used in the Rotron RT300 are two roller bearings, one ball bearing and one needle bearing, applied respectively as shaft supports and as rotor support.

Figure 42: Roller bearing NU 2205 ETPV 2 C3

Figure 43: Ball bearing 6205 ETP C3

Figure 44: Needles bearing K 50x55x30

The dimensions in table 6 characterize the three bearings:

Table 6: Rolling bearings geometrical characteristics

Bearing	Inner diameter [mm]	Outer diameter [mm]	Width [mm]
NU 2205 ETPV 2 C3	25	52	18
6205 ETP C3	25	52	15
K 50x55x30	50	55	30

The bearings geometrical characteristics, combined with the physico-chemical ones of the three lubricant oils, generate different power losses:

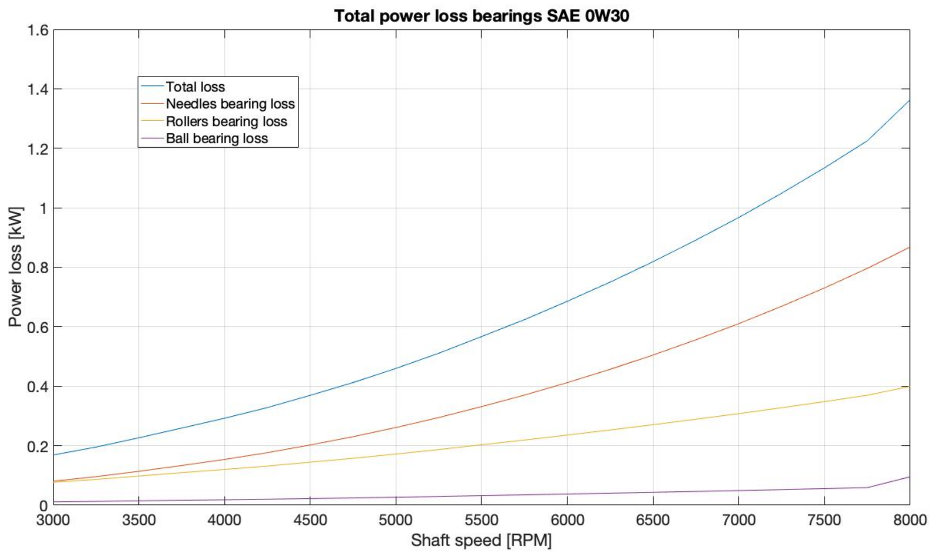


Figure 45: Rolling bearings power loss x SAE 0W30

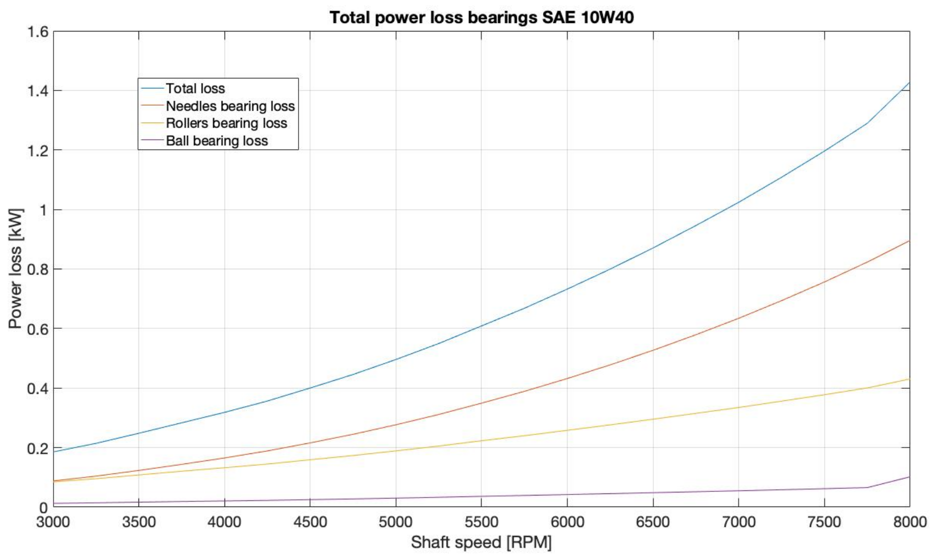


Figure 46: Rolling bearings power loss x SAE 10W40

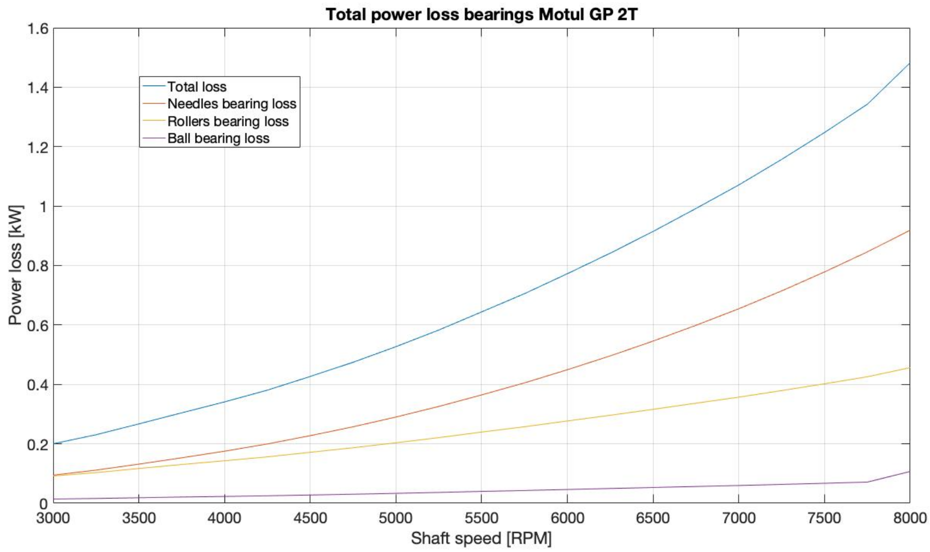


Figure 47: Rolling bearings power loss x Motul GP 2T

As expected, the combination of the same rolling bearings with the three different oils gives different results in term of power loss. In particular the combination with the oil at higher viscosity (Motul GP 2T) gives the highest power loss, at any given rotational speed. On the contrary, the one with lower viscosity (SAE 0W30) gives the lowest power loss.

From the point of view of the efficiency to be obtained, the use of SAE 0W30 as lubricant oil for the supports results to be the most convenient.

4.3.2 Bushings power loss

The rolling bearings used in the Rotron RT300 can be substituted with bushings. This requires different oil delivery methods, structural modifications and implies diverse oil consumption and power losses. In order to understand if their application is more or less convenient with respect to the one of rolling bearings, they are compared from a power loss point of view, using the Petroff equation.

The engine can be endowed with two support bushings and one rotor bushing.

Table 7: Bushing geometrical characteristics

Bushing	Nominal diameter [mm]	Width [mm]
Support	25	32
Rotor	50	30

Figure 48: Support bronze bushing 25x32 **Figure 49**: Rotor bronze bushing 50x30

As happened for rolling bearings, the geometrical characteristics of the bushings, combined with the physico-chemical ones of the three lubricant oils, generate different power losses:

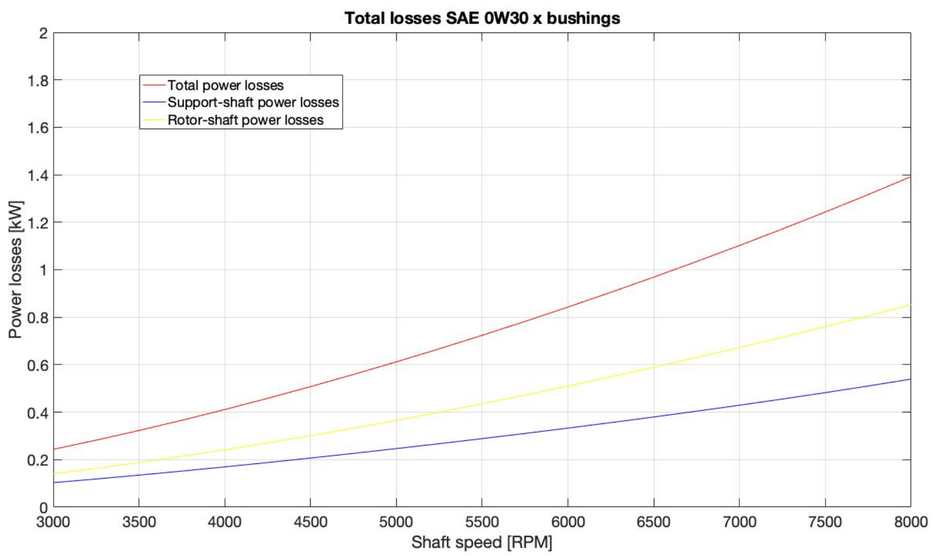


Figure 50: Bronze bushings power loss x SAE 0W30

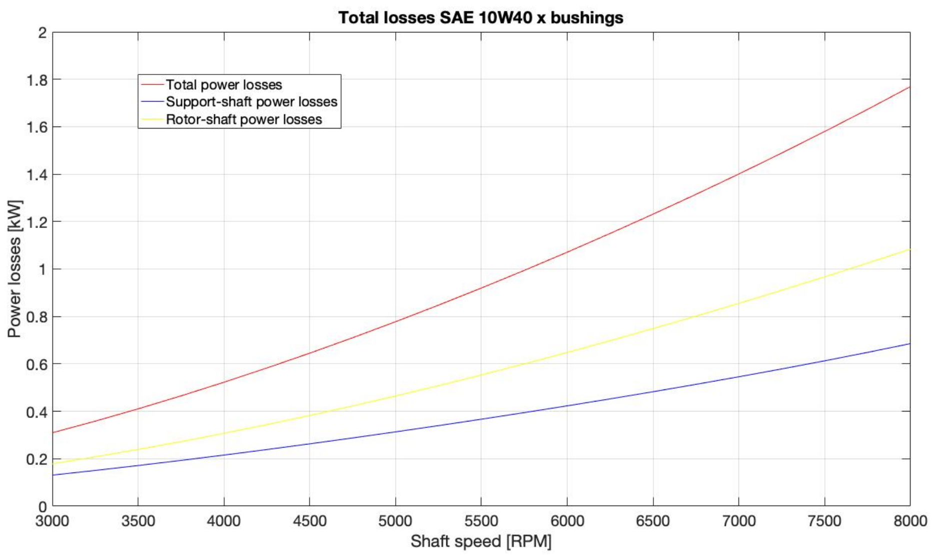


Figure 51: Bronze bushings power loss x SAE 10W40

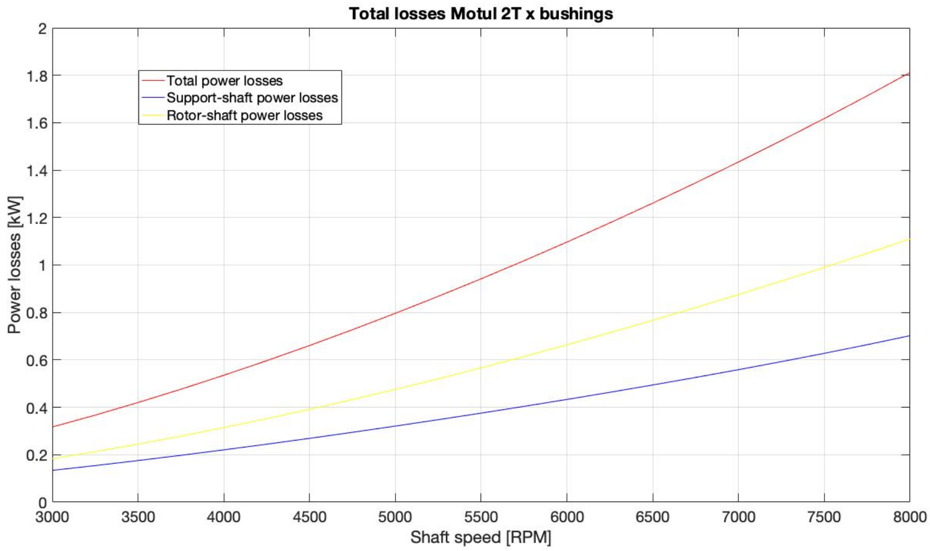


Figure 52: Bronze bushings power loss x Motul GP 2T

As expected, the power losses due to hydrodynamic friction inside bushings follow an almost parabolic trend with the rotational speed. Moreover, they tend to increase linearly with the oil viscosity: the higher the viscosity, the higher the power losses.

$$Losses \propto \frac{\eta \bullet \omega^2}{h_{min}} [W] \tag{4.6}$$

Although the minimum oil film thickness (h_{min}) tends to increase as the rotational speed increases, being the contribute of the speed itself at the power of 2, the increasing quantity of oil between journal and bushing has a lower impact on loss reduction.

As in the previous case the use of SAE 0W30 as lubricant oil for the supports results to be the most convenient, from the point of view of the efficiency to be obtained.

In the following will be analysed the pro and cons of both the support solutions, bronze bushings versus rolling bearings.

As it is possible to see in figures 45, 46 and 47, rolling bearing groups have a lower total power loss for any engine speed and for any lubricant oil applied. The small efficiency variance would have a greater impact on fuel economy and, consequently, on pollutant formation.

The lower friction of rolling bearings has a drawback the net increment of weight, due to the presence of several metal and rubber parts. The total mass of the bushing-

support group is equal to 0,236 kg while the one of the bearing-support group is 0,6 kg [57] [58] [59] [60].
The rigid structure of rolling bearings makes them less prone to deformations, with respect to bronze bushings. In particular, the application as rotor support makes the needles bearing to suffer a high failure risk. The main problem is connected to the fact that, being the rotor exposed to three chambers, it is subjected to a great thermal gradient and consequently to a significant thermal deformation. This is translated into a distortion of the rotor body and of the support as well, which makes it working in a non-design condition. Bronze bushings better compensate this risky deformation, while the cage of rolling bearings risks to brake suddenly. For this reason, rolling bearings need more maintenance, increasing costs and reducing the intervals between interventions.

4.4 Apex rubbing friction

In order to understand which is the contribution of the apex seals rubbing friction in term of power loss, a numerical analysis is now performed.
The sliding contact between seals and housing generates two kind of friction. The first one is due to metal-to-metal contacts between the asperities of the two rough surfaces, while the second one is the hydrodynamic friction due to the presence of lubricant oil.
The procedure starts from the generation of the rough surfaces, of both Apex and housing, imposing two average roughness levels:

- Apex roughness: 2 μm [40]
- Housing internal surface roughness: 1 μm [41]

Table 8: Apex and housing geometrical data

Apex seals	
Base [mm]	3
Width [mm]	66
Housing	
Base [mm]	485
Width [mm]	68

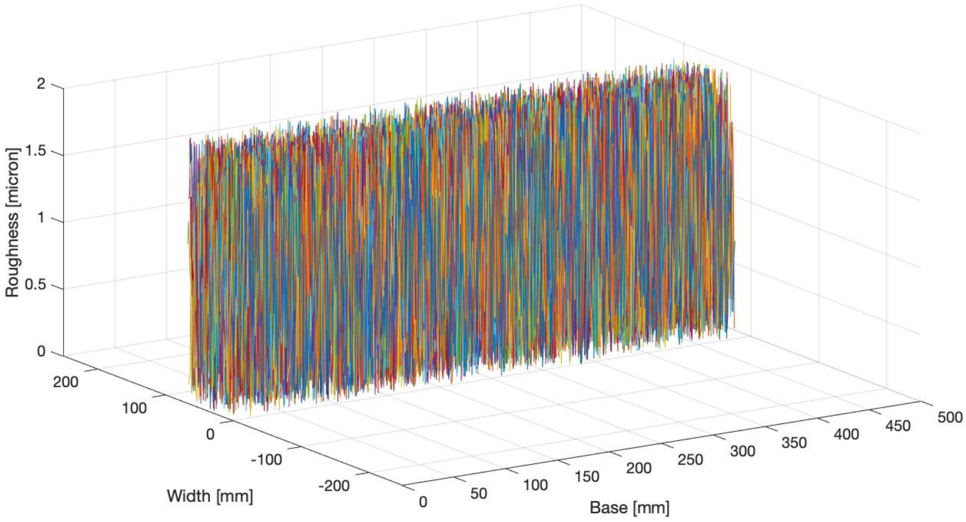


Figure 53: Housing rough surface

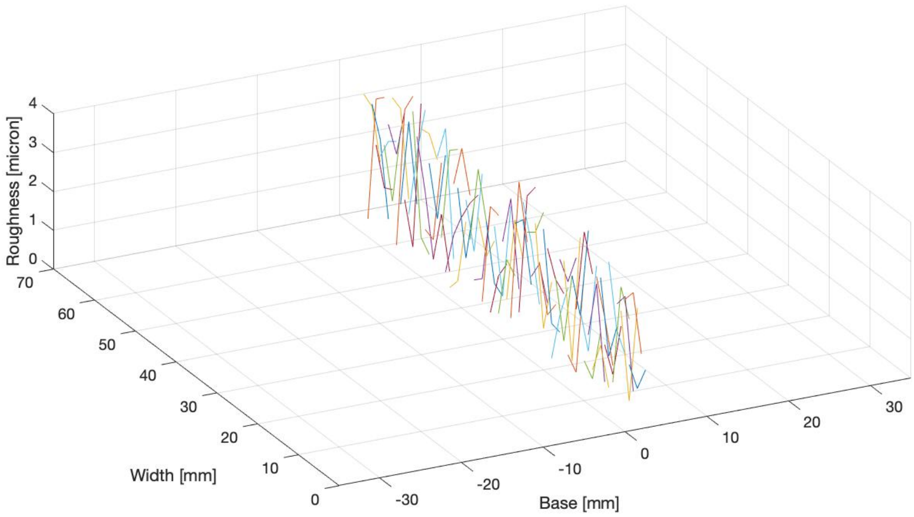


Figure 54: Apex seals rough surface

Once obtained the equivalent rough surfaces, it is possible to put the two profiles into contact, simulating the interaction between the upper surface of the Apex and the lower one of the housing. At this point a film of oil thick enough to generate a boundary lubrication regime is interposed.

$$Composite\ roughness = \sigma = \sqrt{R_A^2 + R_S^2}\ [\mu m] \tag{4.7}$$

$$OFT = h = \sigma * 1.5 = 3{,}35\ [\mu m] \tag{4.8}$$

The oil film thickness varies according to the to the force exchanged between Apex and housing and according to the compressibility of the oil itself. In particular, the chemico-physical characteristic that governs this phenomenon is the bulk elastic property of the material. It represents how much the oil will compress under the application of a given external pressure [62]: it is function of the density, temperature and pressure of the environment in which the oil is applied. So, to understand how the oil will compress and how the contact path will vary according to the force exchanged, per each rotational speed the normal force between seals and housing is computed. It is strongly influenced by the differential pressure between two adjacent chambers, the preload of the Apex spring, the mass of the seals and by the engine speed. Imposing a spring preload of 25 N, an Apex's mass of 9 g, and an engine speed from 3000 to 8000 rpm, the force is computed as in formula 4.9:

$$F_H = m_s a_{rad} \bullet \cos(\varphi) - F_{sp} \bullet \cos(\varphi) - F_{py} \bullet \sin(\varphi) + F_{px} \bullet \cos(\varphi) \tag{4.9}$$

Furthermore, the normal force (F_H) tends to deform the materials of apex and housing. Knowing the geometrical properties of the two and the relative material elastic modulus, it is possible to understand how the parts tend to change shape, changing the geometry of the contact path as well.

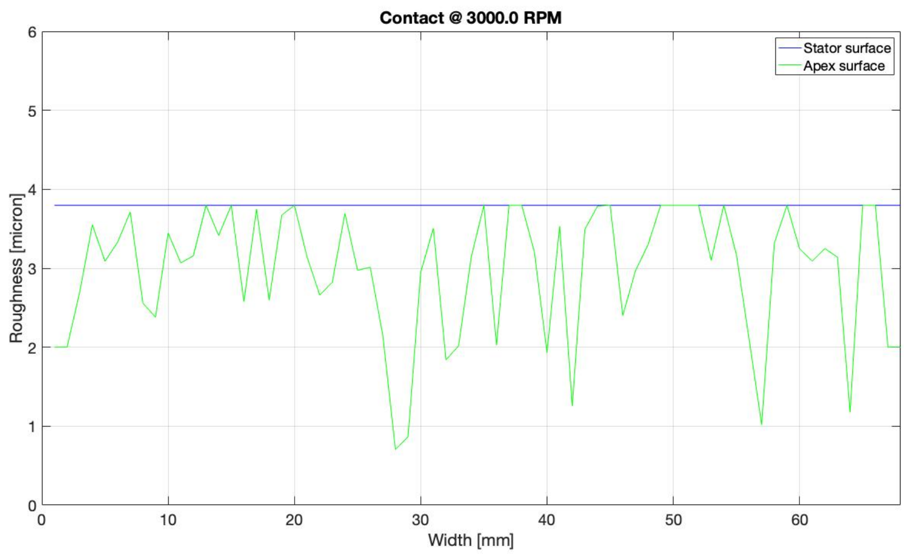


Figure 55: Contact path @ 3000 rpm

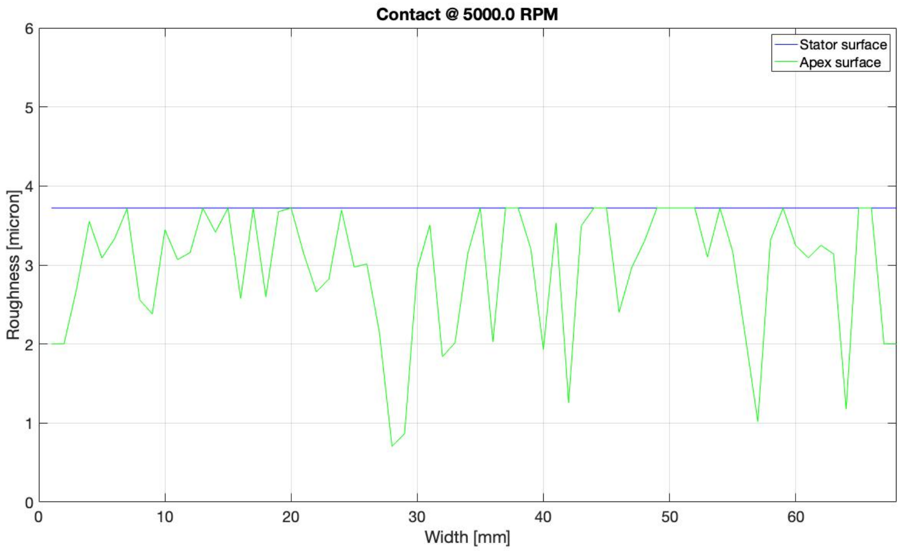


Figure 56: Contact path @ 5000 rpm

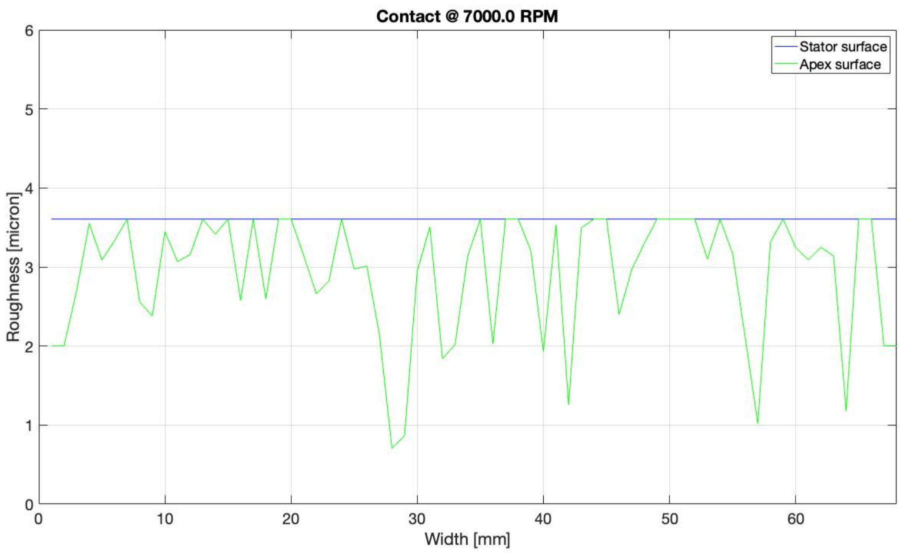


Figure 57: Contact path @ 7000 rpm

For sake of simplicity, the housing surface is approximated with a rigid un-deformable surface so the Apex's surface is the only that deforms.

As it is possible to see, some contact points not present at 3000 rpm are now present at 7000 rpm and, passing through 5000 rpm, some interaction points tend to enlarge due to the material deformation and to the oil compression.

The behaviour of oil and metal makes vary the computed coefficient of friction: having the COF a direct dependency on the oil viscosity and on the exchanged force, for each of the three lube oil and for each rotational speed, it is computed.

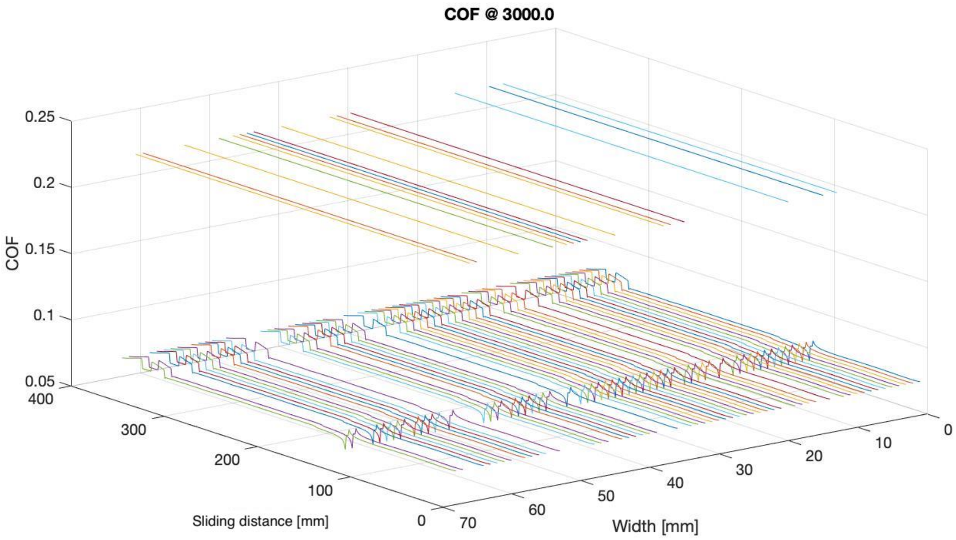


Figure 58: Coefficient of friction for Motul GP 2T @ 3000 rpm

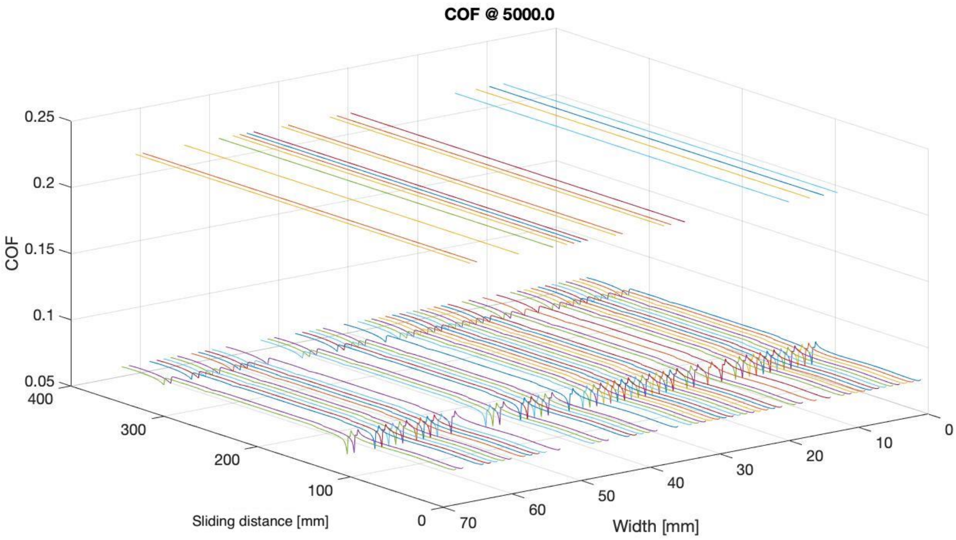


Figure 59: Coefficient of friction for Motul GP 2T @ 5000 rpm

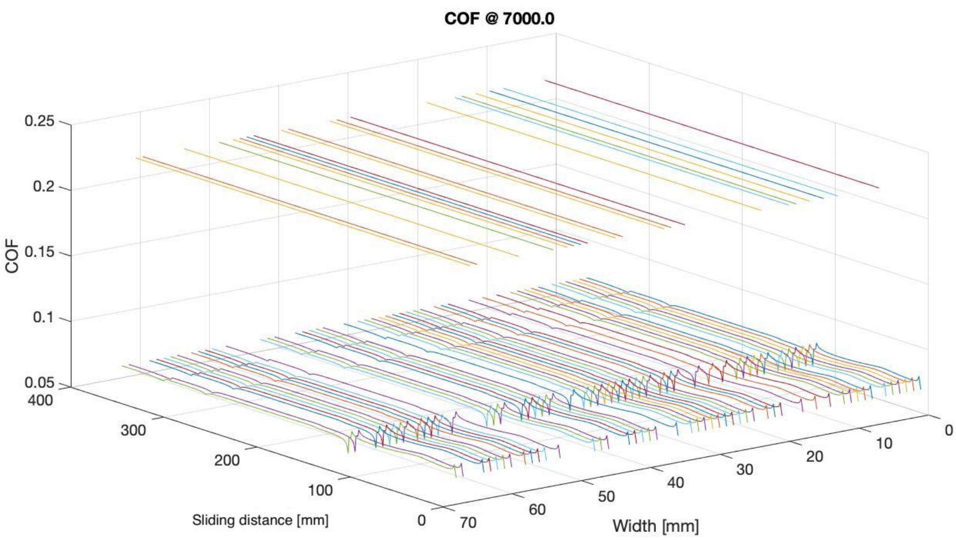


Figure 60: Coefficient of friction for Motul GP 2T @ 7000 rpm

Figures 58, 59 and 60 demonstrate that increasing the rotational speed, the normal force increases and the oil tends to compress more: this leads to an increment of the metal-to-metal contacts, so to an higher amount of point with a COF equal to 0,23 [63]. The same procedure, repeated for oils SAE 0W30, SAE 10W40 and Motul GP 2T leads to different results in term of coefficient of friction, as function of the engine speed:

Table 9: COF from 3000 to 5500 rpm

RPM	3000	3250	3500	3750	4000	4250	4500	4750	5000	5250	5500
Motul GP 2T	0.1048	0.1090	0.1111	0.1104	0.1103	0.1151	0.1152	0.1153	0.1176	0.1177	0.1200
SAE 0W30	0.1002	0.1000	0.0997	0.0989	0.0990	0.0991	0.1016	0.1016	0.1017	0.1063	0.1087
SAE 10W40	0.1071	0.1090	0.1111	0.1126	0.1127	0.1151	0.1198	0.1199	0.1200	0.1200	0.1223

Table 10: COF from 5750 to 8000 rpm

RPM	5750	6000	6250	6500	6750	7000	7250	7500	7750	8000
Motul GP 2T	0.1201	0.1224	0.1247	0.1247	0.1247	0.1247	0.1247	0.1248	0.1248	0.1294
SAE 0W30	0.1110	0.1110	0.1179	0.1202	0.1225	0.1225	0.1248	0.1248	0.1271	0.1272
SAE 10W40	0.1247	0.1247	0.1270	0.1316	0.1339	0.1362	0.1362	0.1362	0.1385	0.1454

The coefficients of friction are calculated as function of the engine speed per each point of the contact path between apex and housing.The values are then averaged in order to have just the dependency on the rotational regime, because the apex power loss will be computed just as function of the angular speed.

From tables 9 and 10, it is possible to deduce that as the engine speed increases, the friction coefficient increases as well and this is a symptom of an increasing number of asperities contacts. Moreover it is possible to see that the higher losses are obtained applying oil SAE 10W40 as lubricant: this is justified by the fact that, making reference to table 5, it has a high viscosity which tends to increase the hydrodynamic friction and it has a low value of density which is not able to impair metal-to-metal contacts.

According to this, what is expected is a higher power loss for oil SAE 10W40 at any regime, then oil Motul GP 2T and lastly oil SAE 0W30.

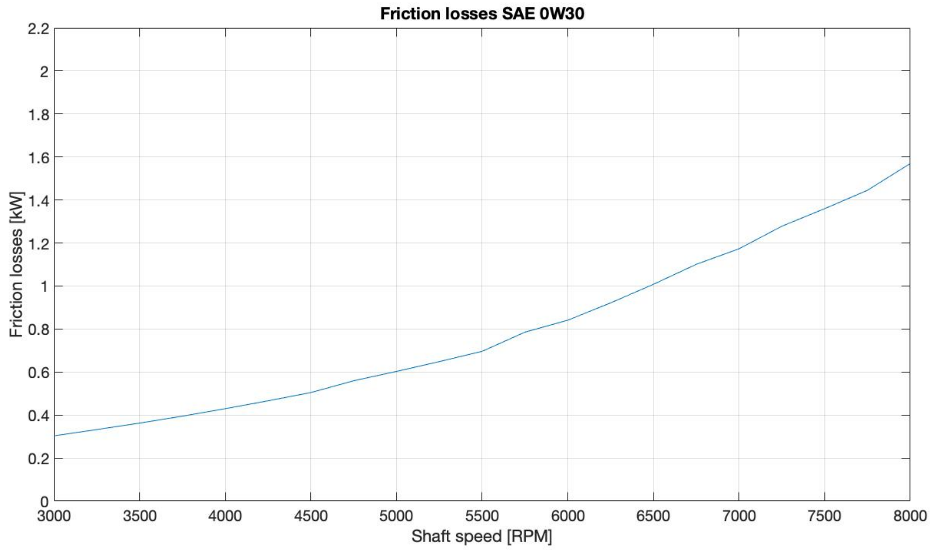


Figure 61: Apex rubbing friction power loss x SAE 0W30

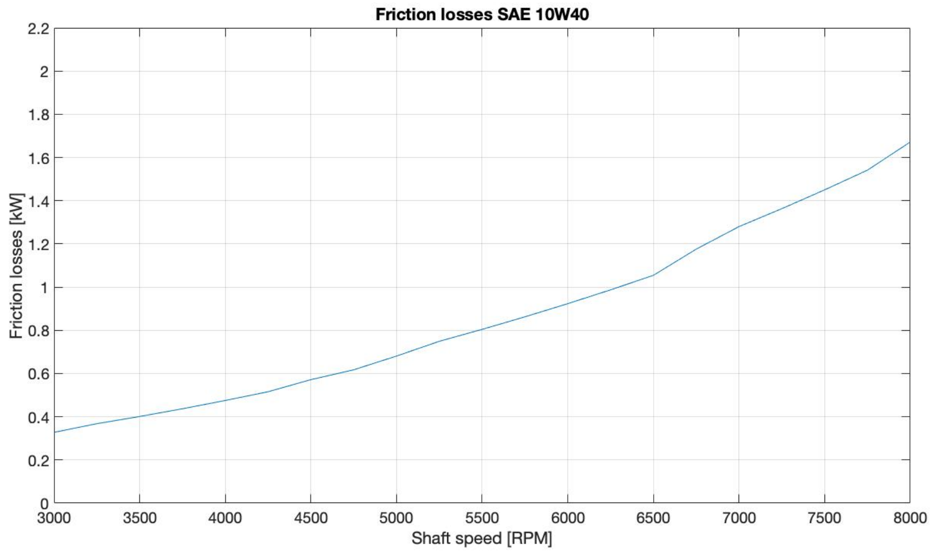


Figure 62: Apex rubbing friction power loss x SAE 10W40

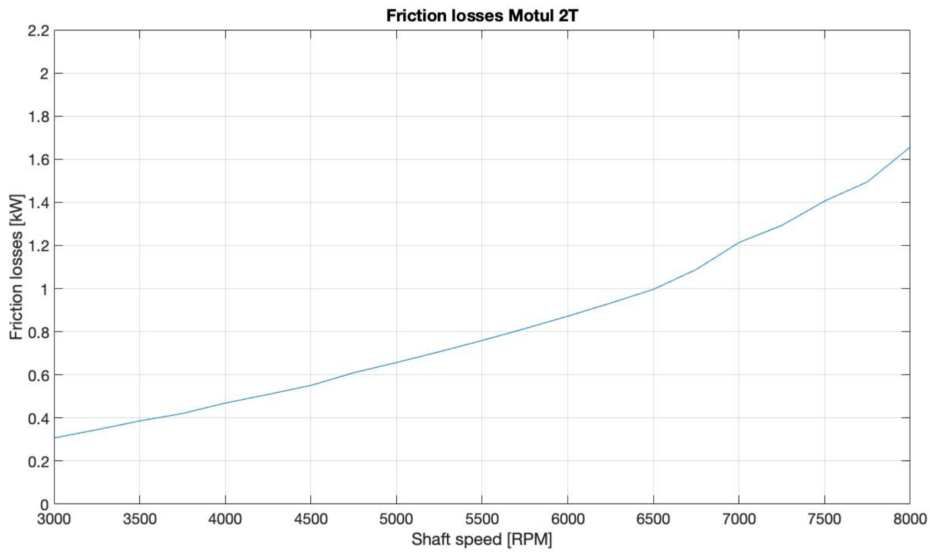


Figure 63: Apex rubbing friction power loss x Motul GP 2T

Figures 61, 62 and 63 demonstrate that oil SAE 10W40 induces a higher power loss with respect to the other two lubricants, although its viscosity is not the highest and its density is not the lowest. This sort of trade-off in term of chemico-physical properties seems to be not so profitable from an efficiency point of view.
Lube oil SAE 0W30, also in this case, appears as the most efficient in term of power loss due to rubbing friction

4.5 Total power loss

Once computed the power losses due to friction inside bushings, bearings and due to the apex-housing contact, they can be summed together to find out which is the total power loss inside the Rotron RT300. According to the supports used and according to the lubricant oil used, a different result appears:

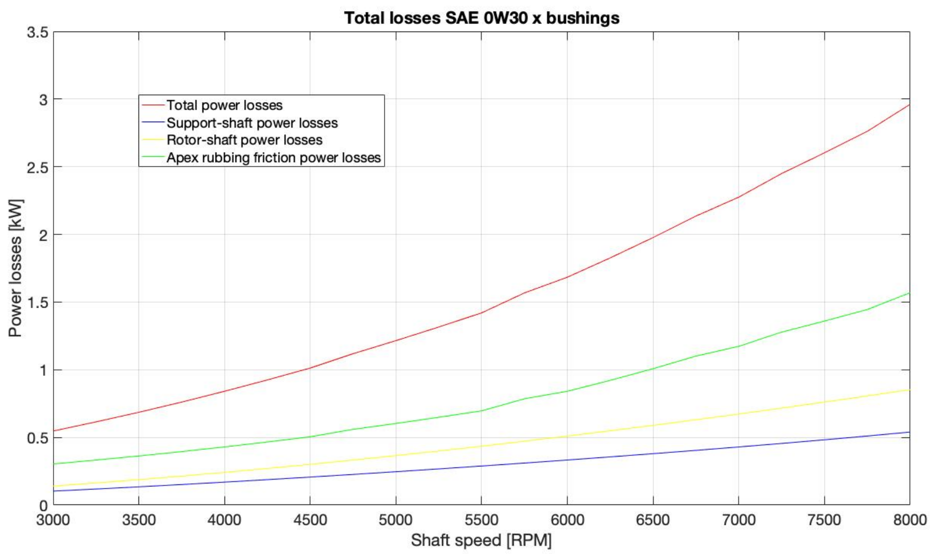


Figure 64: Total losses: SAE 0W30 x bushings

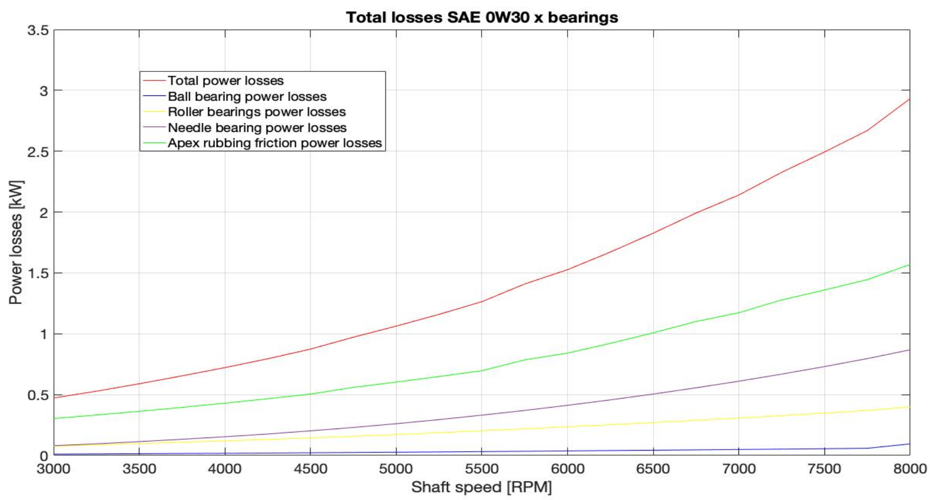


Figure 65: Total losses: SAE 0W30 x bearings

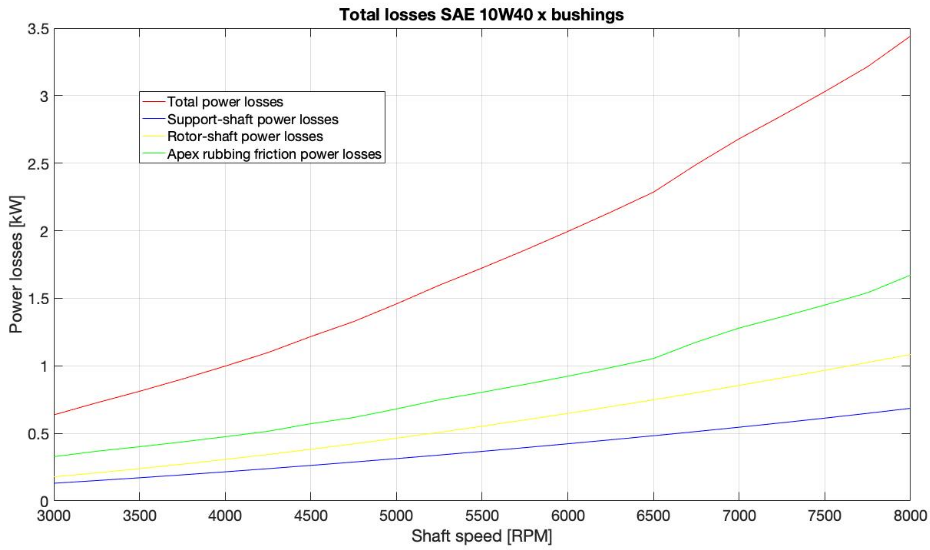


Figure 66: Total losses: SAE 10W40 x bushings

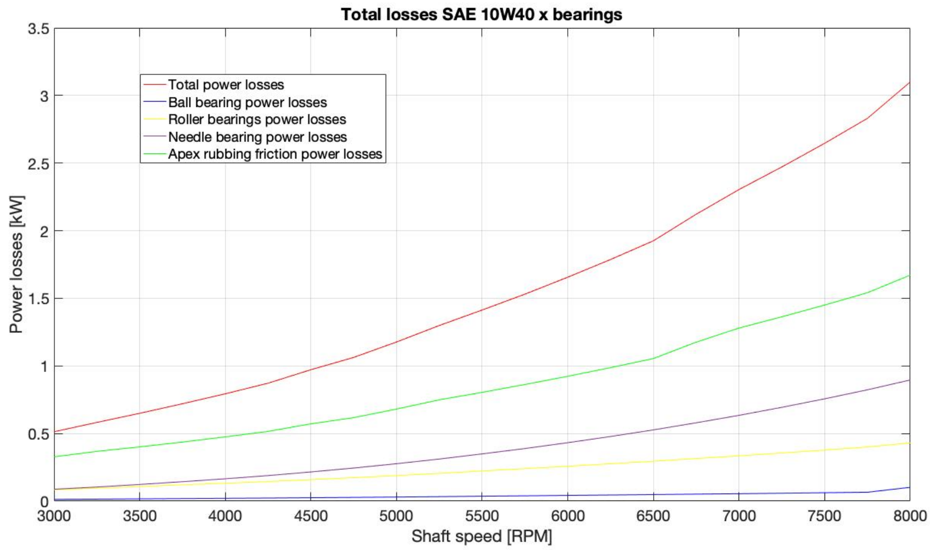


Figure 67: Total losses: SAE 10W40 x bearings

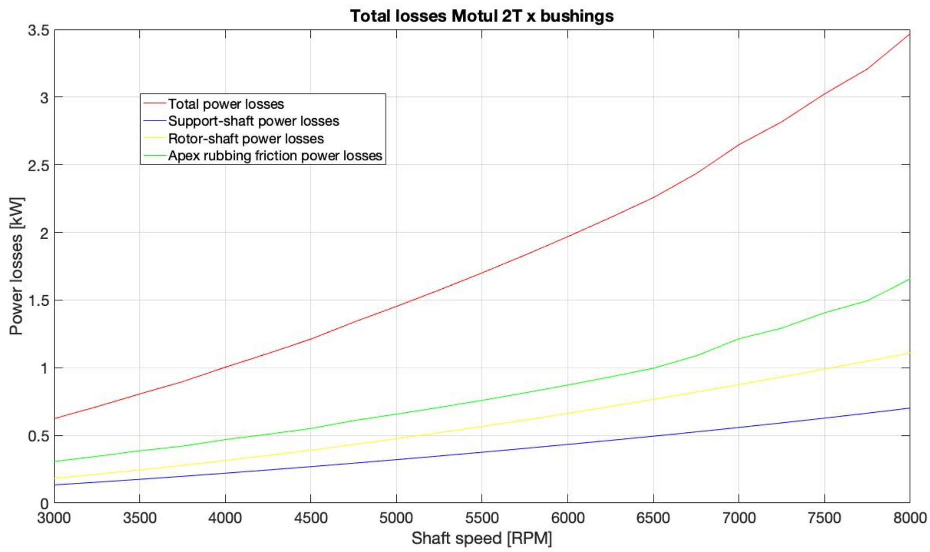


Figure 68: Total losses: Motul GP 2T x bushings

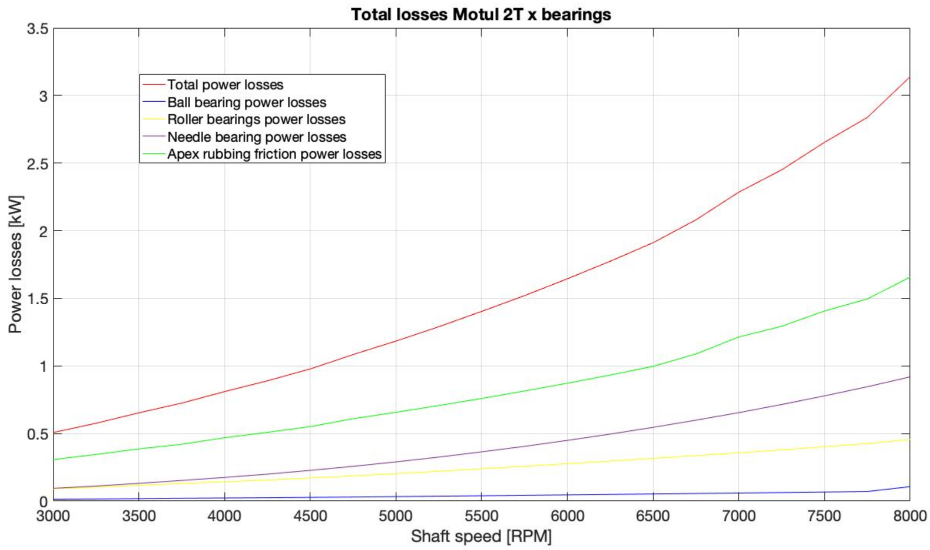


Figure 69: Total losses: Motul GP 2T x bearings

Summing up the obtained results, it is possible to recap what already determined in previous chapters: as supports for shaft and rotor, rolling bearings have a better performance, substantially reducing the friction losses. For what concerns the lubrication, of both apex seals and bearings, the oil with the best performance is SAE 0W30: it shows a total power loss considerably lower than the one shown by the other two lubricants, at any given engine speed.

4.6 Apex seals' wear

The apex seals' wear, and the residual life as well, can be computed starting from the Archard's model.

$$Worn\ volume\ per\ shaft\ rotation = V_{worn} = K \bullet \frac{L \bullet F}{H}\ [m^3] \tag{4.10}$$

$$Seal\ duration = \frac{Sealing\ volume}{Worn\ volume\ per\ shaft\ rotation}\ [Hours\ or\ MCycles] \tag{4.11}$$

As obtained in chapter 4.5.1, the constant wear coefficient (K) is set equal to $10^{-7.9}$. This computed value is perfectly in line with the ones obtained experimentally for a boundary lubrication regime [48].

For what concerns the other parameters to be inserted into formulas 4.10 and 4.11, they are set as follows:

- L, sliding distance: set equal to 485 mm

- F, exchanged normal force: function of the rotational speed, computed in chapter 5.4 (F_H)
- H, hardness of the softer material, cast iron [61]: set equal to $680 \cdot 10^6$ N/m^2 [49]
- Sealing volume: set equal to 70 mm^3

Once defined all these dimensions, it is possible to compute which is the amount of sealing material removed by abrasion per each shaft rotation. The simulation is done following the principle that a hypothetic application of this engine would be as range extender (RE). Range extenders for hybrid electric vehicles generally work at two different but fixed rotational speeds. For this reason, the amount of material worn per shaft rotation and the Apex life are computed as function of rotational speeds from 3000 to 8000 rpm, and not following a driving cycle. In particular the results focus on two typical RE operating speeds: 5000 and 7000 rpm.

Moreover, the computation is repeated per each of the three oils applied as a lubricant for the housing-seals contact.

According to the oil chemico-physical characteristics, what is expected is to find a lower amount of detached material, and consequently a longer apex life, for the oil with the highest density. This is justified by the fact that the higher the oil density, the lower its compressibility and the lower the metal-to-metal contacts, for a given force value so for a given engine speed.

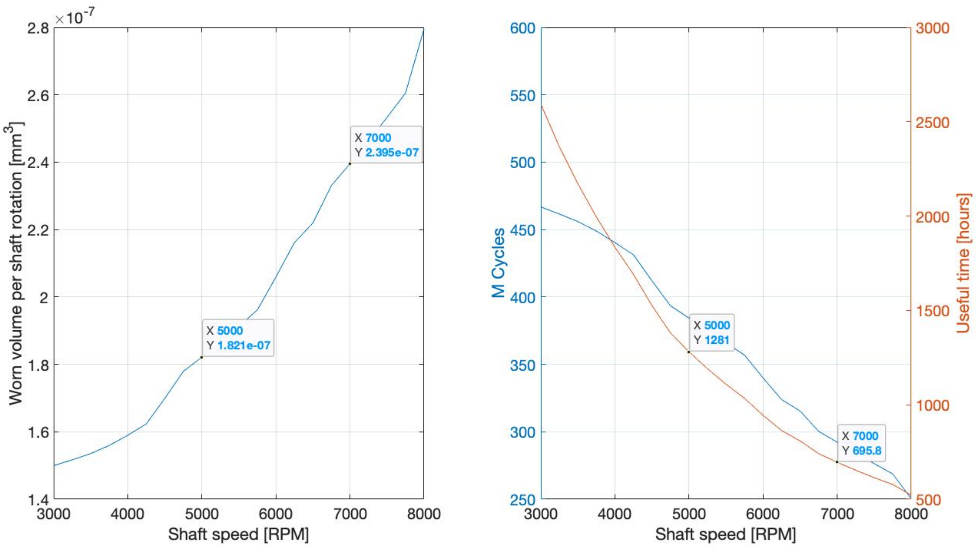


Figure 70: Apex seals life: SAE 0W30

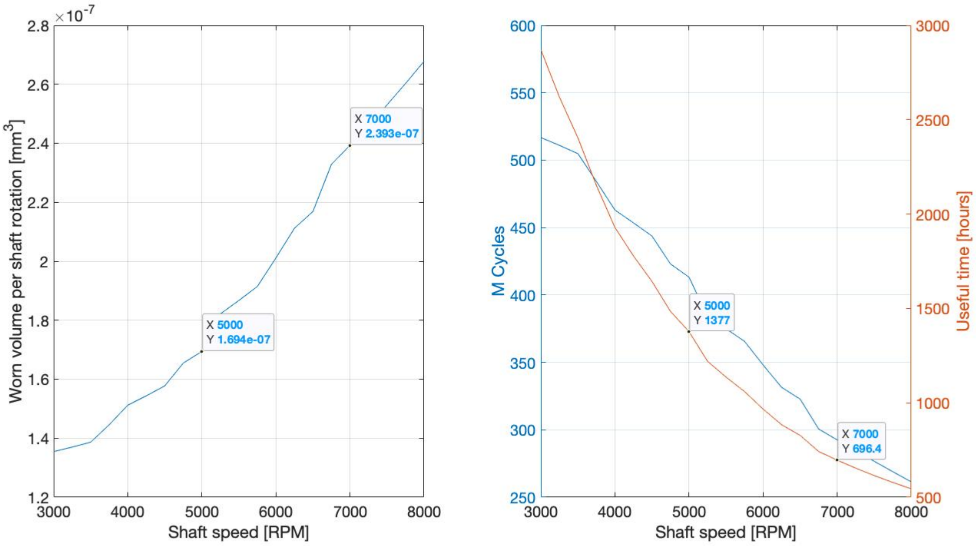


Figure 71: Apex seals life: SAE 10W40

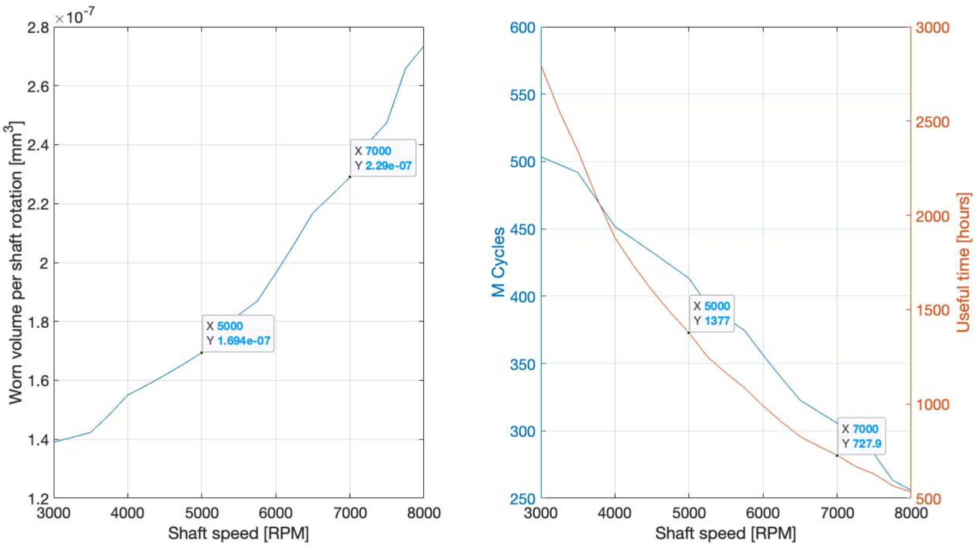


Figure 72: Apex seals life: Motul GP 2T

As it is possible to see in figures 70, 71 and 72, the amount of removed material per shaft rotation, at both 5000 and 7000 rpm, tends to decrease as the oil density increases. The increment in oil density makes reducing the possibility of abrasion due to the asperities of seal and housing. Lubricant oil Motul GP 2T oil ensures a longer Apex seals life, reducing costs and increasing the intervals between interventions.

4.7 Oil consumption

In this chapter will be analysed how the chemico-physical properties of the lube oil can affect its consumption. Mainly, the study focuses on the application of the SAE

0W30, SAE 10W40 and Motul GP 2T oils as sole lubricant for both housing and supports. The variation of oil consumption is expected to depend not just on the oil itself, but on the kind of supports used as well.

The aspect that influences more and more directly the oil consumption is how it is consumed inside the chambers during the combustion process. Wankel rotary engines are characterized by the injection of oil, directly inside the chamber or inside the intake duct, to create a blend at fixed oil-fuel ratio. This solution has been adopted to ensure a correct and complete lubrication of the seals-housing contacts.

Since the oil is completely burned at each combustion cycle, it must be injected three times per rotor spin.

Being the oil sprayed into a hot environment, its behaviour in this area is strongly influenced by the chemical properties. In particular viscosity and volatility play a fundamental role in term of oil evaporation. In order to perform a significant simulation, the walls temperature where the oil deposits has been set equal to 100°C. What is expected is a higher oil consumption for the lubricant with the highest volatility and lowest viscosity (SAE 0W30) and a lower oil consumption for the one with the highest viscosity and the lowest volatility (Motul GP 2T) [56].

Moreover, being the oil injected in a fixed oil-fuel ratio, the lubricant consumption is expected to depend on the fuel consumption as well. Since the fuel consumption is function of the support system efficiency, passing from rolling bearings to bushings an increment in fuel and oil consumption is expected.

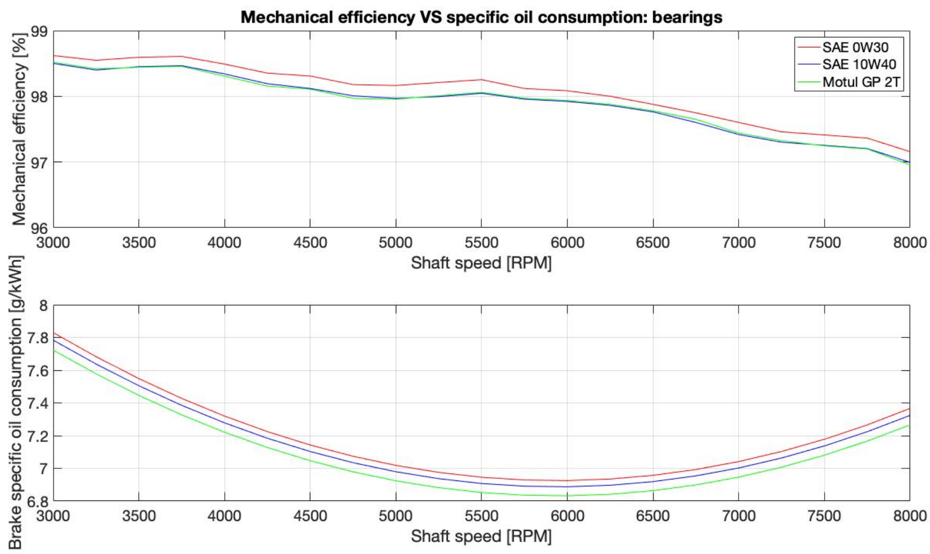


Figure 73: Specific oil consumption vs mechanical efficiency: rolling bearings application

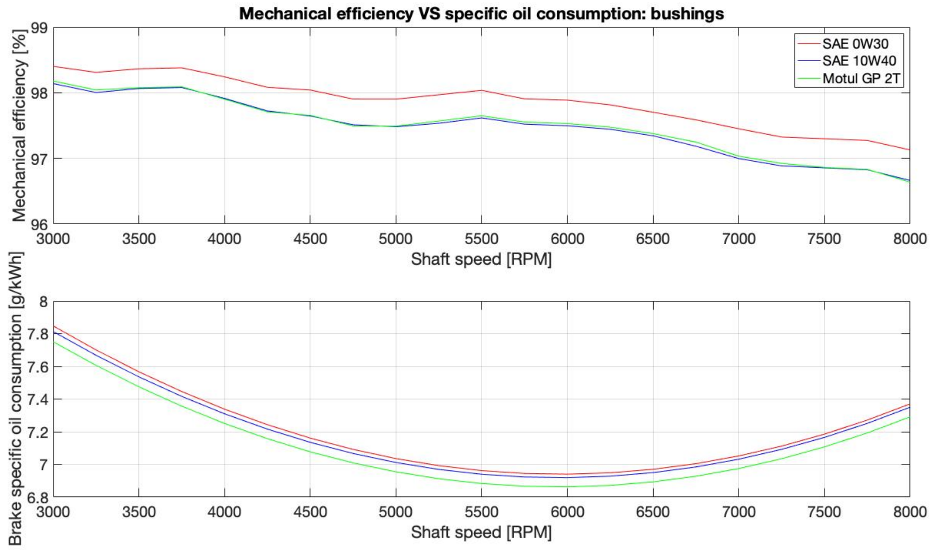


Figure 74: Specific oil consumption vs mechanical efficiency: bronze bushings application

The efficiency range reported in figures 73 and 74 does not take into account the power spent to drive the accessories. The two figures report the efficiency range calculated considering just the apex rubbing friction and the support losses.

The results reported in figures 73 and 74 demonstrate that the lubricant with the highest viscosity and the lowest volatility has the lowest oil specific consumption. Moreover, the application of bronze bushings reduces the supports mechanical efficiency, increasing the fuel consumption and consequently the oil consumption at any given rotational speed.

According to this, it is possible to say that lubricant SAE 0W30, combined with rolling bearings, makes the engine to work at higher mechanical efficiency, reducing the fuel consumption, but its chemico-physical characteristics induce a faster evaporation of the oil inside the combustion chamber. On the opposite, Motul GP 2T ensures a lower mechanical efficiency, increasing the fuel consumption, but its viscosity and volatility make it more convenient from an oil economy point of view. Oil SAE 10W40 falls in the middle.

Since an increment in efficiency induces a decrement in fuel consumption, but at the same time the oil that induces a higher efficiency is the one that tends to evaporate faster, a proper calibration of the oil-fuel ratio must be performed for each lubricant.

4.8 Range extender application

The final step of this project is to simulate the application of a Rotron RT300 as vehicle range extender. The main function carried out by this appliance is to charge the batteries when their state of charge drops below a certain threshold. The engine is expected to work at two different operative points (ECO and BOOST) in order to help the electric motor in meeting the vehicle power demand and, with the energy surplus, to recharge the batteries.

For this simulation several hypothesis must be made [13] [53]:

- The Wankel rotary engine is combined with a battery pack of 42.2kWh energy
- The electric motor can provide a continuous power of 40kW and a transient power of 100kW
- If the battery charge is above 7%, the range extender is OFF and the power is supplied by the battery pack
- If the battery charge is below 7% and the power demand is below 4.5kW, the range extender is OFF and the power is supplied by the battery pack
- If the battery charge is below 7% and the power demand is between 4.5 and 14.7 kW, the range extender is ON and works in ECO mode (5000rpm). The power produced is sent directly to the electric motor and just the surplus (produced-demand) is used to recharge the batteries
- If the battery charge is below 7% and the power demand is between 14.7 and 21.25 kW, the range extender is ON and works in BOOST mode (7000rpm). The power produced is sent directly to the electric motor and just the surplus (produced-demand) is used to recharge the batteries
- If the battery charge is below 7% and the power demand is above 14.7 kW, the range extender is ON and works together with the electric motor supplied by the battery pack to meet the power demand.
- The vehicle power demand is computed simulating the two homologation driving cycles: NEDC and WLTC
- To compute the rolling resistance and the aerodynamic resistance the parameters in table 11 are adopted:

Table 11: Vehicle data for RE simulation [66]

Parameter	Meaning	Value
Cx	Aerodynamic drag coefficient	0,3
ρ	Air density	1,225 kg/m^3
A	Vehicle resistant area	2,35 m^2
kr	Rolling coefficient	0,019
m	Vehicle mass	1400 kg
g	Gravity acceleration	9,81 m/s^2

The data reported in table 11, combined with the hypothesis reported above, make possible to compute the power demand according to the speed profile prescribed by the two homologation cycles:

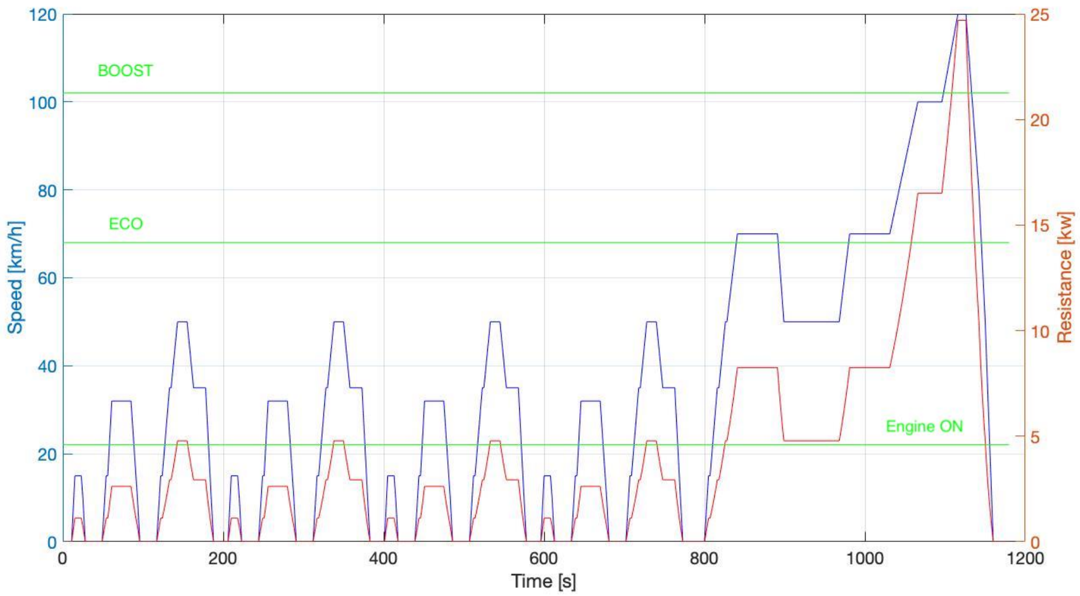


Figure 75: RE simulation: NEDC

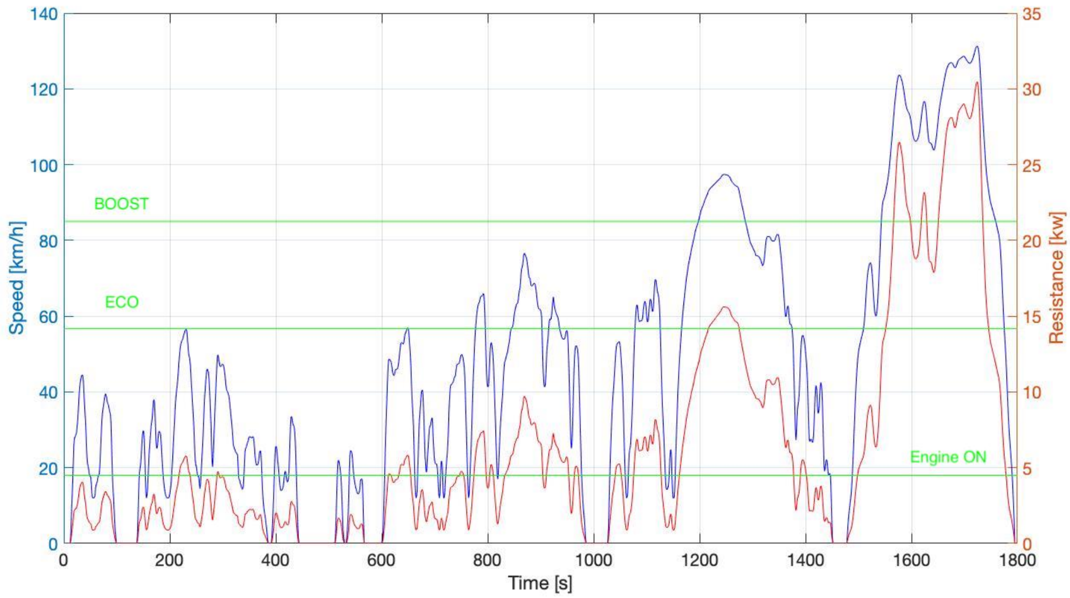


Figure 76: RE simulation: WLTC

The results obtained are validated using the data provided by BMW referring to the i3 range extended car. The aerodynamic and rolling resistance induce a battery discharge: so, according to the capacity of the battery and to the power demand, it is possible to compute the vehicle range in pure electric mode:

Table 12: Results validation

	Declared data [55]	Computed data
NEDC range [km]	359	361
WLTC range [km]	310	312

Since the results of the model tend to converge to the declared ones, with a small difference, the same computational procedure is used to understand how the range extender should work to meet the required power demand.

The simulation as RE is performed applying the Rotron RT300 endowed with rolling bearings, since it results more performing.

Once computed all the parameters related to power loss and efficiency per each lubricant oil, the brake specific fuel consumption is recalculated. Oil SAE 0W30, having a higher efficiency induces a lower fuel consumption so what is expected is a longer extended range when this lubricant is applied.

To work in ECO or in BOOST mode, the Rotron RT300 has to work, respectively, at 5000 and 7000 rpm. At these engine speeds, it provides power outputs equal to 14,7 and 21,25 kW and it has a BSFC that is function of the lubricant:

Table 13: Range extender BSFC

	SAE 0W30	SAE 10W40	Motul GP 2T
BSFC ECO [g/kWh]	331,7	332,4	334, 5
BSFC BOOST [g/kWh]	339,9	340,5	343,6

In order to understand the extended range, a hypothetic fuel tank of 9 litres is used. Knowing which is the fuel consumption for each oil at each operating point and knowing how much time the engine has to work in ECO and in BOOST mode, it is possible to compute the range for both the cycles.

What results is an extended range for the NEDC of 126,5 km for oil SAE 0W30, of 126,2 km for oil SAE 10W40 and of 125,3 km for oil Motul GP 2T. For the WLTC the extended range is 122,2 km for oil SAE 0W30, 121,9 km for oil SAE10W40 and 121,0 km for oil Motul GP 2T.

Together with the fuel consumption, another parameter strongly influenced by the oil is the apex seals wear. This has to be considered in order to understand if the application of a Wankel rotary engine as range extender is competitive with a piston engine, in term of maintenance. Applying the Archard's wear model to the working conditions used to compute the range, the results obtained in term of Apex seals duration for NEDC are: 114000 km lubricating with oil SAE 0W30, 120000 km with oil SAE 10W40 and 122000 km with oil Motul GP 2T. For what concerns the simulation on WLTC, the expected Apex duration is: 107000 km lubricating with oil SAE 0W30, 111000 km with oil SAE 10W40 and 114000 km with oil Motul GP 2T. These durations, expressed in km, must be intended as distances travelled in hybrid mode, so while the range extender is ON and works in charge sustaining mode. If to these distances are added the kilometres travelled in pure electric mode, the total duration of the apex seals must be multiplied by 4 for the NEDC and by 3,57 for the WLTC. This would increment the maintenance interval up to over 400000 km really travelled by the car.

5. Conclusion

The study of the Rotron RT300 Wankel rotary engine lubrication system has been conduced through the development of a suitable mathematical model, used to represent and optimize the Apex, rotor and bearings lubrication. In particular, this model has been applied firstly to compute the forces exchanged between seals and housing, then the power losses due to rolling and rubbing friction, afterwards the efficiency and finally the fuel consumption, the oil consumption and the apex wear.

The main aspects that affect these results are the kind of shaft and rotor supports used and the physico-chemical characteristics of the lubricant oil, such as its kinematic viscosity, dynamic viscosity, density and volatility.

To have a wider representation of the influence of these factors, three different oils have been coupled with rolling bearings or bushings: SAE 0W30, SAE10W40 and a synthetic oil for two stroke applications, Motul GP 2T.

Once understood which configuration has a better performance in term of apex seal life and fuel economy, it has been possible to simulate an application of a Wankel engine endowed with such a lubrication system as vehicle range extender.

As expected, the results that can be observed are [27] [28] [29] [30]:

- The oil with higher density, higher viscosity and lower volatility (Motul GP 2T) ensures lower efficiency, higher fuel consumption but longer apex seals life
- The oil with lower density, lower viscosity and higher volatility (SAE 0W30) ensures higher efficiency, lower fuel consumption but shorter apex seals life
- Oil SAE 10W40 seems to be a trade off between the two
- Rolling bearings applied as supports have higher efficiency with respect to bronze bushings
- From the point of view of a range extender application, oil Motul GP 2T coupled with rolling bearings seems to be the best solution. As can be deduced from a SAE research [67], conventional lubricant oils for 4-stroke applications, do not give the expected results in term of mechanical performance and they are not able to ensure a perfect lubrication of the Wankel rotary engines. With a synthetic oil for 2-stroke applications, it is possible to increase the apex and so the engine life.

These conclusions have been obtained studying the power losses due to hydrodynamic friction inside the bushings, the power losses inside rolling bearings and the rubbing friction due to the seals-housing contact.
For what concerns the evaluation of the application of the Rotron RT300 as vehicle range extender, it has been conduced simulating the driving conditions prescribed by the NEDC and by the WLTC. Once obtained the operating conditions of the engine, the fuel consumption, the drivable range, the apex seals life and so the maintenance interval have been obtained, simulating the application of each of the three lubricants in analysis. What results is an extended range very similar between the three oils applied but a longer maintenance interval, due to apex wear, when oil Motul GP 2T is applied.
The performance levels obtained with Wankel rotary engines applied as range extender are comparable with the ones obtained with conventional reciprocating engines, both in term of extended range and in term of maintenance intervals.

This project can be used as a sort of feasibility study in the case in which a battery electric vehicle needs the application of a range extender, based on a small displacement Wankel rotary engine.
Several companies that produce this kind of engine, such as Rotron or Aixro or Wankel SuperTec, can tune it to reduce fuel consumption, oil consumption, maintenance interval and costs, according to the application the Wankel is used for.

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