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Crankcase pressure control in an internal combustion engine: GT-Power simulation

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CRANKCASE PRESSURE CONTROL IN AN INTERNAL COMBUSTION ENGINE:

GT-POWER SIMULATION

By

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ABSTRACT

The crankcase ventilation system is a very important part of the internal combustion engine but not so much attention is usually paid on what happens in the crankcase. A lot of information is available on what happens in the combustion chamber but not so many papers have been published related to the crankcase ventilation. Actually, efficiency improvements can be obtained with an optimization of the crankcase ventilation system and so fuel can be saved and the overall emissions of CO_2 reduced.

The present is a thesis describing the work done by the author in his internship with Chrysler LLC during the Joint Master Degree program between Politecnico di Torino and University of Windsor.

The work is aimed to simulate the crankcase ventilation system of Pentastar V6 3.6L engine with a 1D/CFD software to see possible ways of improvement and provide some baseline guidance in the initial selection of design parameters for future engines.

DEDICATION

To my family,

which always supported me even from thousands of miles away.

ACKNOWLEDGEMENTS

This is the conclusion of a long year in Windsor, Ontario, Canada, probably the most important of my life up to now. It has been an intense year, practically a blend of hard studying at the University of Windsor and tough work at the Chrysler Automotive Research and Development Centre (ARDC) here in Windsor. The list of people which I met in this year and contributed to my professional and human growth is very long and it is difficult for me to properly acknowledge all of them.

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Abbreviations	
°C	Degree Celsius
1D	Mono dimensional
3D	Three dimensional
Al	Aluminum
bar	Bar pressure
BDC	Bottom Dead Center
CAD	Computer aided design
CAFE	Corporate average fuel economy
CARB	California Air Resources Board
CCV	Closed crankcase ventilation
CFD	Computational fluid dynamics
CG	Center of gravity
CH_3Br	Bromomethane
CH_4	Methane
$CHCl_3$	Chloroform
CI	Compression ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
CRF	Centro Ricerche Fiat
CS_2	Carbon disulfide
cc	Cubic centimeters
EGR	Exhaust Gas Recirculation
EGR	Exhaust gas recirculation
EPA	Environmental Protection Agency
EU	European Union
EV	Electric vehicle
GT	Gamma technologies
h	Hour
hPa	EttoPascal
H_2O	Water
HC	Hydrocarbon
HCN	Hydrogen cyanide
HEV	Hybrid electric vehicle
HP	Horsepower
Hz	Hertz
J	Joule
K	Kelvin
kg	Kilograms
l	Liter
m	Meter
min	Minute
mpg	Miles per gallon

N	Newton
N_2O	Nitrous Oxide
NH_3	Ammonia
NO_X	Nitrogen Oxides
NVH	Noise vibration harshness
O_2	Oxygen
OCS	Carbonyl sulfide
OCV	Open crankcase ventilation
Pa	Pascal
PCV	Positive crankcase ventilation
PCV valve	Pressure control valve
PHEV	Plug-in hybrid electric vehicle
PM	Particulate matter
ppm	Part per million
P-V	Pressure-volume
RPM	Revolution per minute
SI	Spark ignition
SO_2	Sulfur dioxide
TDC	Top Dead Center
VVT	Variable valve timing
W	Watt
WOT	Wide open throttle

NOMENCLATURE

Symbols	
A	Flow area (cross section)
A_{eff}	Effective flow area
A_{ref}	Reference area
A_s	Heat transfer surface area
AFR	Air fuel ratio
$amep$	Accessories mean effective pressure
b	Bore
$bmep$	Brake mean effective pressure
C_D	Coefficient of discharge
C_f	Skin friction coefficient
CPMEP	Crankcase pumping mean effective pressure
D	Equivalent diameter
dp	Pressure differential acting across dx
dV	Differential volume
$dV_{crankcase}$	Differential crankcase volume
dx	Length of mass element in the flow direction (discretization length)
e	Total internal energy (internal energy plus kinetic energy) per unit mass
F	Cylinder axial force
F_f	Friction force
F_g	Gas force
F_l	Liner reaction force
F_n	Thrust force
f	Revolution per second
$lb - ft$	Pound per foot
H	Total enthalpy, $H = e + \frac{p}{\rho}$
h	Heat transfer coefficient
$imep$	Indicated mean effective pressure
$k_{BLOW-BY}$	Coefficient of blow-by
$k_{BLOW-BY_{new}}$	Coefficient of blow-by new
$k_{BLOW-BY_{worn}}$	Coefficient of blow-by worn
l	Con rod lenght
m	Mass of the volume
$\dot{m}$	Mass flow rate
$\dot{m}_{charge}$	Intake mass flow rate
$\dot{m}_{leak}$	Blow-bi mass flow rate
P_r	Pressure ratio
p	Pressure
p_{atm}	Atmospheric pressure

$p_{crankcase}$	Crankcase pressure
p_{turbo}	Boost pressure
R	Universal gas constant
r	Crank radius
$rfmep$	Rubbing friction mean effective pressure
T_{ICE}	Engine torque
$T_{crankcase}$	Crankcase temperature
T_{fluid}	Fluid temperature
T_o	Upstream stagnation temperature
T_{wall}	Wall temperature
U_{is}	Isentropic velocity at the throat
V	Engine displacement
V_{cylbay}	Cylinder bay volume
$V_{crankcase}$	Crankcase volume
V_{total}	Total crankcase bay volume
V_u	Unitary cylinder displacement
$\dot{V}_{blow-by}$	Blow-by volume flow rate
$\dot{V}_{intake}$	Intake volume flow rate
v	Velocity
W	Work
W_a	Accessories work
$W_{crankcase}$	Crankcase work
W_{ICE}	Engine work
W_{ICE}	Engine work
W_{ind}	Indicated work
W_{rf}	Rubbing friction work

Greek Symbols

γ	Isentropic exponent
ζ	Pressure loss coefficient
η_o	Organic efficiency
θ	Crankangle
λ	Con rod ratio
μ	Molecular mass
μ_d	Dynamic coefficient of friction
ρ	Density
ρ_{air}	Air density
ρ_{is}	Density at the throat
ρ_o	Upstream stagnation density

1 INTRODUCTION

1.1 Problem statement

One of the main problems which engineers have to face when they have to design a new car is to meet the always more demanding pollutants regulations. The combustion inside the engine of cars and trucks removes O_2 from the atmosphere and release an equivalent amount of H_2O and CO_2 together with numerous other pollutant compounds as unburned hydrocarbon HC, carbon monoxide CO, nitrogen oxide NO_x, reduced nitrogen (NH_3 and HCN), sulfur gases (SO_2, CS_2, OCS), halo-carbons ($CHCl_3$ and CH_3Br), and particulate matter PM [1].

The emission of these gases in the air due to the burning of fossil fuels has led to global alterations in the composition of our atmosphere. Air pollutants in the atmosphere can generate problems for human health.

In order to sell cars, Car Makers have to face always stricter regulations, which are different from country to country, but all of them prescribe a general trend toward the reduction of the pollutant emissions and CO_2.

Historically, the state of California was the first at introducing regulations for the restriction of the emissions levels for motor vehicles in 1963 and in the following years these directives became always more stringent.

The most important legal restrictions on car exhaust pollutant emissions are [1]:

- CARB (California Air Resources Board) regulations;
- EPA (Environmental Protection Agency) regulations;
- EU (European Union) regulations;
- Japanese regulations.

The first and the second are North American regulations, while the third is European and the last is applied for cars sold in Japan. All of this regulations aim to reduce the pollutants emissions of the cars, so that lower pollution will be generated by future cars.

In these regulations the CO_2 is not mentioned; this gas is a natural component of the atmosphere and is not considered as a pollutant. It is one of the major products of combustion together with H_2O and is generated by the chemical bond of the carbon present in the fuel and the oxygen of the air. Nevertheless, the always increasing number of cars in the world has led to a general increase in the percentage of CO_2 in the atmosphere and also if it is not a direct pollutant for humans it is one of the compounds responsible for the greenhouse effect and the related climate changes. The CO_2 and the other greenhouse gases like methane CH_4 and nitrous oxide N_2O concentration is significantly increased in the industrial area and this increment is due to human activities (see Figure 1.1-1 and Figure 1.1-2).

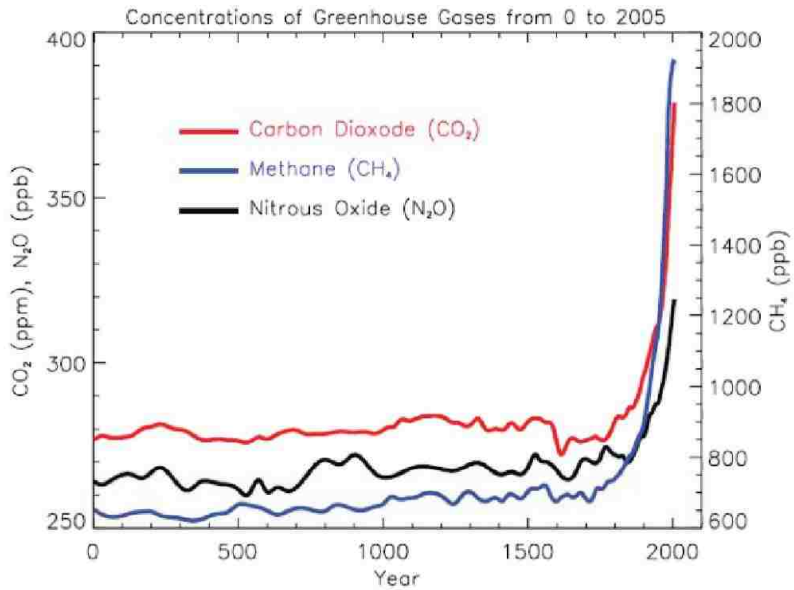

Figure 1.1-1: Concentrations of Greenhouse Gases [1].

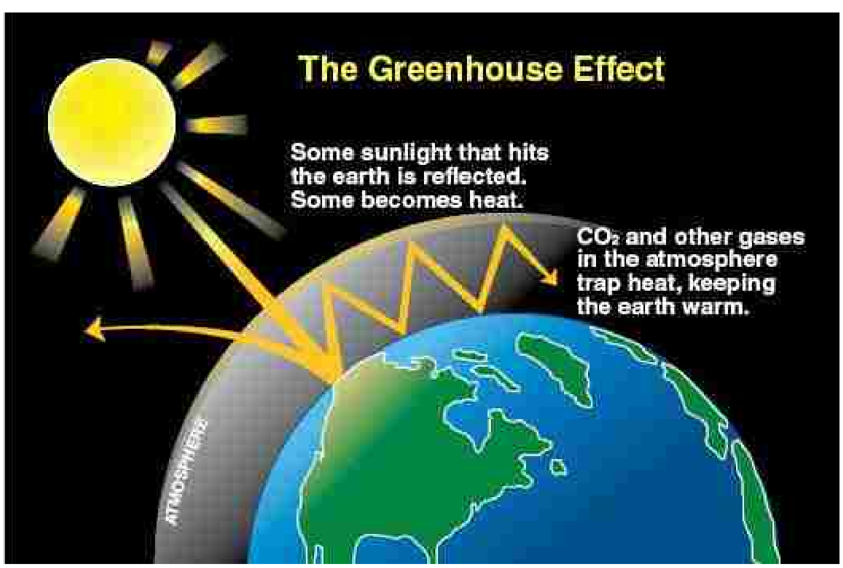

Figure 1.1-2: The Greenhouse Effect [2].

For a given fuel, the amount of converted carbon dioxide in the exhaust is a direct index of fuel consumption. **This means that the only way to reduce carbon dioxide emissions is to reduce the fuel consumption** [1].

An engine which burns less fuel also produces less carbon dioxide. Even if the CO_2 is not a pollutant emission, there are other regulations which limit the production of CO_2 by the cars. For example, in 2009 the EU Parliament approved a law with the aim of reaching new targets for new cars sold; the emissions targets are 130 g/km of CO_2 by 2015 and 95 g/km of CO_2 by 2020 for cars sold within Europe and an even more stringent target range of 68-78 g/km for 2025.

At the same time in U.S.A. the federal standard regulating vehicle fuel economy (and so directly the CO_2 produced by cars) on a fleet-average basis is called CAFE (Corporate Average Fuel Economy); in August 2012, the Obama administration put a new CAFE rule into law, prescribing a target of 155 g/km for 2016 and an even stricter target of 101 g\km for 2025, which means an average fuel consumption of 54.5 mpg (23.1 km/l).

All these CO_2 emission limits are the average maximum allowed for Car Maker's entire fleets, registered in the EU and U.S.A. respectively.

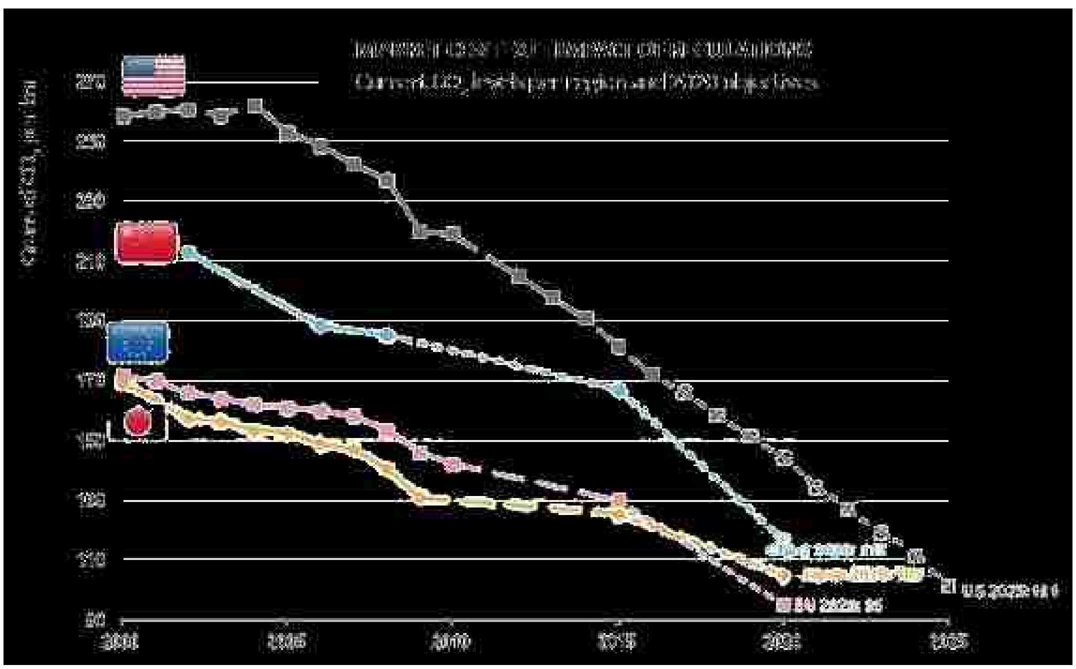

Figure 1.1-3: Market context-impact of regulations (adapted from [39]).

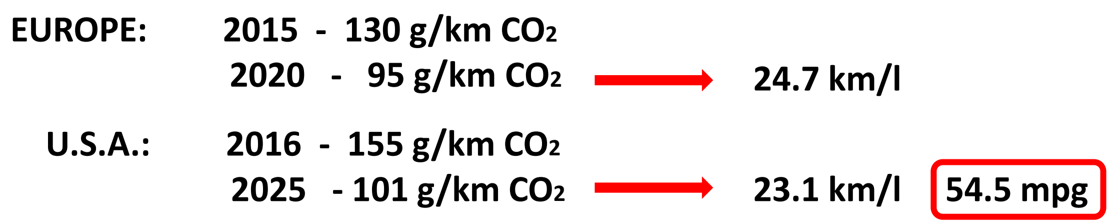

Figure 1.1-4: Future targets for light duty vehicle fuel economy.

In order to reach these targets all the areas of the vehicle should be optimized so that less fuel could be burned for a given mission of the car; the areas of major interest in this context are (see Figure 1.1-5):

- aerodynamic drag reduction;
- transmission and final drive friction reduction;
- vehicle mass reduction;
- tire and rolling resistance reduction;
- engine mechanical and pumping losses reduction;
- electrical power losses reduction.

Figure 1.1-5: Dodge Challenger Mopar 2009 [3].

Despite the spread of electric vehicles (EVs) and hybrid electric vehicles (HEVs) predicted for the following years the internal combustion engine will still represent the major source of power for the cars.

Currently engines, only 30-35% of the energy of the fuel is converted into usable work by a SI engine and 35-40% for a CI engine; this means that improvements are still possible. Some possible ways of increasing the efficiency of the internal combustion engine for a decrease in fuel consumption are:

- advanced thermal control;
- combustion optimization (lean stratified spark ignition);
- downsizing;
- direct injection;
- stop & start;
- turbocharging and supercharging;
- HEV, PHEV;
- mechanical friction reduction;
- more transmission gears (8-9-10 gears);
- hydraulically-actuated variable valve timing (VVT, Multi-air).

At the same time, the use of fuel with a lower content of carbon (the so called "low carbon fuel") as methane and hydrogen will lead to a general decrease in the production of carbon dioxide by transportation.

One of the areas of the engine which is not usually treated in great detail is the crankcase ventilation system of the engine. Here the motion of gases caused by the rotating elements inside the crankcase (crankshaft, counterweights and connecting rods) and by the reciprocating motion of the pistons generates pumping losses which directly reflect on a lower power output of the engine.

Not so much attention is usually paid on what happens in the crankcase. A lot of information is available on what happens in the combustion chamber but not so many papers have been published related to the crankcase ventilation. Instead, the optimization of the crankcase ventilation system will potentially lead to a reduction of the pumping losses of the crankcase and so to an increase of the engine efficiency. This will improve fuel efficiency and reduce overall emissions of CO_2 in the near future.

Figure 1.1-6: Possible areas for engine efficiency improvement [3].

1.2 Target

As previously mentioned, the optimization of the crankcase ventilation system will potentially have a benefit on engine efficiency, thus reducing the fuel consumption of the car. Generally speaking, a reduction of the mean pressure inside the crankcase means lower pumping losses on the engine; on the opposite, an increase of crankcase pressure will lead to increased pumping losses [4]. As a consequence, a crankcase design which is able to keep a lower gas pressure will improve the engine efficiency. A very interesting design of the crankcase is the dry sump with a scavenge pump. The dry sump is the common solution employed in high performance motorcycle and racecar engines; in this design the dry sump replaces the wet sump commonly used and the oil is stored in a separate tank. The scavenge pump provides the vacuum inside the crankcase, sucking out the gases (called *blow-by gases*) which enter the crankcase during the operation of the engine. In this way a considerable reduction of power losses is reached.

In wet sump engines, the gas flow exchange between the crankcase bays introduces power losses; the piston motion displaces fluid in the crankcase which is forced to flow from a high pressure zone to a lower pressure zone. The friction between the fluid and the hardware of the crankcase generates losses, so less power to the shaft.

Also, the interaction of the crankcase gases with the splashing oil in the crankcase increases the fluid density and so the drag which the crankshaft has to overcome in his rotation. The present work aims to understand the most important parameters affecting the pumping losses inside the crankcase. This work provides a set of guidelines for the design of the crankcase ventilation system in order to minimize the flow losses.

A 1D simulation of the crankcase ventilation system has been performed in order to obtain numerical results which bring indications for future designs; the effect of the main parameters on the crankcase pumping losses has been evaluated exploiting a 1D simulation software. The exploited software is GT-Power Version 7.4, by Gamma Technologies. The created 1D model is based on the geometry of the Chrysler Pentastar V6 3.6 liters.

1.3 Methodology

The whole purpose of creating an engine model with the use of software is to simulate its behavior exploiting the high computational capacity of today's computers. The simulation should provide answers which, otherwise, should come from very expensive and time demanding testing. The model, as a consequence, should strictly behave like the real engine, within a given range of accuracy.

The first step is the model creation starting from CAD data or other sources; in any case it is hardly difficult that this first model behaves like the real engine, because some parameters are difficult to be set a priori. Hence, some experimental measurements are necessary to have real data to compare with the output of the software: with those data the model is tweaked until it describes the characteristics of the real engine. This second step is called model correlation or calibration. Once the model is correlated it becomes a powerful tool for engineers to predict the effect of modifications without performing costly testing activity.

In the present work, the crankcase ventilation system of the chosen engine has been modeled with the use of a 1D simulation software for engines, GT-Power Version 7.4 by Gamma Technologies: a model of the "**Engine**" (which includes intake manifolds, cylinders, combustion process, intake and exhaust valves, exhaust system…) and a model of the “**Crankcase Ventilation System**” are necessary. The model of the “Engine” was an existing model, describing in detail the operations and the properties of the Pentastar V6 on the combustion side; it was given to the author by Chrysler LLC, which created and used it for previous researches. There was not an existing model of the “Crankcase Ventilation System” of the considered engine and so it has been created by the author. The “Crankcase Ventilation System” model has been created starting from the geometry of the real engine using the CAD data and then it has been correlated to fit some experimental data (blow-by production as function of speed and load and flow through the PCV valve).

The original model has then been modified in some key parameters in order to evaluate the effect of the change.

1.4 Thesis organization

The remaining part of the present work is organized in the following chapters:

- **CHAPTER 2**: this chapter presents all the literature review performed by the author in the early stage of the project; it treats in detail all the most important aspects of the crankcase ventilation system commonly used by modern engines, in order to give to the reader a better understanding of the topic.
- **CHAPTER 3**: in this chapter some basic equations for the prediction of the crankcase pressure are explained and some numerical results are presented. It also contains some considerations for the reduction of the pumping losses inside the crankcase.
- **CHAPTER 4**: this chapter describes the steps required for the model creation and calibration, along with a description of the simulation process implemented in the software GT-Power.
- **CHAPTER 5**: in this chapter the discussion of the results obtained from the model is presented; the effect of the different variables on the crankcase pumping losses is also evaluated.

- **CHAPTER 6**: the last chapter includes the conclusion and the recommendations. The final simulation results are summarized once again together with the indications for further studies and researches.

2 LITERATURE REVIEW

2.1 Crankcase Ventilation System

In this section a brief description of the state of the art of the crankcase ventilation systems currently used in modern engines is presented.

The main task of the crankcase ventilation system is to evacuate the gases inside the crankcase: the crankcase of an engine is not an empty space but it is full of burned gases, called **blow-by gases**, and air. The combustion chamber represents the major source of blow-by [5]; during the operation of the engine, a fraction of the high pressure gases inside the combustion chamber can escape the sealing of the piston rings and enter the crankcase. The period during an entire cycle of the engine during which the majority of the blow-by gases are generated is during the compression and the expansion stroke. During combustion, a high peak pressure inside the crankcase is generated, and a portion of the gases leak to the crankcase around the pistons rings which are not able to guarantee a perfect sealing of the chamber and are not entirely gas-tight.

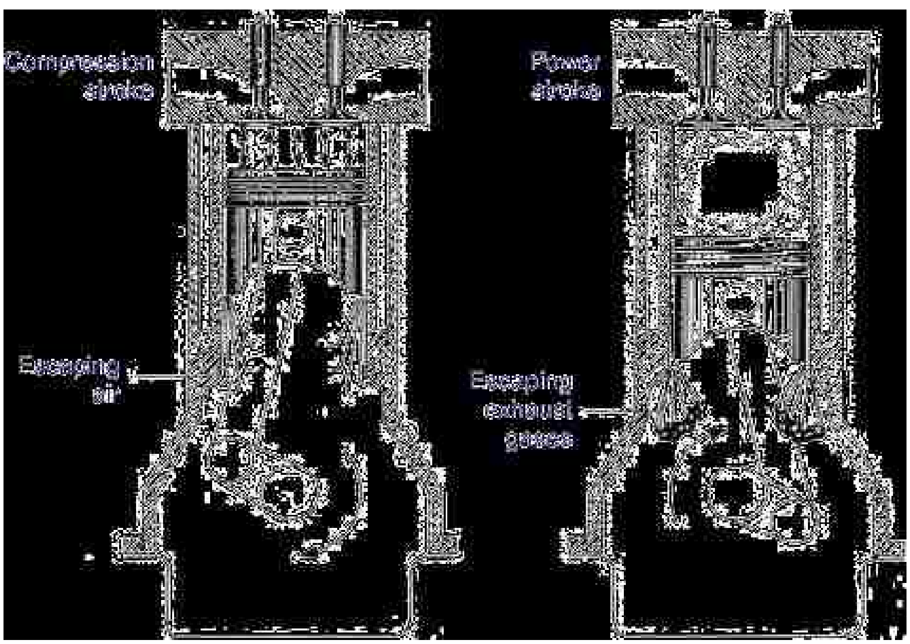

Figure 2.1-1: Combustion blow-by [5].

The blow-by gases production is not constant but changes as function of the engine working conditions like torque, engine speed, throttle position, etc... and so the pressure inside the crankcase. Generally, the blow-by production is greater during high load, accelerations or high engine RPM [6].

Even though the combustion chamber represents the major source of blow-by gases, there are other important sources like turbocharger shaft bearings and in some cases leaky valve stems. The shaft connecting the compressor and the turbine of a turbocharger, for example, is mounted on bearings which need to be lubricated due to the high rotational speed; a small portion of the compressed air on the compressor side can mix with the lubricated oil of the bearing and so be drained back in the oil sump through the oil-drains. The same can happen on the turbine side.

The majority of the blow-by gases is constituted of contaminants like unburned hydrocarbons (HC), carbon dioxide (CO_2) and vapor of water (H_2O) [6]; due to the fact that the majority of the blow-by flow rate happens during the combustion phase, also other pollutant emissions like CO, NO_x and PM are present in the crankcase. These gases are dangerous for the crankcase itself and for the oil in the oil pan; for example in a SI engine, a large amount of blow-by is constituted of hydrocarbons: during the combustion in a SI engine, the high pressure pushes the gases in the crevice regions between piston rings and cylinder wall and the majority of these gases are unburned fuel air-mixture. These unburned hydrocarbons are then transported by the high differential pressure in the crankcase where they condense and accumulate and so impair the quality of the lubricant.

The situation is still worse during a cold start of an indirect injection SI engine: in this case the injected fuel might not vaporize enough and the percentage of fuel which is not vaporized tends not to react and usually a dilution of the cylinder wall lubricant happens [1]. The dilution of the oil is very dangerous for the engine because the lubrication effect of the oil is not optimal.

Also the vapor of water is dangerous for the crankcase: here it can condense and even freeze during cold ambient temperature when the engine runs below operating temperature for short-distance at light load. This can potentially block the oil-supply system and also the crankcase ventilation system, damaging the engine [7]. Moreover, the other pollutant emissions can degrade the oil properties, bringing damage to the engine during normal operating condition.

For these reasons, a way of evacuating this harmful gases from the crankcase should be found; the problem is connected to the fact that in the crankcase, together with air, vapor of water and pollutant emissions, there is also an oil mist, resulting from the crankshaft rotation and the spraying out of the lubricating circuit which creates a haze of oil.

The first target of the crankcase ventilation system is to evacuate the dangerous gases in the crankcase recovering as much as possible of the lubricating oil particles, separating them from the blow-by gases; for this purpose an **oil separator** is used.

The second important target of the system is to prevent the crankcase pressure to build-up, reducing the risk of leakages of oil from the oil pan and keeping the pressure level as constant as possible; a pressure control valve (also called **PCV valve**) is used for this purpose.

Two approaches can be used in order to bring out the gases from the crankcase: the first one is to simply connect the crankcase to the outside environment to vent the gases. This approach was used in the past when there were not so many regulations in terms of pollutant emissions and it is called Open Crankcase Ventilation (OCV). The gases can be filtered or not before they reach the outside environment to recover the oil.

The second approach is the Closed Crankcase Ventilation (CCV); the main idea of a CCV system is to vent the crankcase gases in the intake manifold, in order to burn them in the combustion chamber. The crankcase is connected to the engine head with channels built in the engine block called **drainbacks** or **oildrains**; these channels allow the lubricating oil from the head to flow back in the oil pan exploiting the gravity force. Instead, the blow-by gases flow from the crankcase to the cylinder head thanks to a positive differential pressure. Alternatively, the blow-by gases can reach the cylinder head through the volume between the engine block and the cover which is the housing for timing chains and tensioner arms (**front cover**); from the engine head they reach the intake manifold, passing through an oil separator and a PCV valve.

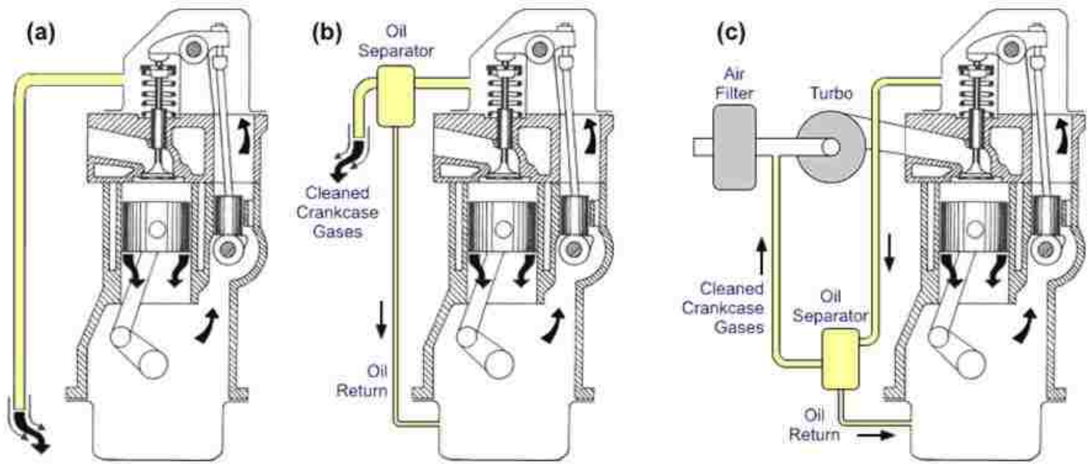

Figure 2.1-2: Crankcase ventilation systems, a) unfiltered OCV, b) filtered OCV, c) CCV [5].

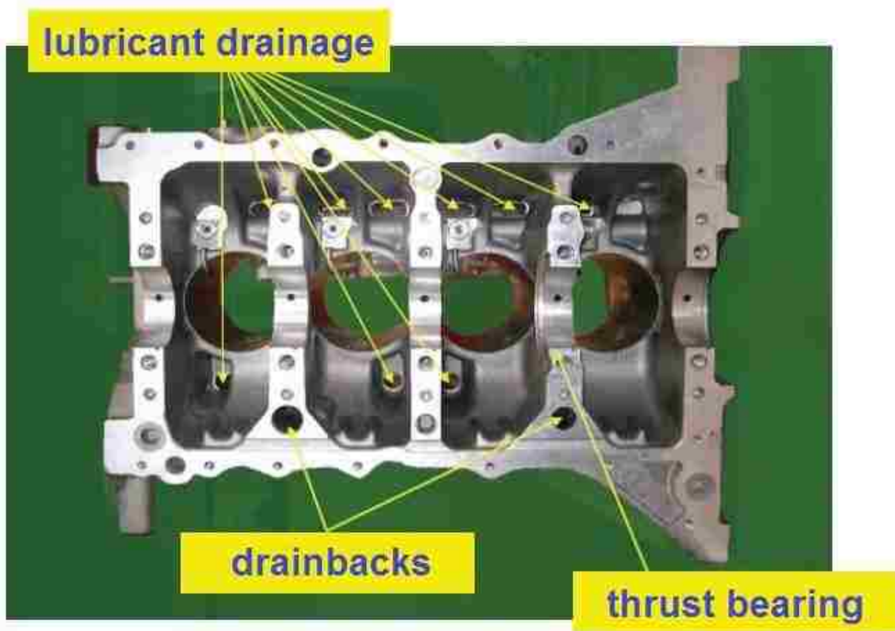

Figure 2.1-3: Bottom view of the crankcase of a 4 cylinder engine [8].

This approach drastically reduces the emissions of the engine and is mandatory for compliance with current regulations. The gases are filtered for two reasons: the first is to recover the oil, reducing in this way the global engine oil consumption and so reduce the oil top up; the second is to avoid that the oil is burned in the combustion chamber which would drastically increase the pollutant emissions.

To scavenge the crankcase, the differential pressure between the crankcase itself and the intake manifold is exploited; it is usually a positive value because the pressure in the crankcase is close to atmospheric pressure (also if important fluctuation of the pressure are possible in the different engine working conditions) while the intake manifold pressure is always lower than atmospheric one in normally aspirated engine due to the pressure losses in the air filter, throttle, intake manifold etc… For this reason, the system is also called “Positive Crankcase Ventilation”, because it exploits this positive differential pressure.

In turbocharged engines, the gases are usually vented before the compressor, because in the intake manifold there is generally a greater pressure and a reverse flow would be possible.

The pressure level inside the intake manifold is not constant but varies in response to the different engine working conditions and the same happens for the pressure inside the crankcase; the crankcase pressure also fluctuates due to the pistons reciprocating motion, so it is a function of the crankshaft position. In common road engines, it is usually required to have a crankcase pressure slightly lower than the atmospheric one because this will impede any oil leakages: common target value for the crankcase pressure is around 20 to 30 hPa lower than the atmospheric pressure [9]. In order to keep the crankcase pressure as constant as possible a PCV valve is used in common engines. It is usually positioned at the end of the cam cover and it is connected through a pipe with the intake manifold, usually in the high depression zone behind the throttle.

The pressure inside the crankcase should be as constant as possible, otherwise the oil in the oil pan would decrease the lubrication property and would move and splash too much in the crankcase.

The pressure inside the crankcase can be lower than the intake manifold pressure in some transient conditions; in this situation the intake manifold air cannot flow in the crankcase through the PCV valve because this is a one-way valve, but another valve can be used to keep the crankcase pressure at the required level: a **make-up-air valve**, which is a two-way valve connecting the air filter (high pressure zone) to the crankcase, is used to push fresh air in the crankcase and scavenge the blow-by gases. In case of high blow-by flow (for example, during high load and accelerations) the flow can pass through the PCV valve and the make-up-air valve and reach the intake side in two points; if the crankcase pressure reaches a too low value, a flow through the make-up-air valve bring the pressure level to the required value. In other engines, a simple pipe (without any make-up-air valve) between intake side and crankcase side is used to put fresh air in the crankcase.

The process of releasing the blow-by gases in the intake manifold through the PCV pipe and admitting fresh air inside the crankcase through the make-up-air pipe is called "**engine breathing**" [10]: it is similar to the normal breathing process of humans, in which air is aspirated and then expelled.

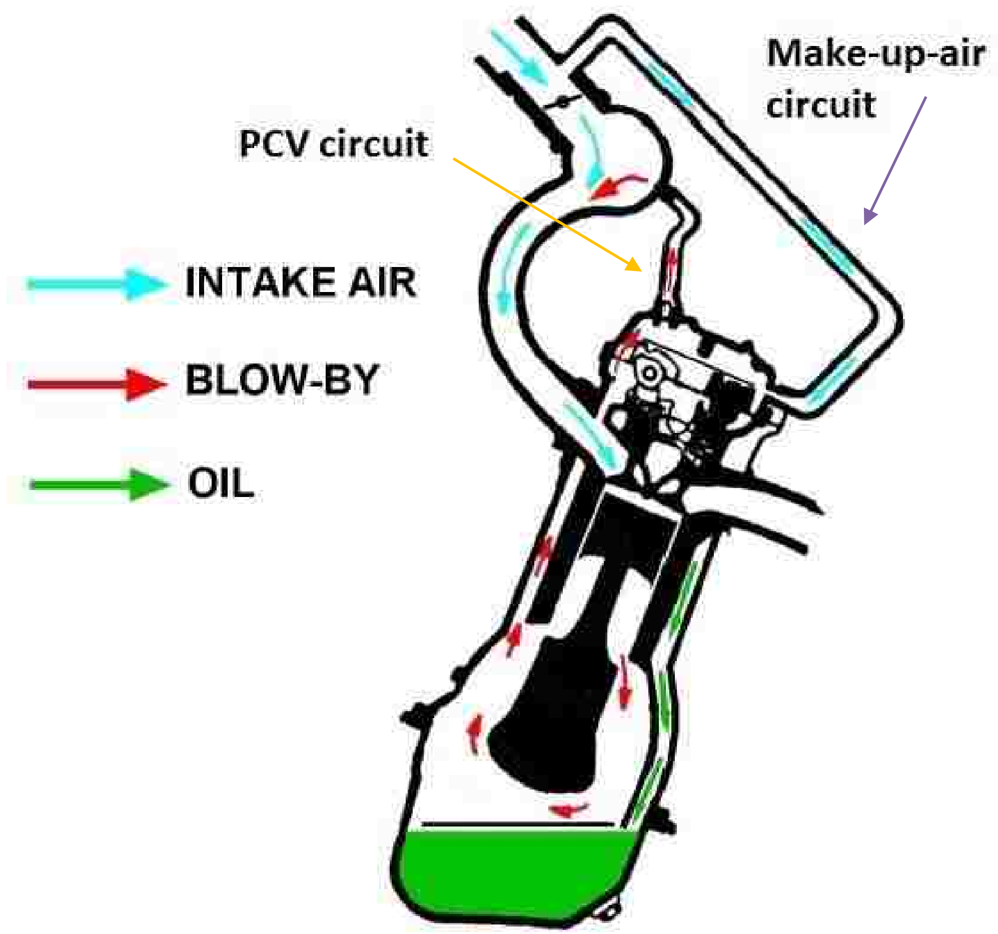

Figure 2.1-4: Crankcase ventilation system (adapted from [11]).

In the following paragraphs the most important elements of the crankcase ventilation system will be treated in detail to give to the reader a better understanding of the topic.

2.2 Crankcase Ventilation components

The components involved in the crankcase ventilation system are:

- Pistons.
- Pistons rings.
- Cylinder liners.
- Connecting rods.
- Crankshaft
- Bearings & Counterweights.
- Crankcase.
- Windage tray.
- Oil sump.
- Drainbacks.
- Front cover volume.
- Oil separator.
- PCV valve and circuit.
- Make-up-air valve and circuit.
- Air filter.
- Turbocharger/Compressor.
- Intercooler.
- Valves.
- Cams.
- Cylinder head.
- Others…

In the present work a detailed description of the most important elements is presented.

2.3 Pistons

The piston is probably the most important component inside the engine: the functioning of the engine is strictly dependent on the motion of the pistons.

The pistons have to accomplish the following tasks:

- perform the intake and compression of the charge;
- discharge the exhaust gases outside the cylinder;
- guarantee the sealing of the combustion chamber;
- transmission of the power to the connecting rod.

Inside an internal combustion reciprocating engine, the high peak pressure inside the combustion chamber pushes the piston down in the cylinder due to the combustion of the fuel; the transitional motion of the piston is converted in the rotary motion of the crankshaft which then guides the rotation of the wheels (after passing through the driveline of flywheel, gearbox, differential and semi-shafts).

The shape of the piston of a gasoline engine is different from the piston of a diesel engine; the main difference is the top surface. In both pistons the top surface is usually machined in such a way to accommodate the valve seats, so that a greater valve lift can be used; in case of diesel engine, on the top surface, a bowl which acts as combustion chamber is present.

The typical diesel piston design is showed in Figure 2.3-1.

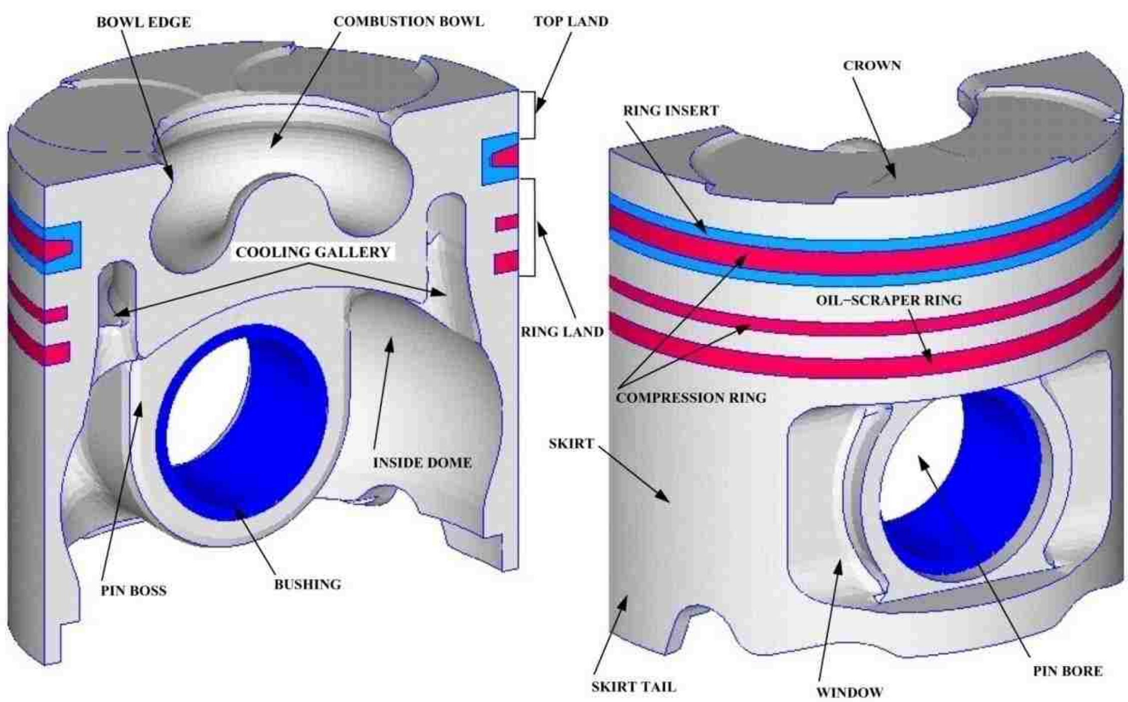

Figure 2.3-1: Typical piston of a diesel engine [8].

Pistons are usually made of aluminum alloy because this reduces the mass and so the inertia forces related to the accelerations and decelerations of them; this has a positive effect on the vibrations delivered by the engine to the car because lower reciprocating masses are involved. To reduce its weight, the piston is usually hollow at the bottom.

The piston itself is not able to seal the combustion chamber because a small radial gap is present between piston and liner: this allows the piston to expand inside the cylinder during the warm up. The sealing is improved with the adoption of piston rings, located in appropriate grooves inside the piston skirt.

The piston moves inside the cylinder liner at very high speed, depending on engine working conditions, and the piston skirt, which is a load-bearing surface, allows the piston to move correctly inside the cylinder, carrying the side load. A thin film of oil separates the cylinder wall and the piston skirt so that the friction and heat generation are reduced.

The piston is the component subject to the highest stress from the thermal point of view; the temperature in the middle of the piston sky can reach 350-380 °C while at the border the temperature is around 260-280 °C [12] .

The clearance between piston and liner has a great influence on the blow-by because allows high pressure gases in the chamber to flow in the crankcase; in any case, this clearance is necessary for the engine functioning because the piston thermal expansion coefficient is much greater than the one of the liner (or cylinder block); as said before the piston is made of aluminum alloy, while the engine block is usually cast iron and has a thermal expansion coefficient which is much less than the aluminum one. A too low clearance will increase the friction in the cylinder and the risk of engine seizure at high load (so high temperature); for this reason, the clearance is usually a trade-off between the need of low blow-by and low friction inside the cylinder.

	Cast Iron	Aluminum alloy
Density $(\boldsymbol{kg/m^3})$	7270	2700
Thermal conductivity $(\boldsymbol{W/m \cdot K})$	52	150
Thermal expansion coefficient $(\boldsymbol{10^{-6}/K})$	12	23
Young' s module $(\boldsymbol{kN/mm^2})$	115	70

Table 2.3-1: Properties of cast iron and aluminum alloy [13].

Table 2.3-1 gives some general value of properties of materials commonly employed in internal combustion engines. The typical composition of an aluminum piston also contains from 10 to 12 % of silicon; the silicon helps to reduce the thermal expansion of the piston inside the cylinder due to the high temperature working condition that it has to withstand [13]. This means that thermal expansion coefficients around 19.5 X $10^{-6}K^{-1}$ can be reached. The high value of thermal conductivity property of the aluminum helps in reducing the risk of hot spots which can lead to surface ignition and abnormal combustion.

In order to reduce the temperature on the piston pin, the piston is usually lubricated with oil sprayed from the bottom by special nozzles in the cylinder block. The oil is then guided in a cooling gallery inside the piston structure to cool it and reduce the thermal stress. Although this sprayed oil is beneficial for the durability of the system, it promotes the formation of oil mist inside the crankcase, so a greater filtering performance is required for the oil separator to recover the oil from the crankcase before venting the crankcase gases in the intake manifold.

Although the primary movement of the piston is translation along the cylinder axis, the piston has also two other degrees of freedom due to the clearance between piston skirt and cylinder liner. These are:

- rotation around the piston pin (tilting);
- translation in direction perpendicular to cylinder axis.

These two movements are explained with Figure 2.3-2 and are called "piston slap movements".

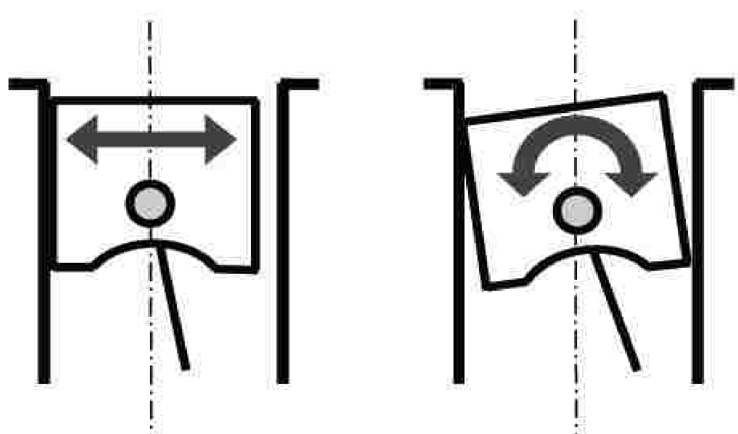

Figure 2.3-2: Secondary piston movements (piston slap movements) [8].

When the piston is close to the TDC at the end of the compression stroke it moves from one side of the liner to the other one (in sectional view as shown in the previous picture) and stays on that side for the entire expansion stroke. When it approaches the TDC it rotates slightly and this movement is called “tilting”. This kind of movements is greater during the warm-up phase of the engine, when the clearances are greater and can be an important source of noise [13]. To reduce the negative effect of the tilting movement of the piston an offset-pin configuration is normally employed; with a pin located centrally on the cylinder axis the tilt movement of the piston happens exactly at the top dead center where high pressure level are reached in the chamber: this is not optimal for the combustion because the combustion chamber is not the ideal one and the correct flame propagation is impeded. If the tilt movement verifies at TDC, wear and noise are great due to the high pressure. Offsetting the pin from the central position anticipates the tilting movement of a few crankangle degrees (1° or 2° are common values), so that it happens before the crankangle at which maximum pressure levels are recorded, thus lower noise and wear can be reached. The chamber is also better sealed during the combustion, so lower blow-by gases are generated.

The piston slap movements have an important effect on the wear of the engine because the piston continually impacts with the liner, wearing it. Thus the piston-to-liner clearance increases with the age of the engine and so also the blow-by flow.

The forces involved in the slap movement are responsible for the wear. They are:

- Cylinder axial force F;
- Gas force F_g (net force considering the difference between the chamber pressure and the crankcase pressure multiplied by the cylinder surface);
- Liner reaction force F_l;
- Thrust force F_n.

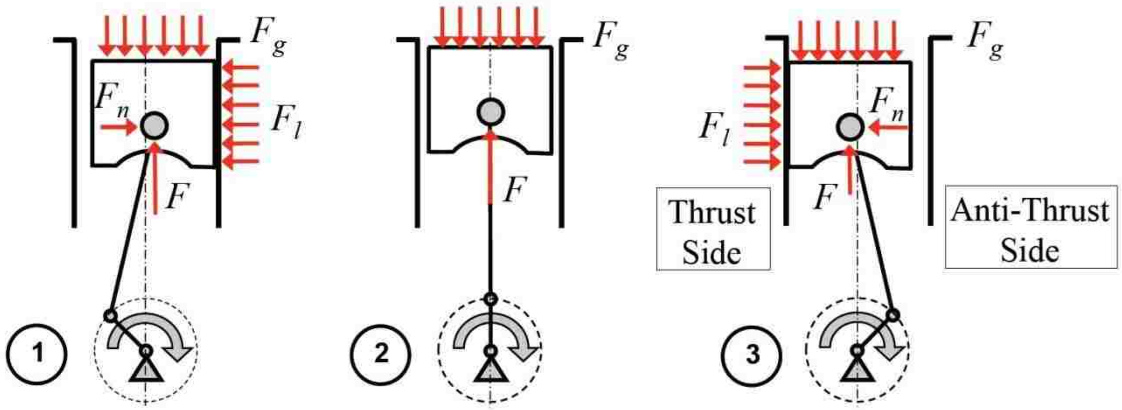

Figure 2.3-3: Forces involved in the piston movement [8].

Considering the equilibrium it is possible to write (not considering the inertia of the piston):

$$F_g = (p_{chamber} - p_{crankcase}) \cdot A \quad (2.1)$$

$$F_g = -F \quad (2.2)$$

$$F_l = -F_n \quad (2.3)$$

The friction force F_f is the one that actually produces wear of the liner and can be computed as:

$$F_f = \mu_d \cdot F_n \quad (2.4)$$

where μ_d is the dynamic coefficient of friction.

The friction is strictly related to the pressure inside the cylinder and so does the wear: this means that the majority of the liner wear occurs during the expansion stroke where the gas force is greater, while it is lower during the other phases. For this reason one side of the liner (viewing it in sectional view as in Figure 2.3-3) has to face a higher friction compared to the other; it is called “**thrust side**” and is loaded during the intake and the expansion stroke, while the “**anti-thrust side**” is loaded during the compression and the exhaust stroke and generally is subjected to lower wear.

2.4 Pistons rings

The piston rings are the elements which mainly limit the blow-by production. According to [13], the main roles of the rings are:

- sealing of the combustion chamber;
- guarantee the correct heat transfer from the piston to the cylinder wall;
- control of the lubricating flow of oil.

The rings are usually built in fine grain alloy cast iron, in order to guarantee excellent heat transfer and wear resistance properties.

Figure 2.4-1: Piston rings [8].

The number of rings is strictly related to the type of engine and its average rotational speed: according to [1], the relation between the mass of charge which leaks in the crankcase and the total charge mass is:

$$\frac{\dot{m}_{leak}}{\dot{m}_{charge}} \; \alpha \; \frac{1}{n} \tag{2.5}$$

This means that for high revving engines the percentage of leakages is lower compared to the one of low speed engines; this can be explained with the lower time available for the charge to escape the sealing. This is the reason while small 2 stroke engines employ only 1 ring while very big stationary engines, with high compression ratio and low rotational speed have many more (up to 10 rings per piston).

Usually the leaked charge is less than one per cent of the total charge in the cylinder for engines of normal cars. It is preferable to limit the number of piston rings, because they are a source of friction and they contribute to the total piston-assembly reciprocating mass and so to the reciprocating forces which are a source of vibration. As a rule of thumb, 2/3 of the total friction in an engine is due to the pistons and rings and 2/3 of this one is due to the piston rings [13].

The leakages are generated due to the non-perfect sealing of the rings which are not closed, but open in order to make the assembly process easier. They are inserted in specially machined grooves inside the piston skirt.

In common road car engines, the number of rings per pistons is usually 3: they constitute the so called piston rings pack. The first and the second ring are compression rings and have to impede the blow-by flow, while the third one is the oil control ring which has to control the oil scraping the cylinder wall (also the other two rings accomplish partly to this task).

The sealing is made by contact of the piston ring external side and the cylinder liner (separated by a thin film of oil) and by the contact of the top or bottom sides of the ring with the bottom or top faces of the ring groove, depending on the direction of motion of the piston. The forces which are responsible for this contact are the spring force of the ring, the pressure force exerted by the gas around the ring and the inertia force of the mass of the ring itself.

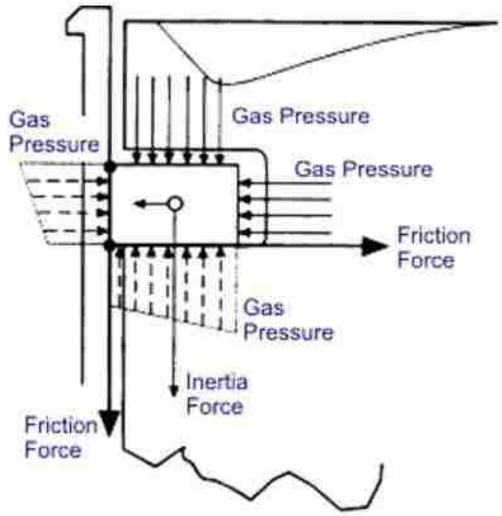

Figure 2.4-2: Piston ring sealing mechanism [14].

During a compression stroke the piston is moving upward and the gas pressure in the combustion chamber is increasing: this means that both the gas pressure and the inertia of the ring keep it in contact with the bottom part of the groove, while the gas pressure also pushes the ring against the cylinder wall together with the spring force of the ring.

The sealing is not perfect because when the piston change its direction the rings have to move from one side of the groove to the other and a small passage opens, letting the gas flowing in the crankcase; the process is shown in Figure 2.4-3.

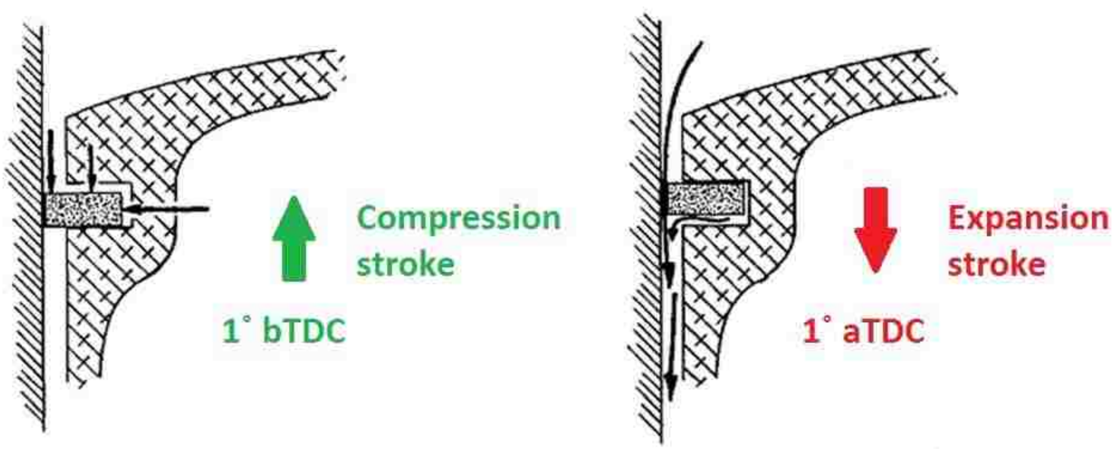

Figure 2.4-3: Sealing breaking (adapted form [15]).

The dimension of the available passage influences the amount of blow-by gases: it is related to the piston ring axial clearance which is the space available to the ring for the movement in axial direction inside the groove and to the piston ring radial clearance, which is the space between the internal side of the groove and the internal face of the ring. According to [16], the major effect on the blow-by amount is the radial clearance of the first ring; the blow-by flow increases increasing the clearance so it is important to minimize this clearance and minimize the wear of liner and ring, which would increase this value. The piston ring axial clearance instead has a minor effect on the blow-by [16]. When the engine is old and worn the clearances are bigger and this is the reason why a worn engine has blow-by flow several times greater than a new one.

The first ring, referring to a common ring pack of three rings, is called "**top ring**" or "**fire ring**" and is the closest to the combustion chamber. Its main purpose is to seal the chamber. It is the ring subjected to the highest pressure and temperature stress because it is the closest to the combustion chamber ; the high pressure in the combustion chamber pushes the ring against the cylinder wall making a very good sealing possible. It usually has a simple barrel shape because a too complicated one will result in high wear and deformation. The current practice is to put the first ring as close as possible to the top of the piston, in order to reduce the crevice regions and so the unburned hydrocarbon emissions of the engine.

The second ring is the "**scraper ring**": it has to guarantee the sealing but also has to scrape the oil from the cylinder wall, making a double function. It has a more complicated shape respect to the top ring because it is subject to lower pressures and temperatures; the most common type of scraper is the one with reverse twist shape; this shape is particularly useful in order to remove the oil from the liner when the piston is moving from TDC to BDC; the contact pressure with the cylinder wall is great due to the small contact area: this is good because the gas side pressure is low and a great contact area will not guarantee a good sealing.

The last ring is the "**oil control ring**" which has the purpose of removing the excess of oil which is not used for lubrication; in this way, only the quantity of oil required for the lubrication is used and the oil consumption is reduced. It is the farthest ring from the combustion chamber and so it is subjected to low gas pressure and usually has a spring to push it against the wall. Many different shapes are possible, all aiming to reach the trade-off between sealing and friction, and it can be made of three different pieces [13].

The oil for the lubrication of the cylinder wall is recovered in the inner part of the piston after being scraped, thanks to a series of small holes in the groove of the oil control ring: in this way the oil can reach again the crankcase.

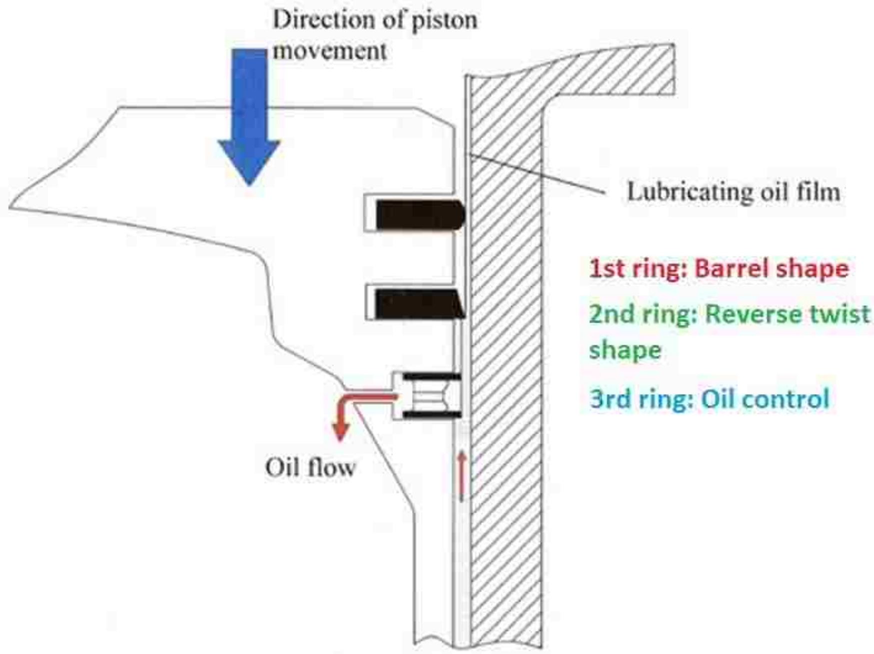

Figure 2.4-4: Piston ring shape [17].

The design of piston rings has to match different requirements: they should be as light as possible to reduce the reciprocating masses and should be narrow so that the contact pressure with the wall guarantees a good sealing. In any case, this will result in high friction and wear.

The dimension of the rings should also be chosen regarding the heat transfer. They should not be too small because they also have to transfer the heat from the piston to the cylinder wall: according to [17], the piston rings have to conduct around three quarters of the heat that the piston exchanges with the cylinder wall and so a trade-off between lightness and heat transfer properties must be reached in the design phase.

Finally, the rings surface is often treated with coatings in order to increase their wear resistance [13].

2.5 Cylinder wall – liner

The engine block can be designed with liners or without them: the difference between the two concepts is in the fact that in the first case the liner is physically present in the engine block, while in the second case the liner is created inside the engine block and not added later. The second solution is usually employed in cars engine, while for trucks and heavy duty diesel engines the first solution is used.

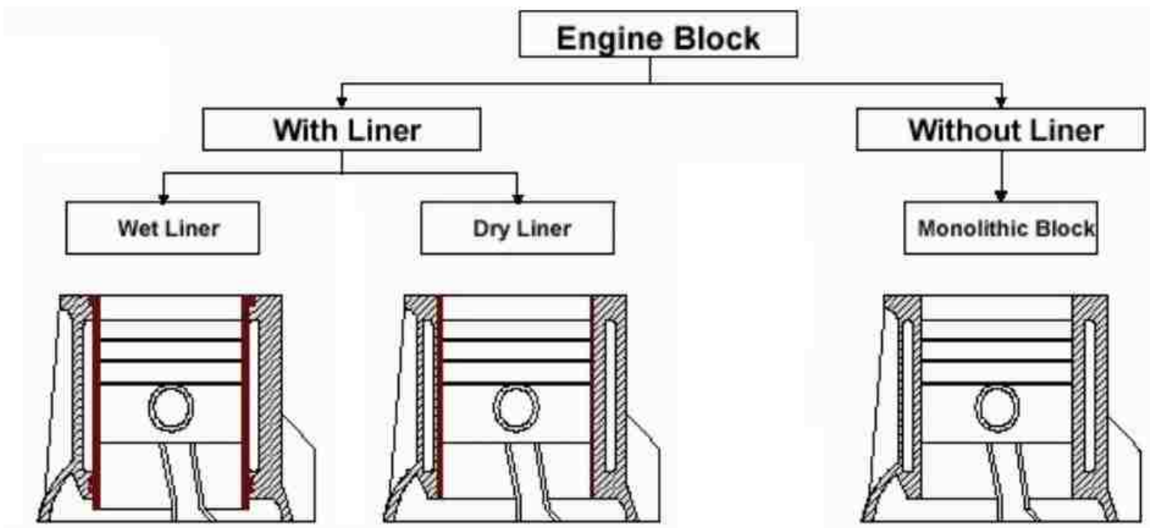

Figure 2.5-1: Cylinder liner design (adapted from [8]).

In engine blocks with the liner physically present it can be directly in contact with the cooling liquid (and it is called "wet liner") or not ("dry liner"); the wet liner has usually better cooling performance because the cooling fluid is closer to the cylinder but presents greater distortion problems respect to the dry solution. The liner deforms due to heat, piston forces and screw forces which connect the cylinder head and the engine block. The liners should deform as little as possible, because this would increase the area available to the blow-by gases; the deformation also influences oil consumption and wear [18].
The cylinder liner, when present, is usually made of Al-alloy.
The most important part of the cylinder liner is the internal surface: it is designed to have low friction with the piston-assembly and to have high abrasion resistance. The internal surface is designed to keep a thin layer of oil on the liner that improves the lubrication. This surface should not be completely flat and polished but it is intentionally made rough,

otherwise the oil slips away and the contact is too dry; a "**plateau finishing**" creates signs on the wall which behave as pockets for the lubricating oil which adheres to the wall [8]. The roughness created in this way is showed in the picture below.

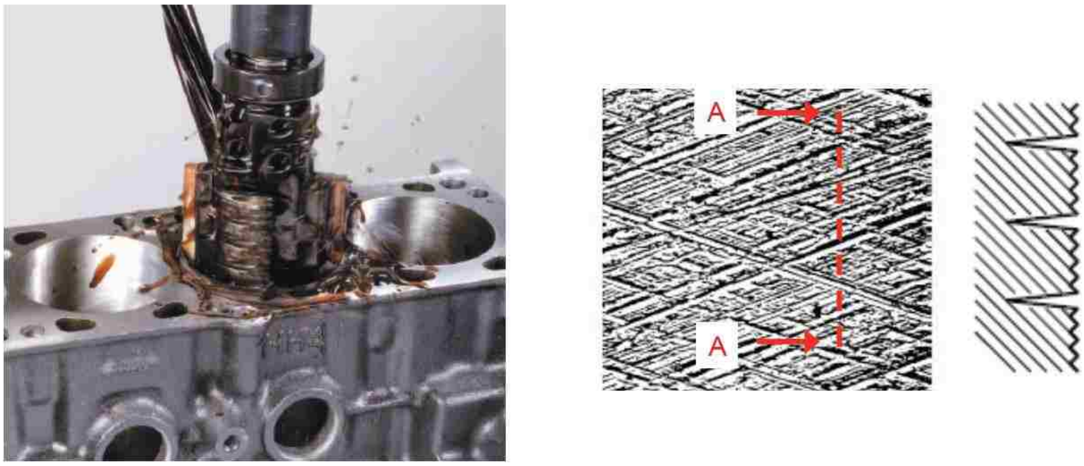

Figure 2.5-2: Plateau finishing liner treatment [8].

Coatings are also applied on the internal surface to improve its characteristics.

The liner wears out increasing the engine mileage. The wear is function of many parameters; the most important are engine design, fuel and lubricant oil used, and operating conditions [18]. The part of the liner subjected to the greatest wear is the portion closer to the combustion chamber: here the hydrodynamic lubrication is lower due to the lower relative velocity between the parts; as a consequence friction increases and so the wear does the same.

2.5.1 Wear

The wear in the liner has a great influence on the blow-by production; the wear increases the volume available for the gases to escape the sealing of the piston rings. The radial clearance of a worn engine is greater compared to a new engine, so the gas leakages are greater and this means lower engine performance (the escaping flow does not produce useful work), oil contamination in the crankcase and greater oil consumption. For these reasons an engine is commonly considered worn when the oil consumption and blow-by production increase too much [16].

The cylinder wear is most likely to occur close to the TDC or the BDC where the quality of lubrication decreases; for the majority of the piston stroke the piston skirt – cylinder wall contact is lubricated hydro dynamically. **Hydrodynamic lubrication** is possible when a fine oil film separates the 2 components which are moving with relative velocity; this relative velocity creates the oil film, provided that the sliding velocity is high enough: when the velocity decreases, the thickness of the oil film decreases as well, until the two sliding surfaces go in contact and the friction coefficient increases as a consequence. This second type of lubrication is called **boundary lubrication** and usually happens in piston assembly when the sliding velocity is small and the oil film cannot be sustained, so when the piston is approaching the Dead Centers. As said before, the coefficient of friction is greater for a boundary lubrication type and this explains the reason for a greater cylinder liner wear close to the BDC and TDC. In particular, the region close to the TDC is the one which presents greater problems; here the high pressures and temperatures due to combustion decrease the oil properties which are also deteriorated by combustion product which mix with the oil film during the combustion phase, making good lubrication even more difficult to be achieved [18].

The wear is a function of many parameters: the first and most important is the mileage of the engine but also the composition and quality of the oil and fuel play an important role. An important oil parameter which influences wear is its viscosity; a greater viscosity allows the oil film to adhere to the liner also at low relative speed, shifting the transition from hydrodynamic to boundary lubrication. In a compression stroke, if more viscous oil is adopted, the transition from the mid-stroke hydrodynamic lubrication to the boundary lubrication typical of the upper part of the stroke happens closer to the TDC, thus a lower part of the stroke is covered in metal-to-metal contact. In any case, a too great oil viscosity means greater effort for the pump and a general reduction in fuel economy especially in cold start. The common oil employed in current cars are “multigrade-oils” which perform differently according to the outside temperature so that the oil presents low viscosity for cold start at low temperature and becomes thicker at higher temperature. Also the fuel has an important role; the sulfur content in the fuel has a negative effect on the wear because during combustion sulfur acid is formed and absorbed by the oil film and it causes corrosive wear of the liner [18]. In any case, current available fuels present low sulfur content so this negative effect is reduced compared to past years. For example, in diesel fuel, the content of sulfur has been reduced in Europe from 350ppm in 2000 to 10 ppm in 2009 , while in the US it was reduced from 500 ppm in 2002 to 15 ppm in 2010 [13].

Also the widespread use of EGR has a negative effect because the sulphur-based acids generated during the combustion are not exhausted but recirculated [18].

The oil degradation is increased in today’s engines in which the combustion chamber’s pressure and temperature are very high, but the increased oil quality mitigates this negative effect.

Another pollutant of the lubricating oil is the soot, which accumulates in the oil film and causes mechanical abrasion of the liner [18]. The major problems are related to compression ignition engines which generally generate a greater quantity of soot compared to a spark ignition engine.

The use of wear resistant materials, the current trend toward the adoption of low-sulfur content fuels and the engine control strategies aimed to reduce the soot formation inside the combustion chamber limit the wear of the liner; this, together with the adoption of lubricant oil with better properties, allows oils with lower viscosity to be used, which reflects in a general improvement in the engine fuel economy.

2.6 Windage tray

The crankcase environment is very chaotic and there is motion of air, blow-by gases and oil mist; the flow field is generated by the motion of the pistons and connecting rods which reciprocate in the cylinders and the rotation of the crankshaft and counterweights [4]; a tray is commonly employed in order to avoid that this motion of gases aerates the oil. It is called **windage tray** and it is a simple sheet of metal which is directly connected with bolts to the engine block; it is shaped in a proper way in order to not come in contact with the rotating counterweights and basically separates the crankcase volume from the sump volume. The windage tray usually has holes in its structure in order to allow communication between the two volumes.

With the adoption of the windage tray the oil aeration decreases and also the quantity of oil mist in the crankcase is lower, thus the pumping losses are reduced because the crankcase fluid density is lower.

Figure 2.6-1: Windage Tray of Hemi 426 cubic inches of '69 Cuda [19].

2.7 Oil pan

The oil pan is the lower part of the engine and in most of the cases its function is as container for the lubricant oil of the engine. In the majority of the engines it is a simple sheet of steel properly shaped but it can also be built in die cast or shell cast aluminum alloy for complex shapes. It contains the oil which comes from the oildrains. The oil is then pumped to the lubrication galleries with an oil pump which can be directly located in the oil sump and usually is driven by the crankshaft with a chain. This solution is called wet sump because the sump is full of oil. The shape is very important because it should allow the exhaust pipe to reach the underfloor pipe in engines which mount a frontal exhaust system. The shape of the sump should make the oil change easy and fast and should impede that the oil level falls below a threshold under high transversal acceleration during corners; this could have a detrimental effect on lubrication if the oil pick up tube does not draw out oil. The oil pan can also be designed as a structural element to increase the bending stiffness of the powertrain [8]. Both the structural and NVH properties of the pan should be considered in the oil pan design, while it should be as light as possible.

Figure 2.7-1: Oil pan in steel sheet of FIAT 8V Fire engine [8].

Instead, race car engines and motorcycle engines adopt a dry sump solution. This system provides better performance respect to the wet solution. The oil is not held in the pan but in an external container and for this reason the sump is said dry; in this way there is no oil motion inside the engine under high accelerations during corners and the lubrication system always performs well independently on vehicle dynamic. This solution is wide common in motorcycles where the engine tilts with the motorcycle during cornering or in racecar which are subjected to high lateral forces.

Figure 2.7-2: Dry Sump with scavenge pump from Dailey Engineering for Mopar 360 [20].

The oil which comes from the oildrains is sucked up by a vacuum pump, directly driven by the crankshaft, which maintains the crankcase volume under the atmospheric pressure; this has a positive effect on the engine efficiency because the crankcase pumping losses are reduced due to lower pressure involved and lower crankcase fluid density.

The pan simply closes the bottom part of the engine; the shape is much simpler and the vertical height is lower so the engine can be installed in a lower position, improving the handling of the car. Also the oil reservoir can be positioned in a suitable location to improve the overall weight distribution; the pressure pump, which has to pump the oil in the lubrication system and is usually in a lower position respect to the reservoir, performs better due to this positive pressure differential. The external reservoir allows a greater oil quantity to be carried respect to the wet sump solution, where all the lubricating oil is held in the pan, thus the thermal management of the lubricant is easier and the cooling performance is greater.

Finally, the low position of the scavenge pump allows the gravity force to be exploited for scavenging the crankcase.

The disadvantages of this solution are the energy required to drive the vacuum pump and the major costs of the system.

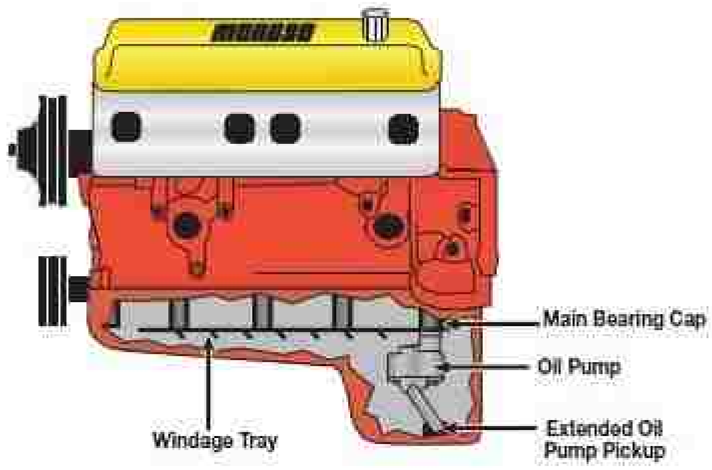

Figure 2.7-3: Moroso Wet sump system. [21]

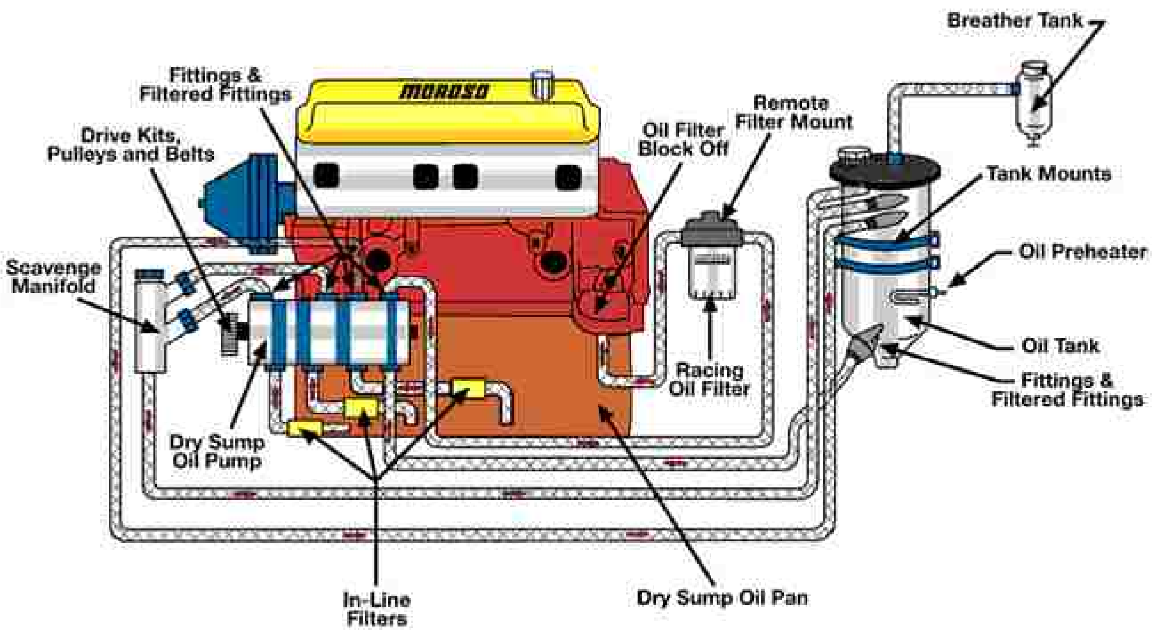

Figure 2.7-4: Moroso dry sump system [22].

2.8 Oil separator

The oil separator is a very important element for the system: the task is to recover as much as possible of the oil mist which is mixed with the blow-by gases in order to reduce the oil consumption of the engine and avoid the burning of oil in the combustion chamber and the generation of high pollutant emissions. If not filtered, these particles can deposit in the intake line, reducing the overall engine performance; the engine components which are more affected are:

- Turbocharger: turbocharged engines are always more common due to the high specific power. The turbine can reach high rotational speed around 180,000 – 200,000 rpm; the clearance between the turbine and the turbine case is always very small and also small deposits of soot can reduce the efficiency of the turbine, thus reducing the engine performance.
- Intercooler: oil particles can accumulate on the inner surface area of the intercooler, creating an insulating layer; if deposits accumulates, the cooling efficiency is reduced and the air temperature at the output increase; warmer intake air means lower air mass inside the combustion chamber and so lower fuel can be burned, with low engine power output.
- Inlet valve: oil deposits in the inlet valves area can reduce the engine power, increase the fuel consumption and the pollutant emissions, due to the not perfect sealing of the valve covered with deposits [23].

A "selective separation" should be reached: ideally the oil must be completely recovered, while the fuel condensates, soot, air and water must be mixed with the intake air in the intake manifold.

The oil separator is a service life component; it should not be replaced and should last for the whole vehicle life.

The efficiency of an oil separator is the ratio between the number of the filtered oil particles and the total particle number. The separation efficiency is function of the average oil droplet size in the flow, which in turn is function of the engine working conditions: for this reason, the separation efficiency changes during engine operation: the target for the design and choice of a separator is to reach the highest separation efficiency close to the engine working condition which most frequently occurs, like the highway driving.

According to [7], the oil particles with diameter smaller than 1 μm are called "micro-oil", while particles with diameter smaller than 10 μm are usually called "fine oil"; oil particles which are visible to naked eye are called “coarse oil”. On modern engines, there could be two type of separators, one for the fine-micro oil particles and one for the coarse particles; the fine oil separator does not properly work if it is invested with coarse oil and so a coarse separator, as a baffle separator, is employed in front of the fine oil separator to filter the oil particles with greater size, letting the smallest one to be filtered by the fine oil separator [7].

Higher separation efficiency can be obtained increasing the pressure drop through the input and output of the separator; due to the always smaller pressure difference between crankcase and intake manifold, an even smaller pressure drop is available for filtration purpose in modern engine. A trade-off between filtration efficiency and pressure drop should be found in the design stage. The main problem in modern separators concerns the micro-oil particles, which are difficult to be filtered.

One approach to increment the separation efficiency is to adopt a blower to increase the pressure drop available for the separator; this approach is called “active crankcase ventilation” [9] and it is a highly expensive and complex system.

Many technologies can be exploited for the oil separator; as said before, for the coarse particles, a baffle type separator is commonly used: the baffles immersed in the flow simply prevent surge oil from the crankcase (which can happen in some engine working conditions) to invest the fine oil separator, damaging it and blocking the flow of gases. Another separator, instead, filters the fine oil particles [7]. For this kind of separator there are many options, which provide different separation efficiencies for different costs.

Only one separator is used for cost reasons in the majority of the applications.

The most common types of oil separator used are described in the following paragraphs.

2.8.1 Volume separator

This type of separator exploits the inertia of the oil particles: it is the simplest and cheapest solution but does not guarantee high separation efficiency. It is based on the principle that when a flow slows down, the oil particles cannot be suspended and falls down under the gravity force. As a rule of thumb, if the flow speed decreases under 1 m/s the oil droplets will fall out [11]. The functioning principle of this separator is based on the following equation:

$$\dot{Q} = A \cdot v \tag{2.6}$$

For a given flow, if the section increases then the velocity decreases; in a hose with a greater section the flow speed decreases, thus separation occurs and the oil is drained in the oil pan. The volume separation principle is shown in Figure 2.8-1.

The problem with this type of separator is the size of the system which is hardly achievable under the hood of modern cars , where downsizing is the key priority; in any case, in engine with camshafts driven by chain, the chain enclosure can be designed as a large communication passage between the crankcase and the cylinder head in order to scavenge the crankcase; the chain case can be designed in order to have a section great enough to slow down the flow, increasing the separation efficiency with an acceptable system size.

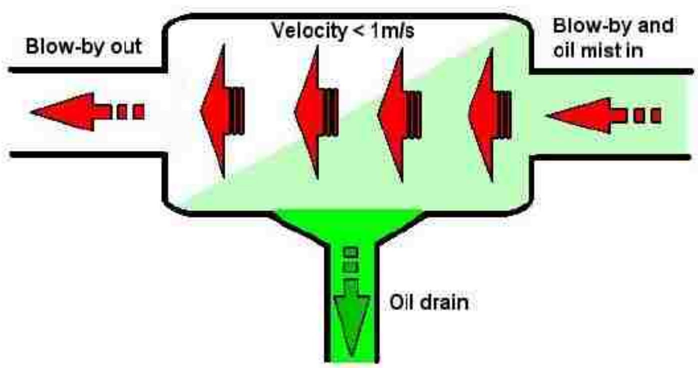

Figure 2.8-1: Volume separator [11].

2.8.2 Labyrinth separator

This type of separator exploits the inertia of the oil droplets inside the flow. The oil has a greater density with respect to the gas and so it has more inertia with respect to the gas in the flow; in this type of separator the labyrinth forces the flow to change repeatedly direction through a maze of strict channels. Separation between oil and flow is possible due to the fact that the ratio between the density of the oil and the density of the gas is very high; the droplets are not able to change direction as fast as the gas and impact upon the wall. At the same time, when the flow changes direction, it also slows down and it is not able to keep the oil droplets in suspension; the gravity takes over at low speed and the oil falls out; separation occurs and the oil is drained through the oildrains into the oil pan [11].

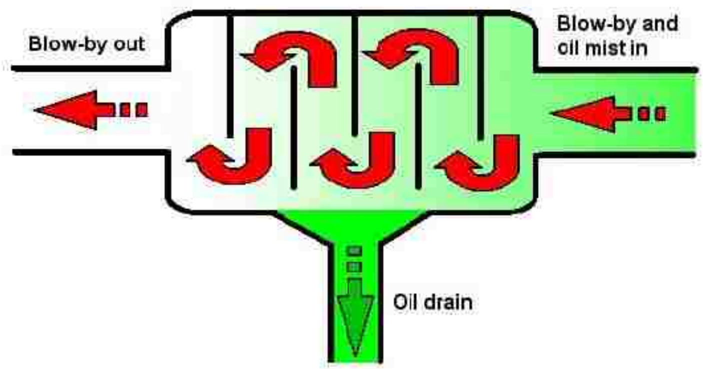

Figure 2.8-2: Labyrinth separator [11].

2.8.3 Centrifuge separator

This is another type of separator that works exploiting the inertia of the flow; the inertia is generated by the centrifugal force which acts on the flow spinning in a cylindrical chamber. The motion of the flow can be generated by an active system, like an electric motor, or passively, simply directing the flow to tangentially enter the chamber. The idea is to separate the oil droplets from the gas flow under the effect of the centrifugal force, exploiting the greater inertia of the liquid respect to the gas.

If an electric motor is exploited to move the fluid, a greater centrifugal force is generated and better separation is performed. In any case, the complexity of adding an electric motor only for separation purpose makes this solution almost totally impractical for a road car engine: this approach can be used for large engines with high blow-by flow, like truck engines.

Alternatively, the oil separator can be guided by the camshaft: this solution is exploited in the Chrysler Pentastar V6 engines.

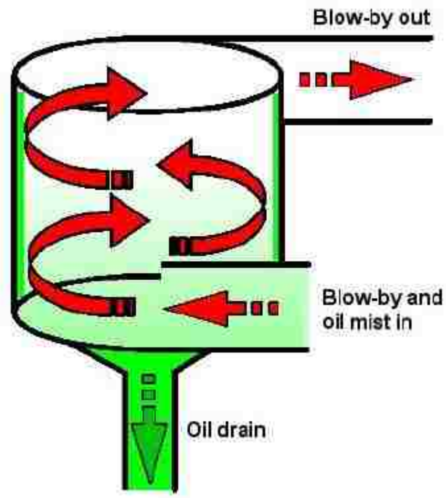

Figure 2.8-3: Centrifuge separator [11].

2.8.4 Impactor separator

The last separator described in the present work is the impactor separator: also this one is an inertial separator. With reference to Figure 2.8-4, the inlet flow is deflected and forced to pass through the small nozzles on the top of the separator; here the flow is accelerated due to the restricted section and then it impacts against a filtering material collecting plate, where the oil particles are separated due to their greater inertia and cannot follow the gas flow.

The special developed filtering material improves the agglomeration of small droplets in bigger droplets so that they can be drained back in the oil pan, while the clean gas is vented in the intake manifold. The large size droplets are no more affected by the flow due to their greater weight.

The filtering material is a specifically designed coalescing material to improve agglomeration; on the top of the system there is also a by-pass valve which opens when the pressures inside the crankcase reaches a too high value, allowing a greater area for the flow: also the by-pass flow is filtered by the coalescing medium. According to [24] the impactor separator is the most efficient one compared to the other separator types.

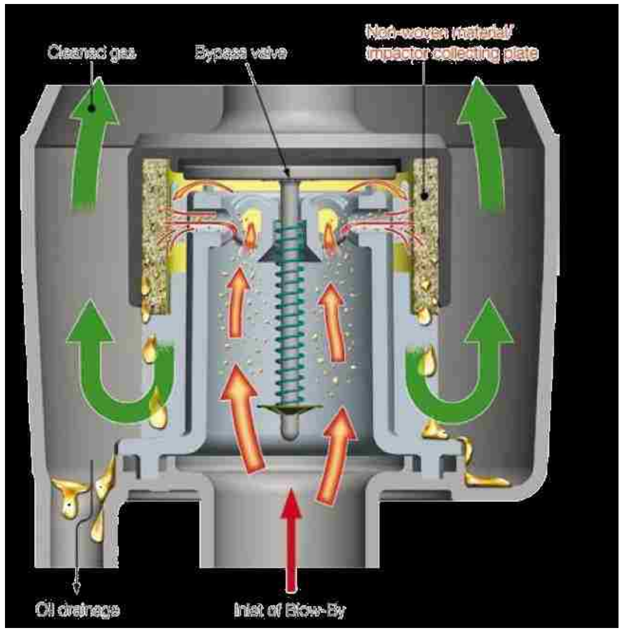

Figure 2.8-4: Impactor separator [24].

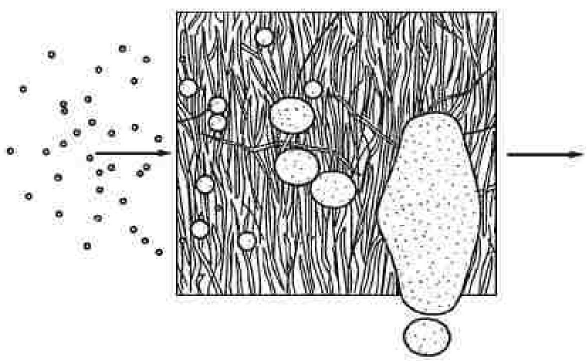

Figure 2.8-5: Coalescing filter material [5].

2.9 PCV valve

The abbreviation PCV can stand both for “positive crankcase ventilation” or “pressure control valve”. As mentioned before, the main target of this valve is to keep the pressure level inside the crankcase as constant as possible in the different engine working conditions. The differential pressure between the crankcase and the intake manifold is used to evacuate the gasses from the crankcase; if it is too low there is not enough energy to evacuate them, while if it is too high it means that the pressure level inside the crankcase can be too great and this increases the pumping work inside the crankcase.

The second case is not a big problem because an increment of the section area of the valve increases the available flow and so the pressure inside the crankcase decreases; the first case is more complicated because it is always more frequent in modern engines: today's trend is to apply turbocharging and direct injection, which both involve a reduction in the throttling of the engine, so a lower intake manifold vacuum can be exploited to clean the crankcase.

The PCV valve is usually put on the cam cover, after the oil separator, and it enables a different gas mass to flow, according to the differential pressure across it. The flow is usually released close to the throttle for a normally aspirated engine, where high turbulence and low pressure is available, or before the compressor in a supercharged engine; the discharge point in the intake manifold is usually in a low pressure region after the throttle.

It is important to note that the PCV valve is only a one-way valve so it only permits a mono directional flow, from the crankcase to the intake manifold.

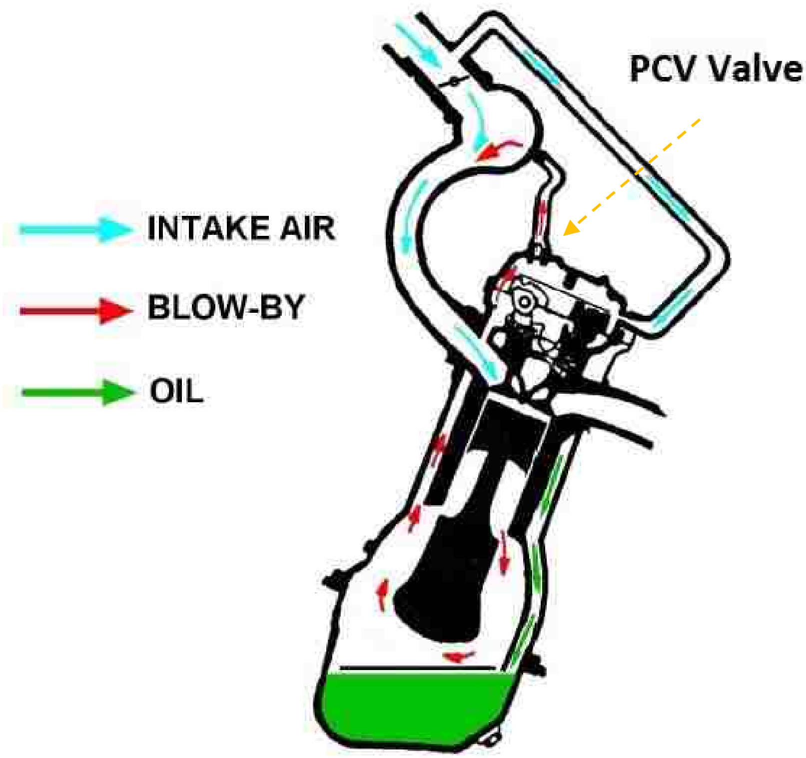

Figure 2.9-1: PCV valve on the cam cover [11].

Historically, the PCV valve introduction was preceded by simply connecting the crankcase to the manifold with fixed orifices, mounted over the valve cover: though this system is simple and cheap it does not scavenge the crankcase in every engine working conditions. The reason is because the blow-by production of the engine does not match the intake manifold vacuum: for example, at low load the intake manifold pressure is lower than the atmospheric pressure (this is usually called “engine vacuum” [6]), while the blow-by production is low due to low combustion chamber pressure; the great vacuum available is likely to decrease too much the pressure in the crankcase, generating problems of seal integrity in the oil pan as too high or too low pressure can blow out the engine seals. Moreover, it is necessary to maintain lower blow-by flow rate in the intake manifold at idle conditions for idle control.

At the same way, at high load the blow-by production is high but the engine vacuum is very low due to the completely open throttle, and so it is difficult to scavenge the crankcase. For this reason, the solution with fixed orifices has been substituted with a PCV valve, which allows accounting for the different blow-by production in the different engine working conditions.

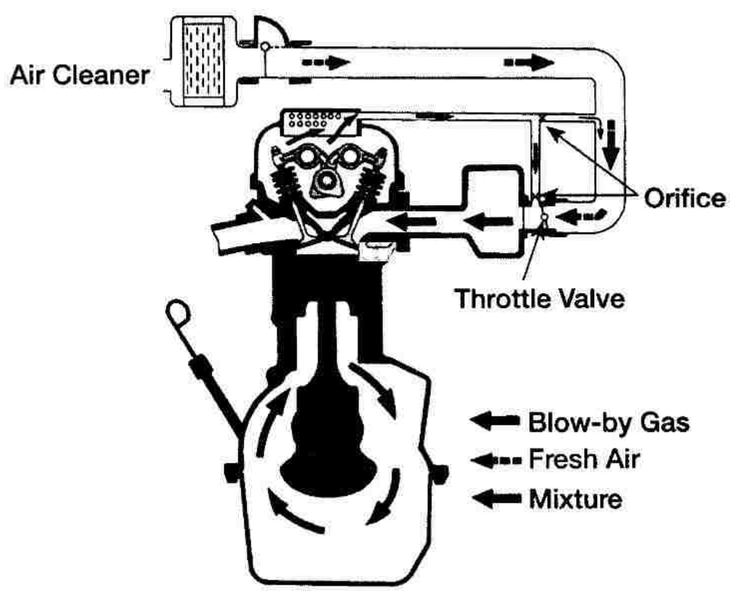

Figure 2.9-2: PCV system with fixed orifices [25].

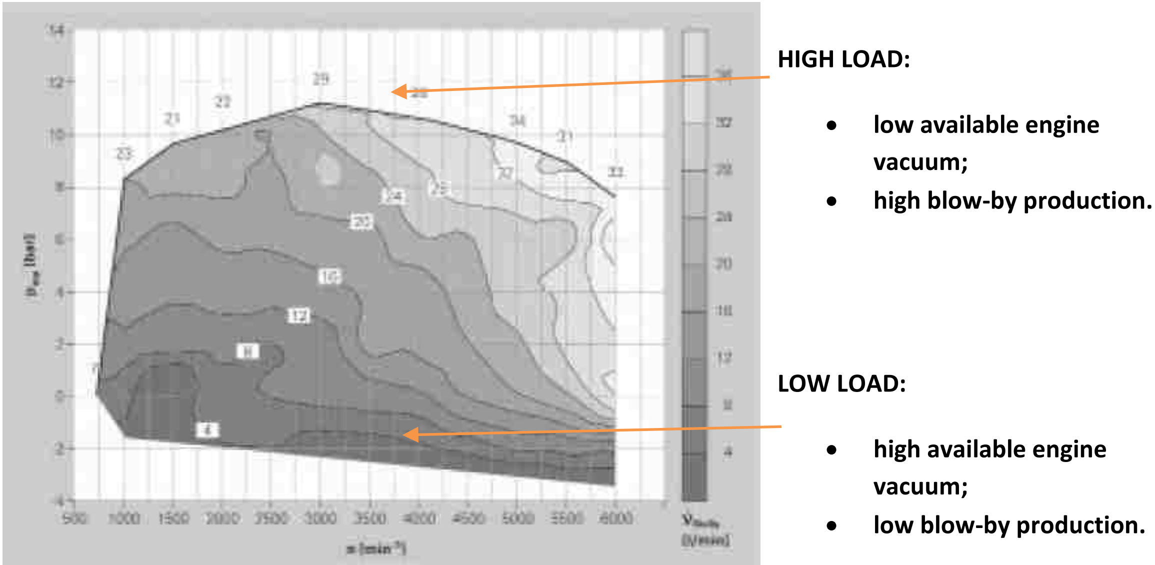

Figure 2.9-3: Mechanical characteristic of a SI engine. The differential pressure between intake manifold and the atmospheric pressure is showed [7].

The PCV valve should not freeze in the winter due to the freezing of the vapor of water in the crankcase and should have an unlimited resistance to the blow-by gases so that it does not block the whole ventilation system. The structure of a common pressure control valve is represented in Figure 2.9-4.

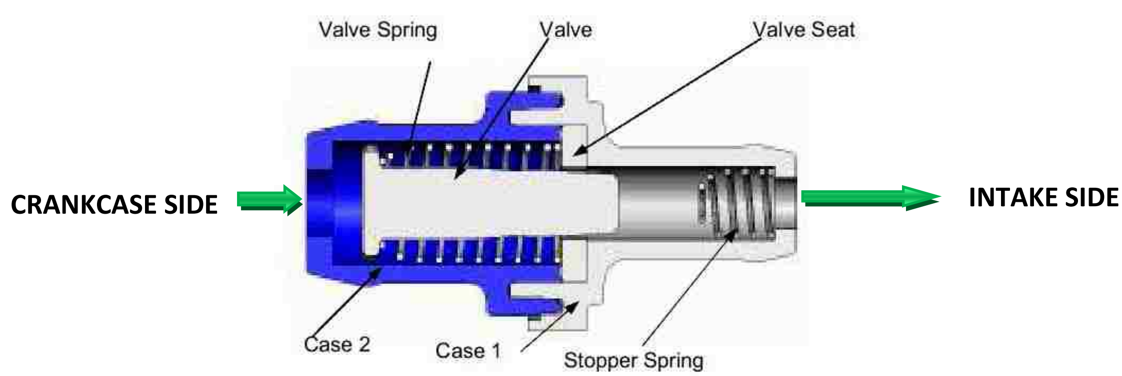

Figure 2.9-4: PCV valve structure [10].

Two springs are present: the first one guides the valve inside the case, while the second is the stopper spring which stops the valve stroke. The valve has a shape so that it decreases its shape toward its extreme: this enables the available flow area to be changed as function of the differential pressure between the two ends.

The functioning principle of the system is shown in Figure 2.9-5, Figure 2.9-6, Figure 2.9-7 and Figure 2.9-8.

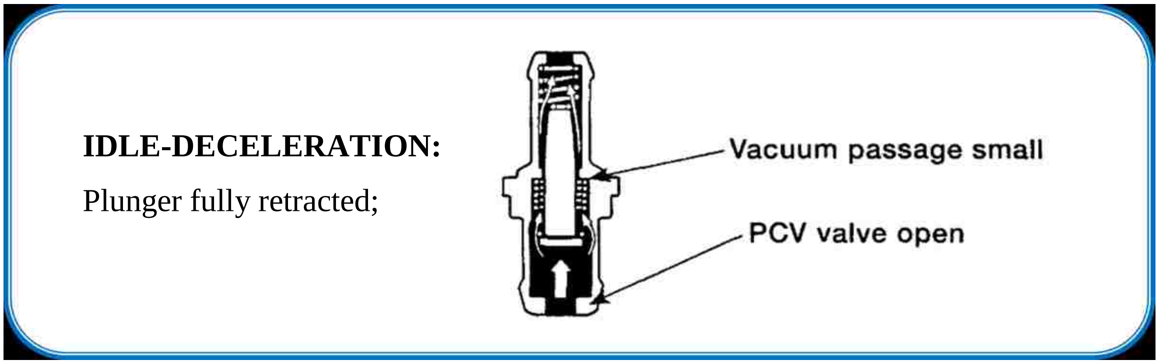

Figure 2.9-5: PCV valve operation: idle-deceleration [25].

During the idle condition the engine vacuum is very high and so the differential pressure is high; the plunger compress the spring under the effect of the pressure and so the available area is small, leading to a small blow-by flow.

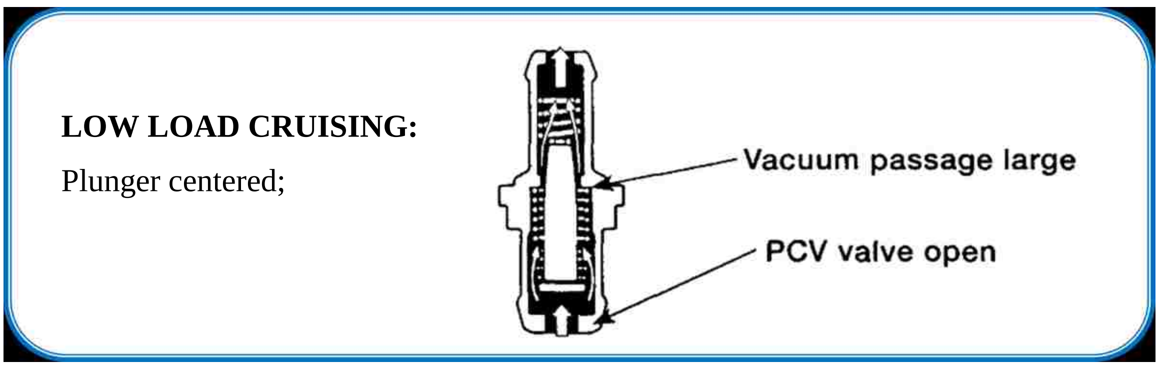

Figure 2.9-6: PCV valve operation: low load cruising [25].

During the low load condition lower differential pressure is available respect to the previous case and the blow-by production is greater; the plunger moves down increasing the passage and so the flow.

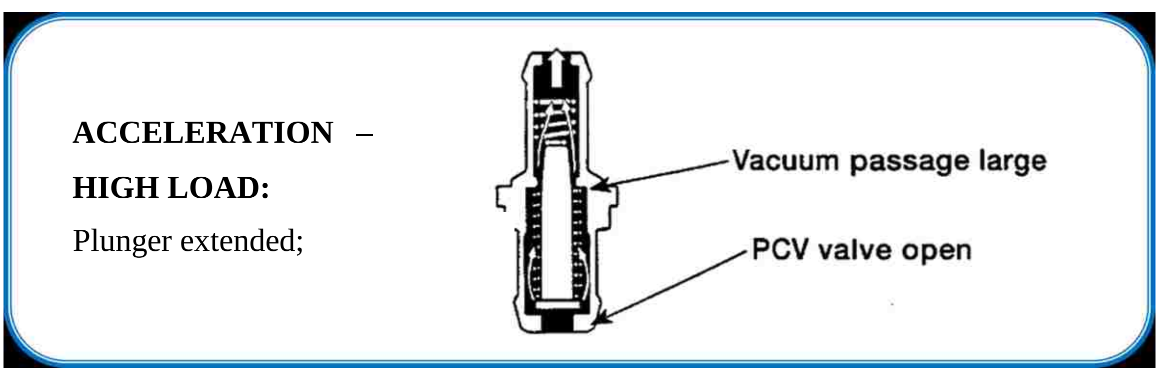

Figure 2.9-7: PCV valve operation: acceleration-high load [25].

During accelerations the blow-by production is huge and the engine vacuum is small due to the wide-open-throttle; the small differential pressure does not generate a high force on the plunger which is so pushed down by the spring until a balance is reached. The flow area is maximum in order to evacuate the great amount of blow-by.

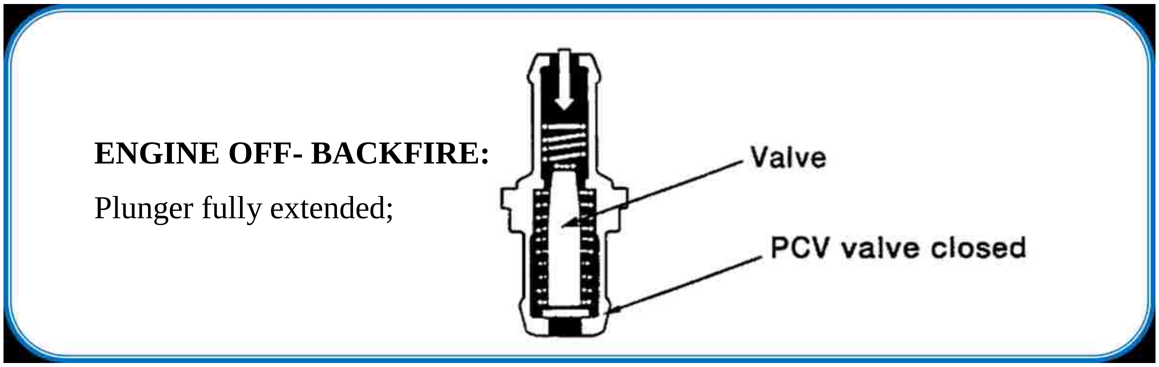

Figure 2.9-8: PCV valve operation: engine off-backfire [25].

When the engine is off, no differential pressure is present between crankcase and intake manifold: only the force of the spring acts against the plunger. The spring is fully extended and the plunger completely closes the valve intake, so that no leakages or air from the intake manifold can fill the crankcase. This situation impedes the stagnation of vapor of water in the crankcase when the engine is off, thus reducing the risk of rust and ice during the winter in the crankcase. Also in the event of backfire, the closed valve impedes any flame to diffuse in the crankcase.

The valve can be considered as a convergent nozzle: with reference to Figure 2.9-9, the section 2 is the restricted area which changes thanks to the plunger movement, where the flow is accelerated.

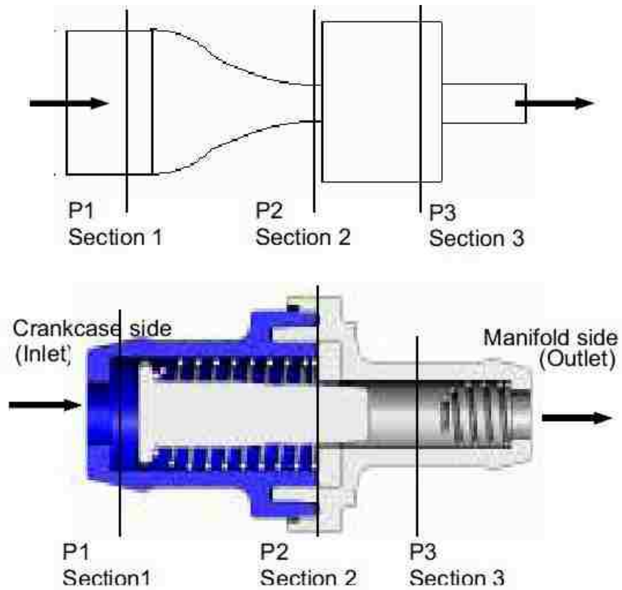

Figure 2.9-9: Comparison between a convergent nozzle and a PCV valve [10].

The flow through the PCV valve strongly depends on the pressure ratio between the pressure in the restriction (section 2) and the pressure in the crankcase side (section 1).

The flow is accelerated due to the restriction and sonic condition is often reached; in this case, any further reduction of the manifold pressure will not increase the flow through the valve. The sonic condition is reached when:

$$\frac{p_2}{p_1} \leq 0.528 = critical\ pressure\ ratio \tag{2.7}$$

During working conditions where low engine vacuum is present, for example during an acceleration, the differential pressure between crankcase and manifold is small, so the pressure ratio between section 2 and section 1 is above the critical value.

When the engine vacuum increases, sonic condition can be reached and any further increase of differential pressure (due to further reduction of intake manifold pressure) will not increase the flow.

The choice of the PCV valve diameter should be made in order to guarantee the necessary flow in every working condition. The ventilation requirements are described by the blue line in Figure 2.9-10: usually the gas flow is greater during high pressure ratio (acceleration, high load) and decreases progressively decreasing the pressure ratio (throttling more and more the engine) until sonic state is reached in the restricted flow area, after which the flow keeps constant for any further pressure ratio reduction. In order to match these requirements, the plunger diameter should be small for high pressure ratio so that a greater restricted area enables a greater gas mass to flow. Inversely, the increment of plunger diameter leads to a reduction in the available area and so the flow is reduced; the valve diameter can be kept constant after the critical pressure ratio is reached in order to have a constant mass flow rate.

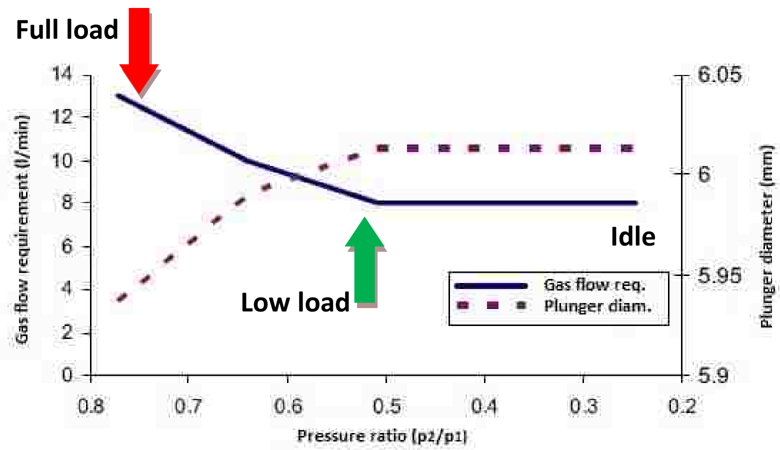

Figure 2.9-10: Gas flow requirement and plunger diameter as function of pressure ratio in a common PCV valve [10].

The numerical values in the previous figure are only indicative and general.

Typically, the relationship between the blow-by production of the engine and the characteristics of the PCV valve has the path described by Figure 2.9-11 [25].

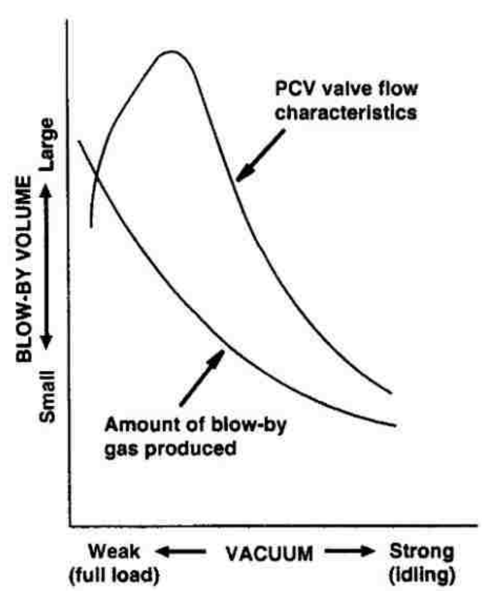

Figure 2.9-11: PCV valve flow characteristic with blow-by production characteristic [25].

The ideal valve characteristic perfectly matches the blow-by production; nevertheless, a crankcase ventilation system is often designed for 2-3 times the real blow-by production because it can increase a lot with the engine wear [24].

2.10 Windage inside the crankcase

The majority of road car engines adopts a **wet sump** solution: this means that the oil reservoir is built under the crankcase and here is collected and then sucked by the oil pump and sent to lubricate all engine components. The other possible solution is called **dry sump**, in which the oil reservoir is external to the engine: this solution is used mainly by motorcycle engines or high performance race engines.

Let's consider a wet sump engine: due to the motion of rotating elements inside the crankcase (like crankshaft, connecting rods, counterweights…) part of the oil adheres to these elements and create a sort of "**windage cloud**" surrounding the rotating assembly [26].

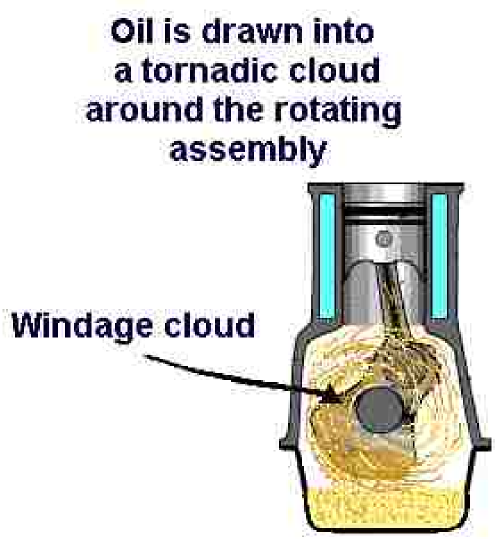

Figure 2.10-1: Windage cloud [26].

The intensity of this windage is directly related to the engine rotational speed and to the engine sump design: as engine speed increases, also the quantity of oil droplets inside the cloud increases. According to [26], the quantity of oil which can be suspended in the cloud due to the rotation at high speed of the crankshaft and counterweights can reach one quart. The motion of the gas in the crankcase has a negative effect on the oil in the sump: the gas interacts with splashing oil and, as a consequence, the oil is ventilated. Oil ventilation (or aeration) can decrease the oil properties, because gas bubbles can appear in the liquid oil and, when they reach the oil pump, they can cause cavitation and loss of head, blockage of the filter and loss of precision control (i.e. with the Multi-air system) [4]. This can also reduce the oil service life. In engine blocks where a short skirt solution is adopted the ventilated area over the oil level is large and so the oil aeration effect is great.

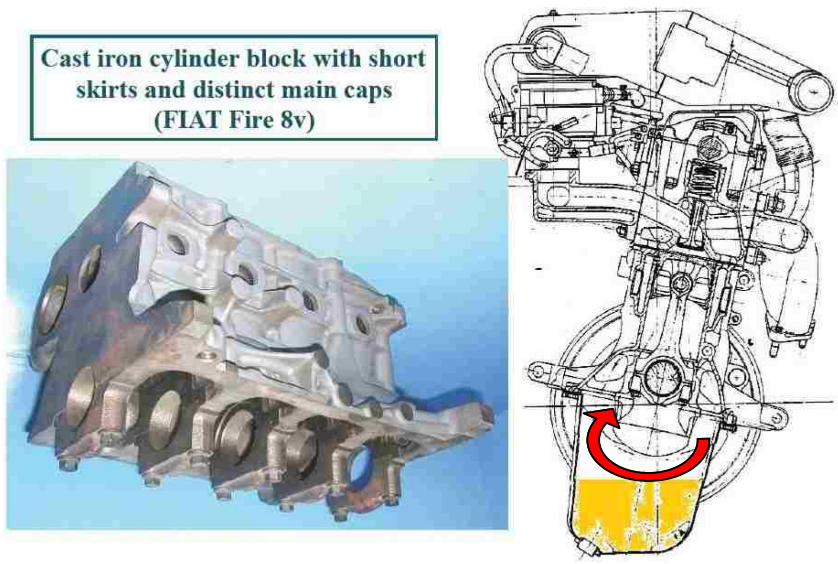

Figure 2.10-2: Oil aeration in a short skirt cylinder block [8].

In order to reduce the oil aeration the windage tray is usually employed: this simple sheet of metal which separates the crankcase from the oil sump acts as a physical barrier and is designed with holes so that communication is possible: the size of these holes is very important and is the trade-off between the need of low oil aeration and the need of pressure release. If the holes are too big the pressure inside the engine bay is reduced because a large amount of gas can flow in the sump volume but the oil foaming will be high; instead if the holes are too small the oil will not be aerated but the pressure level in the crankcase bay will rise up and pumping losses will increase [4]. The optimization between these two requirements is essential.

The performance of the engine can be improved by using a **crank scraper**, with the aim of reducing the motion of oil in the crankcase and so the parasitic drag which the crankshaft has to face. The scraper is a plate positioned between the engine block and the oil pan and it is fastened to the main bearing caps; it is specifically shaped in order not to come into contact with the rotating elements; the crank scraper physically removes the oil in the crankcase flow because it is positioned very close to the crankshaft without touching it.

The scraper is usually made of steel but there are also applications made of Teflon that allows the clearance between the scraper and the crankshaft to be smaller, close to 0.010'', [26].

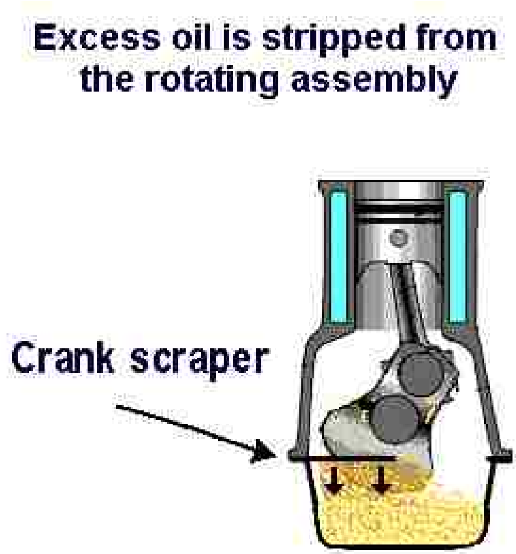

Figure 2.10-3: Crank scraper [26].

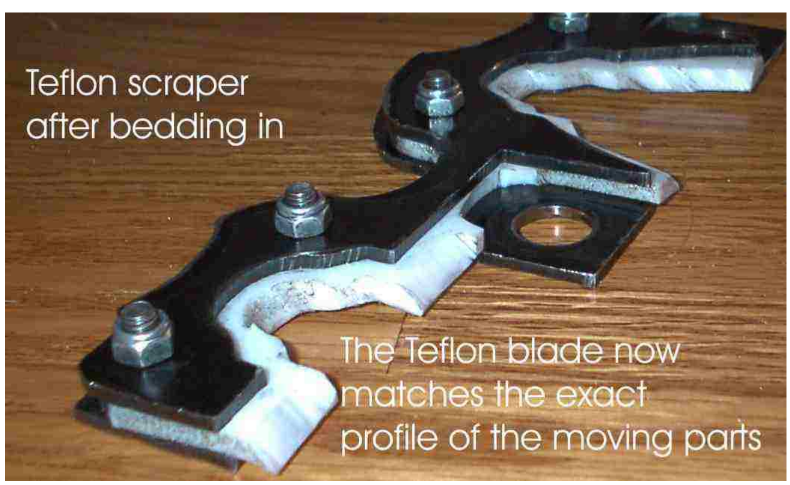

Figure 2.10-4: Crank scraper made of Teflon [26].

3 THEORETICAL DERIVATIONS

3.1 Blow-by flow computation

In this section a very simple method to predict the quantity of blow-by volume flow rate is presented. The evaluation of the blow-by volume flow rate is very important in the design stage, when engine designers have to design the size of the system for all engine working conditions, also keeping in mind the wear of the engine.

When the engine is worn the clearance between piston rings and cylinder liner increases, leading to an increment of the leakages of gases from the combustion chamber to the crankcase.

According to [7], the amount of blow-by flow corresponds approximately to 1-2% of the intake mass flow rate: defining the blow-by coefficient $k_{BLOW-BY}$ as:

$$k_{BLOW-BY} = \frac{blow - by\ volume\ flow\ rate}{intake\ air\ volume\ flow\ rate} \tag{3.1}$$

assume for a new engine

$$k_{BLOW-BY_{new}} = 0.5\% \tag{3.2}$$

while for a worn engine

$$k_{BLOW-BY_{worn}} = 1.5\% \tag{3.3}$$

For computing the blow-by volume flow rate it is necessary to compute the intake air volume flow rate; this is function of the displacement of the engine, of the volumetric efficiency, of the speed of the engine and of the type of the engine (2 stroke or 4 stroke engine). Supposing a 4 stroke engine with an engine displacement of 1600 cc., rotating at 6000 rpm with a volumetric efficiency of 0.8 the intake volume flow rate can be computed in the following way:

$$\dot{V}_{intake} = \lambda_v \cdot \frac{V \cdot n}{2} = 0.8 \cdot \frac{1.6\, l \cdot 6000\, rpm}{2} = 3840\ \frac{l}{min} \tag{3.4}$$

From this value the blow-by volume flow rate for a new engine can be computed as:

$$\dot{V}_{blow-by} = k_{BLOW-BY_{new}} \cdot \dot{V}_{intake} = 0.005 \cdot 3840 \frac{l}{min} = 19.2 \frac{l}{min} \tag{3.5}$$

and for a worn engine:

$$\dot{V}_{blow-by} = k_{BLOW-BY_{new}} \cdot \dot{V}_{intake} = 0.015 \cdot 3840 \frac{l}{min} = 57.6 \frac{l}{min} \tag{3.6}$$

Instead, if a turbocharged engine is considered, the intake air volume flow rate increases due to the turbocharged effect: assuming that the turbocharger does not influence the volumetric efficiency (this is not true, but it is just for this very simple computation) and a pressure ratio between output and input of the compressor of 1.75, the intake air volume flow rate can be computed as:

$$\dot{V}_{intake} = \frac{p_{turbo}}{p_{atm}} \cdot \lambda_v \cdot \frac{V \cdot n}{2} = 1.75 \cdot 0.8 \cdot \frac{1.6\, l \cdot 6000\, rpm}{2} = 6720 \ \frac{l}{min} \tag{3.7}$$

and the related blow-by volume flow rate for a new engine as:

$$\dot{V}_{blow-by} = k_{BLOW-BY_{worn}} \cdot \dot{V}_{intake} = 0.005 \cdot 6720 \frac{l}{min} = 33.6 \frac{l}{min} \tag{3.8}$$

while for a worn engine:

$$\dot{V}_{blow-by} = k_{BLOW-BY_{worn}} \cdot \dot{V}_{intake} = 0.015 \cdot 6720 \frac{l}{min} = 100.8 \frac{l}{min} \tag{3.9}$$

This is quite a big flow which has to be evacuated from the crankcase, otherwise the crankcase pressure will build-up to high values in short time; this will increase the crankcase pumping losses and the risk of leakages from the oil sump. These very simple computations give the designer a rough idea of what will be the flow to treat in the design stage of the crankcase ventilation system, also considering the deterioration of the performance of a worn engine.

3.2 Effect on engine performance

As said before, the crankcase pressure has an effect on engine performance. The pumping losses increase as the crankcase pressure increases, because greater crankcase pressure means greater crankcase gas density, so the rotation of the crankshaft is hampered more. In this section the evaluation of the crankcase pressure on the engine performance is presented.

Let’s assume the indicated pressure-volume diagram of a 1 cylinder, 4 stroke internal combustion SI engine as the one showed in Figure 3.2-1.

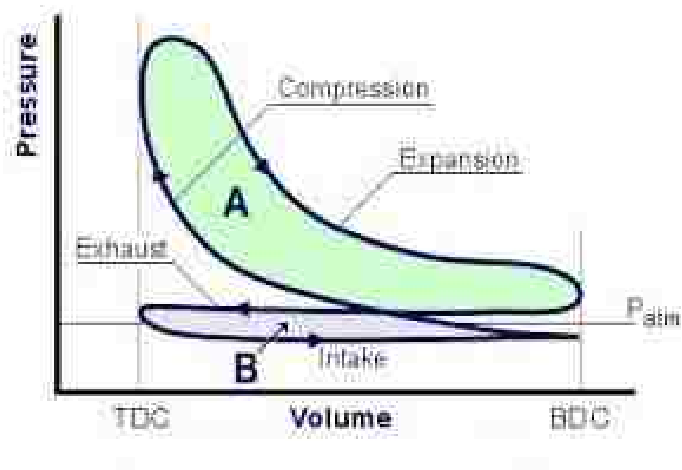

Figure 3.2-1: Pressure-volume diagram for an internal combustion SI engine [27].

The *A area* is generated with a clockwise evolution, so it is the positive work delivered by the engine to the outside environment; the *B area* instead is generated with a counterclockwise process so it is negative work, absorbed by the engine to change the fluid in the combustion chamber during the exhaust phase and the intake phase. The net work delivered by the engine is the difference between the *A area* and the *B area*, [28]. This net work can be computed with the integral over an entire engine cycle (2 revolutions because the engine is a 4 stroke engine) of the instantaneous pressure by the differential volume; this is called the **indicated net work per cycle**:

$$W_{ind} = \oint p \cdot dV \tag{3.10}$$

and

$$W_{ind} = A\ area - B\ area \tag{3.11}$$

where V is the cylinder displacement.

If the instantaneous pressure data of the gas in the combustion chamber are available over an entire engine cycle, the work delivered by the gas to the piston can be computed with Equation 3.10. The work computed in this way is called “indicated” because it derives from the pressure measurements.

The same net area of Figure 3.2-1 can be computed considering a constant value of pressure, which, multiplied by the cylinder displacement, will give the same value of indicated work; this pressure value is called indicated mean effective pressure and is an important indicator of the performance of the engine.

$$W_{ind} = \oint p \cdot dV = imep \cdot V \tag{3.12}$$

$$imep = \frac{W_{ind}}{V} \tag{3.13}$$

The concept is shown in Figure 3.2-2.

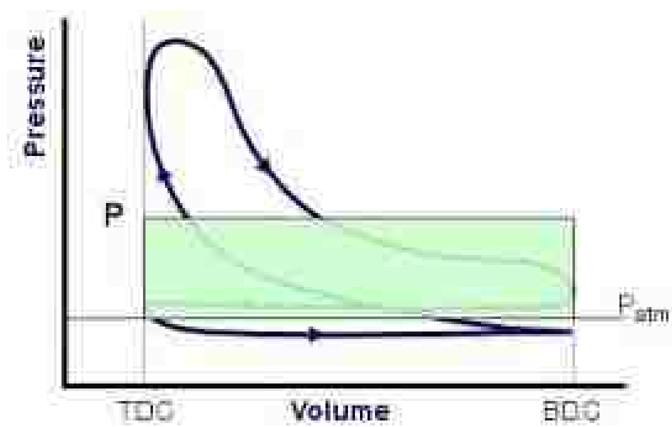

Figure 3.2-2: Indicated mean effective pressure diagram [27].

The indicated work is not the work really used to move the car: only a fraction of W_{ind} is available at the crankshaft, while the other part is used to overcome the internal frictions of the engine and to drive the engine accessories required for the engine functioning itself:

$$W_{ind} = W_{ICE} + W_{rf} + W_a \tag{3.13}$$

where:

- W_{ICE} represents the internal combustion engine useful work, the one which can be measured at the dynamometer test cell;
- W_{rf} represents the rubbing friction work, the one dissipated to win the friction of the internal components of the engine due to their relative motion (this includes friction between piston rings and piston skirt with the cylinder wall, the friction between big rod end and crankshaft, in the wrist pin, in the valve mechanism and camshaft bearings, friction in the pulley and belt or chain and gear used to drive the camshaft [1]);

- W_a represents the accessories work, the work used to drive all the engine accessories like water pump, oil pump, fuel pump, power steering pump, fan, AC system, alternator, etc.

The same decomposition can be seen from the point of view of the mean effective pressure, breaking up the indicated mean effective pressure in the sum of its subcomponents:

$$imep = bmep + rfmep + amep \tag{3.14}$$

where:

- *bmep* = brake mean effective pressure;
- *rfmep* = rubbing friction mean effective pressure;
- *amep* = accessories mean effective pressure.

The *bmep* is usually obtained from the measurement of the torque at the dynamometer test cell with the following expression:

$$bmep = \frac{T_{ICE} \cdot 4\pi}{V} \tag{3.15}$$

where T_{ICE} is the measured torque at the dynamometer.

The brake mean effective pressure is an important indicator of the ability of the designer to obtain useful work from a given engine displacement. Common values of maximum *bmep* for a naturally aspirated spark-ignition engine are around 10-12 bar, for an engine with maximum torque obtained at 3000 rpm; for a turbocharged compression-ignition engine the maximum *bmep* value are usually greater due to the greater pressure involved and they area in the range of 20-23 bar, [1].

At the same way, looking at what happens in the crankcase, it is possible to compute the crankcase pumping work lost in an engine cycle as:

$$W_{crankcase} = \oint p_{crankcase} \cdot dV_{crankcase} \tag{3.16}$$

where $p_{crankcase}$ is the crankcase pressure and $V_{crankcase}$ is the crankcase volume.

Dividing the crankcase work over the crankcase displacement it is possible to gain an important constant pressure value, called **crankcase pumping mean effective pressure**:

$$\text{CPMEP} = \frac{W_{crankcase}}{V} \tag{3.17}$$

The reduction of the crankcase pumping mean effective pressure is the main task of the present work. Notice that in the previous equation the work in the crankcase is divided by the engine displacement and not by the crankcase displacement; this is done in order to have a value to compare with the other indicators of engine performance like *bmep*, *rfmep*, *imep* and *amep* which are both related to the engine displacement. This is also confirmed by the literature [29], [30]. In GT-Power, instead, the CPMEP is computed dividing the crankcase work by the crankcase volume.

From Equation 3.16, it is clear that ***a decrease of the crankcase pressure will result in a decrease of the crankcase pumping mean effective pressure***.

The CPMEP is part of the *rfmep*, so that:

$$\text{CPMEP} \in rfmep \tag{3.18}$$

This means that a reduction of the CPMEP is a reduction of the *rfmep* and so, for a given *imep* value, the *bmep* increases: this leads to an increase of the engine overall efficiency.

$$\begin{gathered} p_{crankcase} \downarrow \\ \text{CPMEP} \downarrow \\ rfmep \downarrow \\ bmep \uparrow \\ \eta_o \uparrow \end{gathered} \tag{3.19}$$

Generally, the gas behavior inside the crankcase can be described with the Universal Gas Law:

$$p_{crankcase} \cdot V_{crankcase} = \frac{m}{\mu} \cdot R \cdot T_{crankcase} \tag{3.20}$$

where

$$\frac{m}{\mu} = number\ of\ moles \tag{3.21}$$

Let's suppose for a moment of closing the PCV and the make-up air circuit; the blow-by gasses cannot be evacuated. The number of moles increases because there is no way to evacuate the molecules of gas and, as a consequence, the crankcase pressure increases.

An increase in crankcase pressure increases the crankcase pumping work and so the engine efficiency decreases. For this reason, a crankcase ventilation system correctly designed in order to evacuate the gases and impede the pressure to build up is important for achieving greater engine efficiency. If the system is not correctly designed (for example, it is undersized) it will not be able to scavenge the crankcase and the engine efficiency will be affected.

In any case, also a too low pressure inside the crankcase is not optimal for the engine because below the crankcase, in normal wet sump engine, there is also the oil in the oil pan; the oil property and motion will be affected by the too low pressure level, increasing even more the splashing. A system correctly designed should provide a pressure level inside the crankcase as constant as possible, with low amplitude of variation. As said before, the reason for this is for engine seal integrity, as a too high or too low value of pressure can blow out the engine seals. The target pressure value is in the range 20-30 hPa lower than atmospheric pressure [9] .

3.3 Pressure variation inside the crankcase

The pressure inside the crankcase changes due to different factors:

- Blow-by flow;
- Piston movement;
- Breathing bay-to-bay flow;
- PCV valve flow;
- Make-up-air valve flow.

The blow-by flow increases the crankcase pressure because the number of moles in the crankcase increases as explained in the previous paragraph. The piston movement has the greatest effect on the pressure variation [31]: it affects the crankcase pressure because it makes the crankcase volume reduce and increase continually during an engine cycle. The breathing bay-to-bay flow is the flow which occurs between one portion of the crankcase to another one, in multi-cylinder engines; the portion of crankcase under a cylinder is called “bay” and holes (called breathing holes) are generated in the crankcase structure in order to have communication between them: the bay-to-bay communication makes one bay pressure to be affected by the adjacent bays pressure. The bay to bay communication will be discussed in detail in the following section. The PCV valve flow and the make-up-air flow guarantee the engine breathing with the outside environment releasing the pressure and destroying the vacuum.

In steady state working condition there is equilibrium between the blow-by flow, the PCV flow and the make-up-air flow.

The piston movement has the major influence on the pressure inside the crankcase. The actual system is an open system: in the present work in order to evaluate the piston movement effect on the crankcase pressure it will be assumed as a closed system, with no communication with the outside environment.

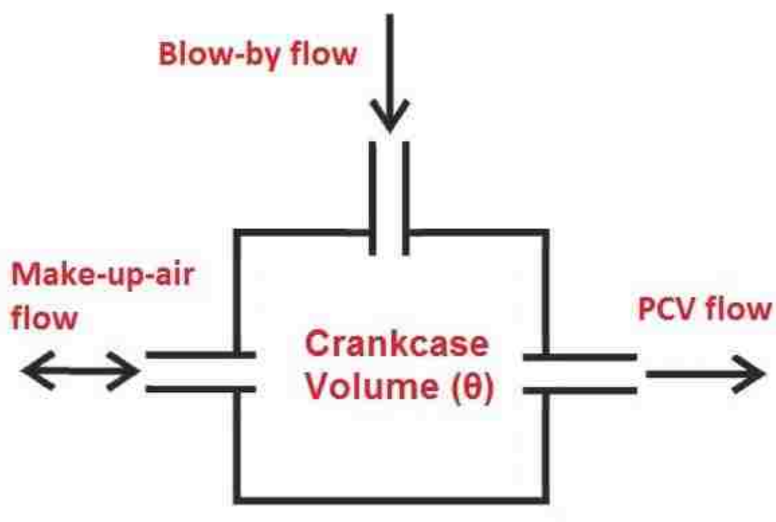

Figure 3.3-1: Crankcase open system (real).

Figure 3.3-2: Crankcase closed system (simplified).

The assumption is acceptable because the gas exchange with the outside has a little effect on the pressure variation respect to the piston movement. The flow displaced by the piston is much greater than the blow-by flow, so the last one has not a great effect on the results [32].

In order to have a rough idea of how much the pressure in the crankcase changes due to the piston motion, the crankcase volume has to be computed as function of the crank angle; let's assume a mono-cylinder engine with the following characteristics:

EXAMPLE ENGINE CHARACTERISTICS	
Displacement [V]	400 cc
Number of cylinder	1
Bore [b]	79.5 mm
Crank radius [r]	40.25 mm
Rod lenght [l]	135 mm
Conrod ratio [λ]	0.298
Combustion compression ratio	9.5:1
Crankcase compression ratio	1.2:1
Crankcase volume [$V_{crankcase}$]	400 cc
Cylinder bay volume [V_{cylbay}]	2 l

Table 3.3-1: Characteristics of the engine in the example.

The instantaneous total volume can be computed with the following equation:

$$V_{total}(\theta) = V_{cylbay} + V_{crankcase}(\theta) = V_{cylbay} + V_{crankcase} - \left(\pi \cdot \frac{b^2}{4}\right) \cdot r \cdot \left[(1-\cos\theta) + \frac{1}{\lambda} \cdot \left(1 - \sqrt{1 - \lambda^2 \cdot (\sin\theta)^2}\right)\right] \quad (3.22)$$

where $V_{crankcase}$ is the volume under the piston in the cylinder liner (blue region in Figure 3.3-4), V_{cylbay} is the constant volume for the crankshaft housing under the liner (red dashed region in Figure 3.3-4) and V_{total} is the sum of them. The maximum crankcase volume is obtained when the piston reaches the Top Dead Center (TDC), while minimum crankcase volume is when the piston approaches Bottom Dead Center (BDC).

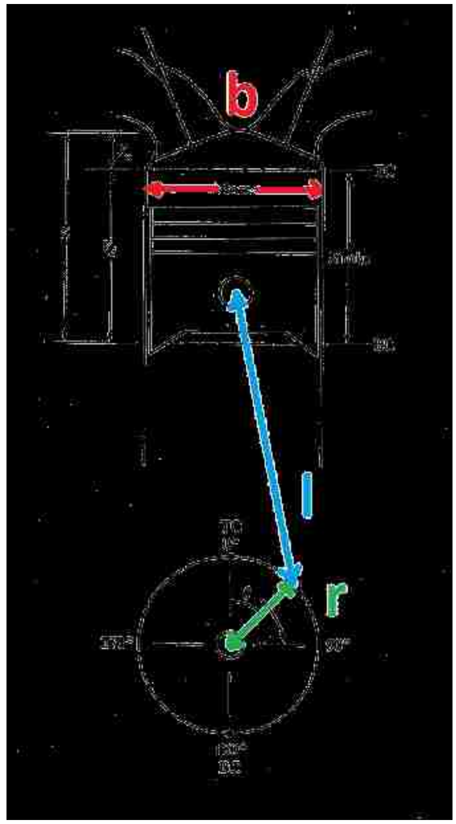

Figure 3.3-3: Basic geometry for an internal combustion engine [28].

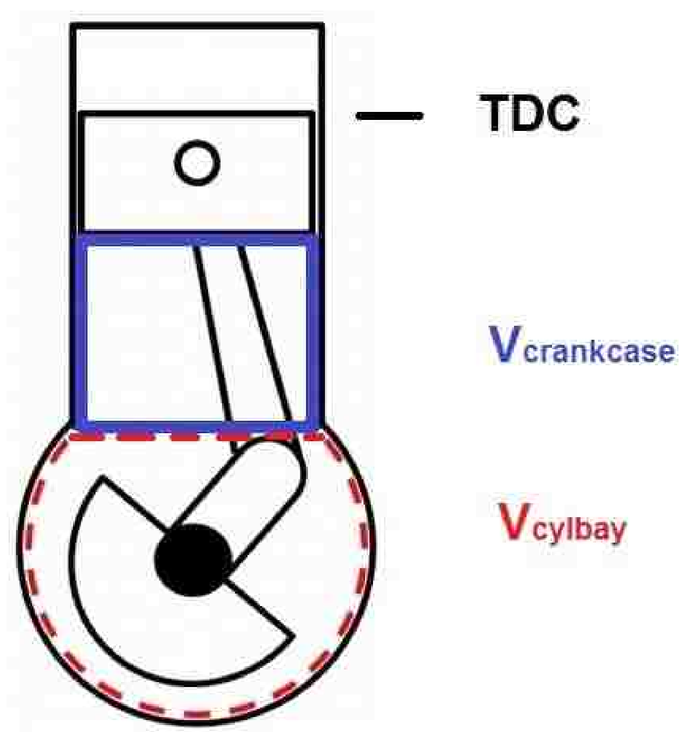

Figure 3.3-4: Crankcase bay volume variation [33].

According to [29], the behavior of the gas inside the crankcase can be described quite appropriately with a polytropic evolution, because there is no mass flow which enters or exits the crankcase volume:

$$p_{crankcase} \cdot V_{crankcase}{}^{\gamma} = constant \qquad (3.23)$$

The polytropic process is a general, arbitrary process in which heat transfer takes place; in the previous expression the term γ is the polytropic exponent, which should be chosen in order to match the experimental results. For this computation, just for simplicity, pure dry air has been considered as crankcase gas instead of a mixture of burned gases, and the isentropic exponent:

$$\gamma = 1.401 \qquad (3.24)$$

characteristic of dry air at 100°C has been chosen.

Computing the instantaneous volume with the Equation 3.22 and introducing the value in the Equation 3.23 it is possible to compute the instantaneous pressure value inside the crankcase.

The behavior of pressure and volume as function of the crank angle is plotted in Figure 3.3-1.

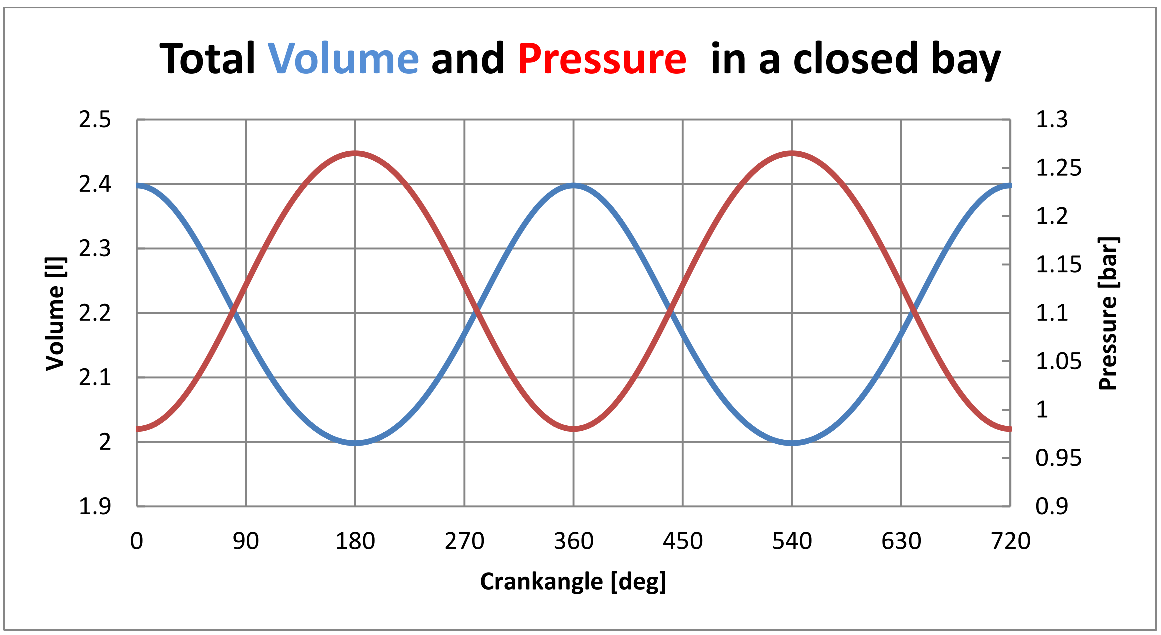

Figure 3.3-1: Pressure and volume variation as function of crank angle.

From this very simple model of the crankcase it is possible to notice that the fluctuation of the crankcase volume is close to the sinusoidal function (not exactly sinusoidal as expected from Equation 3.22): as a consequence the pressure follows the opposite trend, reaching the maximum value when the volume is minimum and the minimum value when the volume is maximum.

In order to compute the work lost in the crankcase due to the pumping, let's recover the following equation:

$$W_{crankcase} = \oint p_{crankcase} \cdot dV_{crankcase} \tag{3.25}$$

For an entire engine cycle the crankcase pumping work has the trend plotted in Figure 3.3-2.

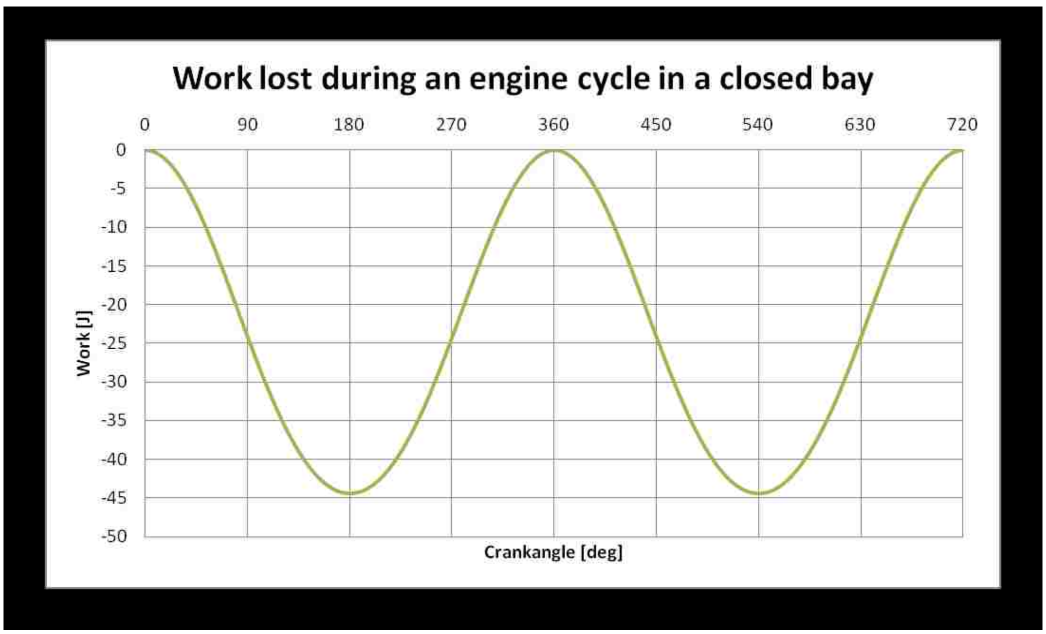

Figure 3.3-2: Crankcase pumping work during an engine cycle.

Ideally, with an isentropic process, all the work lost during the compression (when the piston is moving down during the intake phase) is recovered in expansion (the compression phase for the combustion chamber); the same happens for expansion and exhaust phases. The final work, integrated over the whole cycle is so zero, because if the crankcase pressure hinders the piston motion when it is moving toward the Bottom Dead Center, it also thrust the piston when it is moving up toward the Top Dead Center.

In this ideal case the crankcase pumping mean effective pressure is also zero.

$$W_{crankcase} = 0 \tag{3.26}$$

$$\text{CPMEP} = \frac{W_{crankcase}}{V} = 0 \tag{3.27}$$

From this very simplified model it is possible to conclude that **ideally, with an isentropic process, a crankcase completely closed, with no communication with the adjacent bays, will results in zero crankcase pumping mean effective pressure**. This is also confirmed by the literature [32], [29].

In any case a null CPMEP is impossible in a real engine with a common wet sump, and some CPMEP cannot be avoided for many reasons: first, due to the non-perfect symmetry inside the crankcase, not all the work lost in compression is recovered in expansion; in the proposed model, the crankcase is simply treated as a volume which is compressed and then expanded: this is not the real crankcase where there are also rotating elements like crankshaft, connecting rods, counterweights etc. but just a simplified model. If the engine has an oil pan, the oil motion has to be considered: the oil agitation will also absorb energy from the fluid, which will not be given back to the system, and will be lost. Oil agitation losses reduce engine power and aggravate the work for the crankcase ventilation system. Furthermore, heat transfer losses and friction of the flow with the hardware of the crankcase do not allow all the compression work to be recovered in the expansion of the crankcase.

From Figure 3.3-2, it can be seen that the amplitude of pressure variation is around 25000 Pa, which is a too great value for the correct operation of the crankcase; with such a high pressure variation, the oil is moving to much inside the crankcase of a wet sump engine and it would be difficult for the oil pump to suck the oil and guarantee the correct lubrication. The high pressure fluctuation will lead to a high oil mist generation inside the crankcase, so that a high effort is required to the oil separator; surge oil will reach the oil separator and could easily saturate it, decreasing the overall system performances. The high peak pressure will also generate leakages of oil from the crankcase gasket. Moreover, the high peak pressure will impede a correct oil drain from the cylinder head to the crankshaft bearings, reducing the lubrication, with potential damages to the engine. For these reasons, a completely closed crankcase bay is not applicable for an engine with an oil-sump, and another way of pressure release should be found to keep the crankcase pressure fluctuations as small as possible. Usually, the pressure amplitude of variation inside the crankcase is around 5000-7000 Pa at 6000 rpm.

3.4 Pressure release inside the crankcase: breathing holes

The solution commonly adopted to release the high peak pressure in a bay is the creation of some holes in the crankcase structure: the bay-to-bay communication is possible because the gas can flow through these holes; the fluid can pass from one bay to another one and the high pressure is released.

Figure 3.4-1: Breather (or breathing) holes between cylinder 1 and cylinder 2 [30].

These holes are called breathing holes, or breather holes, or ventilation holes and, together with the area between the windage tray and the main bearing caps in a wet sump engine, make the bay-to-bay gas exchange possible.

The whole area of communication between bays and oil pan is called ventilating area or breather/breathing area.

It consists of:

- Bay to bay holes.

- Windage tray holes.
- Area under main bearing caps.
- Windage tray side areas.

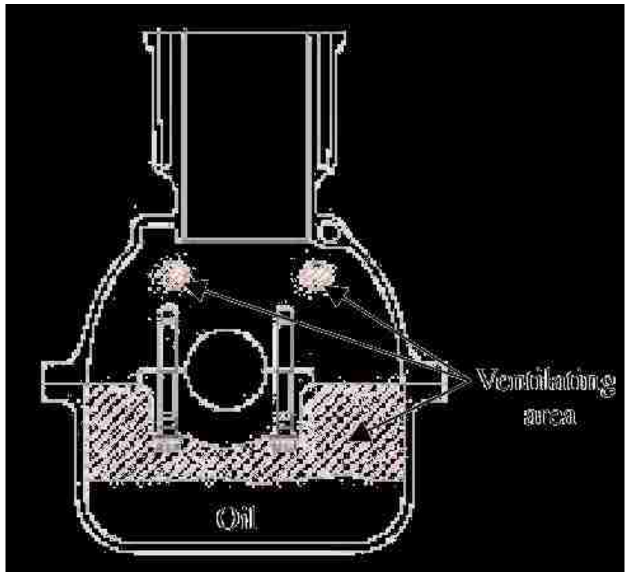

Figure 3.4-2: Ventilating area [34].

The ventilating area allows the amplitude of pressure variation inside the crankcase to be reduced, but induces another problem: the gas flow has to pass through these holes and resistance is generated; this directly hinders the piston motion. The frictional resistance which is generated when the gas pass through a pipe causes a drag force to appear and this lead to pumping work inside the crankcase.

3.5 Crankcase pumping mean effective pressure

In this section the crankcase pumping work and mean effective pressure due to the flow through the ventilating area are evaluated in order to provide an analytical description of the problem. Let's assume to have a 2 cylinder 4 stroke engine, without oil sump, with phase shift between the cylinders of 180° degree as the one in Figure 3.5-1, considering for simplicity only 1 breather hole. When the first cylinder is moving down, the bay pressure is increasing due to the reduction of the volume, while the second piston is moving up and the related bay pressure is decreasing; the flow between the breathing hole allows the pressure in the two bays to be nearly constant.

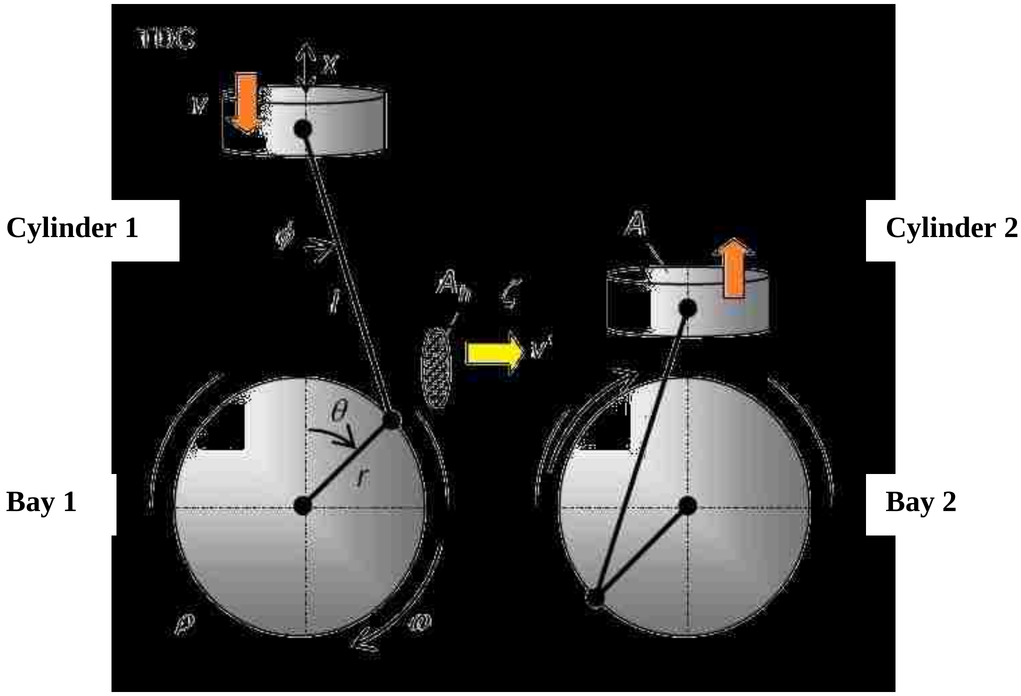

Figure 3.5-1: Pumping flow in a 2 cylinder engine crankcase [30].

In Figure 3.5-1 the symbols have the following meaning:

x = piston position wrt TDC *[m]*

v = piston speed *[m/s]*

ϕ = rod angle *[rad]*

A = piston surface *[m ²]*

ρ = gas density *[Kg/ m ³]*

r = crank radius *[m]*

θ = crank angle *[rad]*

l = rod length *[m]*

λ = r/l = conrod ratio

Let's first compute the flow velocity in the breathing hole: the flow displaced by the piston motion must be equal to the flow across the hole.

$$A \cdot v = A_h \cdot v' \tag{3.28}$$

$$v' = \frac{A}{A_h} v \tag{3.29}$$

The losses due to the flow through the breathing hole generate pressure losses between bay 1 and bay 2:

$$\Delta p = \zeta \frac{1}{2} \rho v'^2 \tag{3.30}$$

where ζ is the pressure loss coefficient and ρ is the fluid density (oil and gas).

The pressure loss coefficient is a constant value which considers the effect of pressure losses across an orifice: there are many modes in which the pressure loss can happen, as described in Figure 3.5-2 [33]:

- loss due to contraction;
- loss due to expansion;
- loss due to hole thickness friction;
- loss due to face friction.

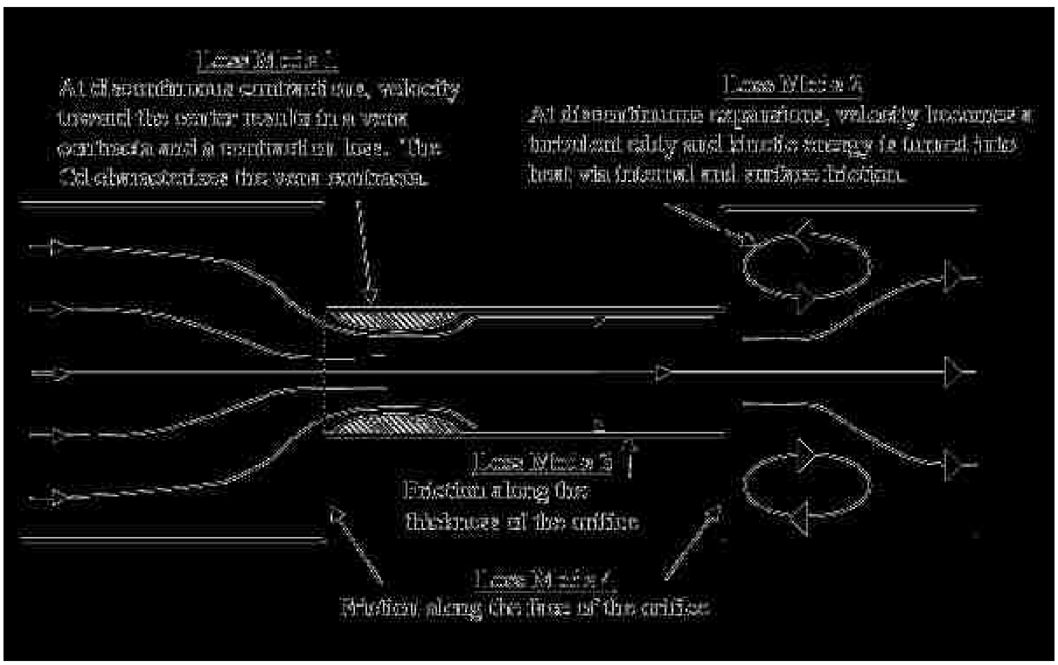

Figure 3.5-2: Pressure losses [35].

The pumping power lost can be computed multiplying the drag force due to the flow through the hole by the velocity of the flow in the hole.

$$\dot{W}_{crankcase} = F_d \cdot |v| = \Delta p \cdot A \cdot |v| = \zeta \frac{1}{2} \rho v'^2 \cdot A \cdot |v| = \frac{1}{2} \zeta \rho \frac{A^3}{{A_h}^2} v^2 |v| \tag{3.31}$$

The work lost during an engine cycle is computed integrating the power over the cycle time: for computation purpose, it is more useful to integrate the power in terms of crank angle instead of cycle time, keeping in mind that:

$$\frac{dt}{t_c} = \frac{d\theta}{4\pi} \tag{3.32}$$

and

$$\omega = 2\pi f = \frac{4\pi}{t_c} \tag{3.33}$$

The expression of the work assumes the form of Equation 3.34:

$$W_{crankcase} = \frac{1}{t_c}\int_0^{t_c} \dot{W}_{crankcase} \cdot dt = \frac{1}{4\pi}\int_0^{4\pi} \dot{W}_{crankcase} \cdot \frac{d\theta}{2\pi f} \tag{3.34}$$

It is now necessary to compute the piston velocity as function of crank angle. Applying a simple geometric law for a piston-crank system the piston position can be computed in the following way:

$$x \cong r\left[(1-\cos\theta) + \frac{1}{\lambda}\left(1-\sqrt{1-\lambda^2(\sin\theta)^2}\right)\right] \tag{3.35}$$

Deriving the piston position respect to the time, under the assumption of constant crank rotational speed ω the piston position can be computed with the Equation 3.36:

$$v \cong r\omega\left(sin\,\theta + \frac{\lambda}{2} sin\,2\theta\right) \tag{3.36}$$

Now introducing Equation 3.36 and piston velocity in Equation 3.34 the pumping work lost during an engine cycle assumes the form:

$$\begin{aligned}
W_{crankcase} &= \frac{1}{t_c}\int_0^{t_c} \dot{W}_{crankcase} \cdot dt = \frac{1}{4\pi}\int_0^{4\pi} \dot{W}_{crankcase} \cdot \frac{d\theta}{2\pi f} \\
&= \frac{1}{8\pi^2 f}\int_0^{4\pi} \dot{W}_{crankcase} \cdot d\theta \\
&= \frac{1}{8\pi^2 f}\int_0^{4\pi} \frac{1}{2}\zeta\rho\frac{A^3}{{A_h}^2} v^2|v| \cdot d\theta \\
&= \frac{1}{4\pi^2 f}\zeta\rho\frac{A^3}{{A_h}^2} r^3\,\omega^3 \int_0^{\pi}\left(\sin\theta + \frac{\lambda}{2}\sin 2\theta\right)^3 \cdot d\theta \\
&= \frac{1}{4\pi^2 f}\zeta\rho\frac{A^3}{{A_h}^2} r^3\omega^3\left(\frac{4}{3} + \frac{4}{5}\lambda^2\right)
\end{aligned} \tag{3.37}$$

Here, instead of the integration of the trigonometric function over 2 revolutions (which will have brought a null value) it has been integrated from 0 to π and then multiplied by 4 for giving a not null value of the work: this is acceptable due to the symmetry of the system.

Introducing in the previous equation the rotational speed and the cylinder volume it can be rewritten as:

$$W_{crankcase} = \frac{\pi}{4}\zeta\rho\frac{{V_u}^3}{{A_h}^2}f^2\left(\frac{4}{3}+\frac{4}{5}\lambda^2\right) \tag{3.38}$$

where

$$V_u = A \cdot 2r \tag{3.39}$$

It is better to relate the work to values which can be directly measured from the engine: instead of writing the work as function of the rotational frequency *f*, the engine rotation per minute *rpm* is introduced:

$$f = \frac{rpm}{60} \tag{3.40}$$

and the work is rewritten as:

$$W_{crankcase} = \frac{\pi}{14400}\zeta\rho\frac{{V_u}^3}{{A_h}^2}rpm^2\left(\frac{4}{3}+\frac{4}{5}\lambda^2\right) \tag{3.41}$$

At this point it is possible to compute the crankcase pumping mean effective pressure as the ratio between the pumping work during one engine cycle and the engine displacement.

$$\text{CPMEP} = \frac{W_{crankcase}}{V} = \frac{W_{crankcase}}{2V_u} = \frac{\pi}{28800}\zeta\rho\left(\frac{4}{3}+\frac{4}{5}\lambda^2\right)\left(\frac{V_u}{A_h}rpm\right)^2 \tag{3.42}$$

It is possible to notice that the crankcase pumping mean effective pressure is proportional to the squared value of cylinder displacement V_u and engine rotational speed per minute rpm and inversely proportional to the squared value of the ventilating area A_h. It is also directly proportional to the pressure loss coefficient ζ and to the fluid density inside the crankcase ρ.

CPMEP should be minimum in order to increase the engine efficiency: the size of the breathing holes should be as great as possible, but usually engine designers have to choose a trade-off between power losses and crankcase stiffness. Common values of bay-to-bay holes diameter for road car engines are in the order of 20-30 mm: usually there are two breather holes for the communication between two adjacent bays.

Also the fluid density has an important effect on the CPMEP: if the oil concentration is high in the windage cloud, the fluid density is high so the CPMEP is great: for this reason a windage tray is usually adopted, which has the target to limit the oil mist formation inside the crankcase and reduce the oil aeration, which will affect the lubrication system.

4 GT-POWER MODEL CREATION

4.1 Pentastar V6 3.6 liters

The engine considered for the simulation activity is the Chrysler Pentastar V6 3.6 liters which was introduced in 2011 and is under the hood of many Chrysler, Dodge and Jeep. The general specifications are summarised in the table below:

PENTASTAR 3.6-Liter V6 GENERAL SPECIFICATIONS

Type and Description	60-degree V type, Dual Overhead Camshaft, liquid cooled
Displacement	220 cubic inches - 3604 cc
Bore x stroke	3.38 in x 3.27 in - 96.0 mm x 83.0 mm
Crankshaft	Fillet rolled, nodular cast iron
Connecting Rods	Forged, sintered cracked
Pistons	Cast Aluminum
Valve System	Chain driven DOHC with dual independent cam phasing; 24 valves, hydraulic end pivot roller rockers
Valve Dimensions and angle	Intake 39 mm, 17 degrees; Exhaust 30 mm, 18.8 degrees
Combustion Chamber	Four valve, pent roof, 52.7 cc
Fuel Delivery	Sequential, multi-port, electronic, return less
Construction	Aluminum, deep skirt block with aluminum alloy cylinder heads
Engine Length	503 mm

Engine Width	567 mm
Compression ratio	10.2:1
Fuel requirement	Unleaded regular (antiknock index 87), E85 capable
Oil Capacity	6 quarts (5.7 l) with filter
Oil Change Intervals	8,000 miles / 12,875 km
Coolant Capacity	14.0 quarts (13.25 l)
Firing Order	123456
Emission Controls	Dual three-way catalytic converters, heated oxygen sensors and internal; Tier-II Bin 5
Performance	Max Power: 283 HP (211 kW) at 6350 rpm Max Torque: 260 lb.-ft. (353 Nm) at 4400 rpm

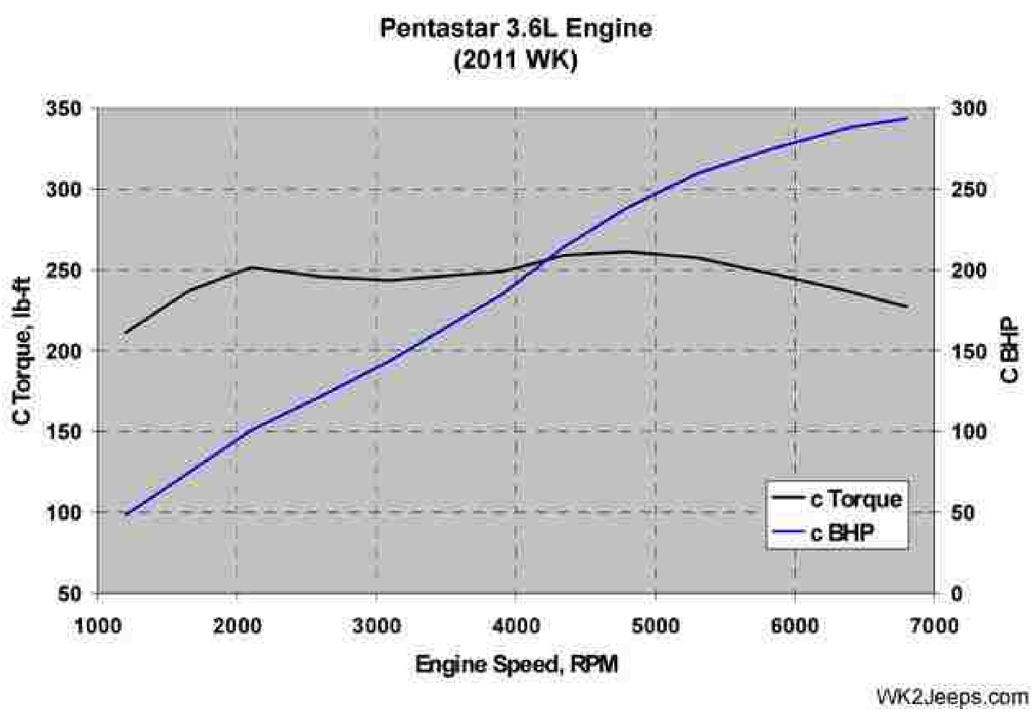

Figure 4.1-1: Pentastar V6 3.6L engine [36].

4.2 GT-Power

GT-Power is the leading engine simulation software used by the majority of engine manufacturers to simulate the performances of their engines. It is a 1D simulation tool for computational fluid dynamic (CFD); it is a licensed product from the package GT-Suite, provided by the company Gamma Technologies. It can be used to simulate and analyze all the problems which are relative to the engine performance. The way the fluids behave in the modeled internal combustion engine (intake air, fuel, combustion products, blow-by gases…) is described by means of equations. The behavior of these fluids is predicted by solving the Navier-Stokes equations, i.e. conservation of continuity, momentum and energy equations.

In GT-Power, the solutions are based on one-dimensional fluid dynamics: the equations used to describe the behavior of the fluids are resolved in one dimension and this basically means that all the quantities (pressure, temperature, velocity, density…) are averages through the direction of the flow [35]. The software also computes the heat transfer and the friction related to the flow in order to give reliable results.

With the use of this software, the author simulated the behavior of the crankcase ventilation system. In order to do that the two models described in paragraph 1.3 are required; these models basically discretize the volumes touched by the flow (so in the case of blow-by gases, the crankcase volume, oil pan volume, front cover volume, cylinder head, cam cover volume…) in nozzles, pipes and flow splits. These are called "objects" following the terminology of the software and in every object sub- models of equations are implemented in order to describe the flow behavior as discussed above.

4.3 Pre-processing phase

In order to create the model, some pre-processing operations are required; the aim is to grab the exact volume which is touched by the blow-by gases. This can be done with the use of specific software together with the CAD file of the modeled engine. The pictures below show the empty spaces considered.

Figure 4.3-1: Engine block.

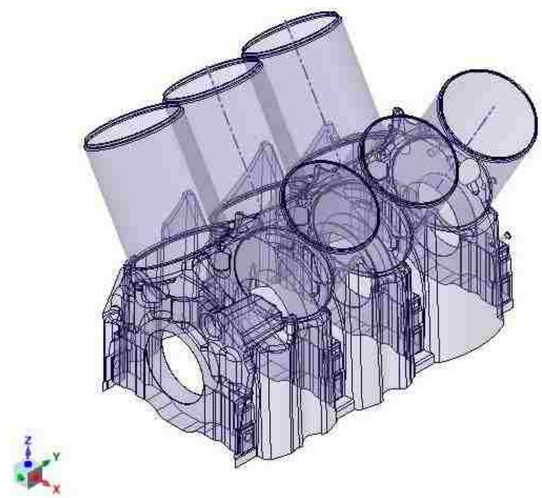

Figure 4.3-2: Empty space of the engine block.

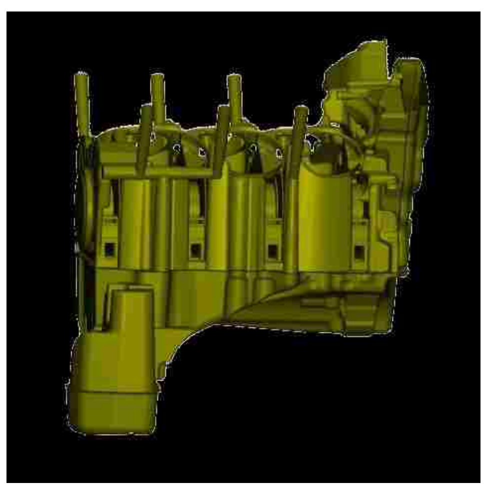

Figure 4.3-3: Empty space crankcase, front cover, oil pan and oildrains.

In particular, starting from the CAD of Pentastar V6, the author has grabbed the volumes which need to be considered; he has been helped by a Chrysler employee, namely Badema Mulaosmanovic who did the majority of the work. The used software are GT-Spaceclaim (which is a software included in the GT-Suite package) and NX. This preprocessing phase was quite long (2 months) and required a lot of effort because the crankcase, oil pan, front cover and heads have very complex geometries.

4.4 1D model creation

After the creation of all the required volumes, they need to be discretized: this means that they need to be split in smaller volumes. In GT-Power, the whole system has to be divided in many small sub-volumes and for each of them the software computes the scalar variables (like pressure, temperature and density) which are supposed to be uniform in the considered sub-volume [35]. This means that large volumes will poorly describe the real behavior of the fluid because only one value for scalar quantities is considered and the small variations are not caught by the system. On the opposite, if the discretization is too small the computational time will be high and the results will not bring effective additional information.

The vector quantities (like mass flux and velocity) are instead computed at the boundary of each sub-volume [35].

According to the GT-Suite Flow Theory Manual [35], the equations implemented in the software and used to compute the outputs are:

$$Continuity: \quad \frac{dm}{dt} = \sum_{boundaries} \dot{m} \tag{4.1}$$

$$Energy: \quad \frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum_{boundaries} (\dot{m}H) - hA_s\left(T_{fluid} - T_{wall}\right) \tag{4.2}$$

$$Momentum: \quad \frac{d\dot{m}}{dt} = \frac{dpA + \sum_{boundaries}(\dot{m}v) - 4C_f\frac{\rho v|v|}{2}\frac{dxA}{D} - \zeta\left(\frac{1}{2}\rho v|v|\right)A}{dx} \tag{4.3}$$

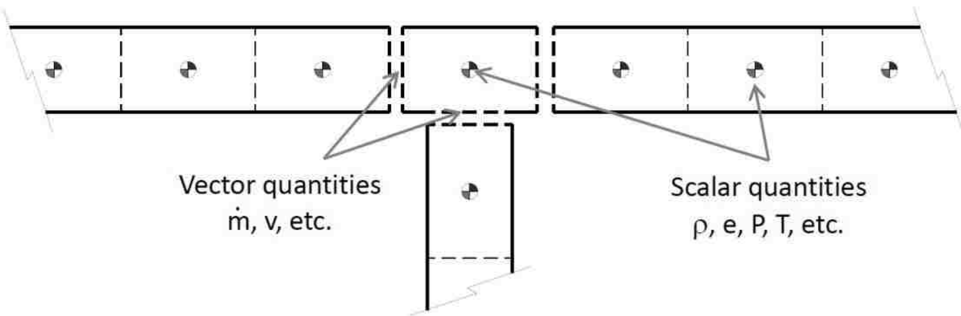

Figure 4.4-1: GT-Power discretization of a T pipe [35].

A detailed description of all the equations and sub-models implemented in the software GT-Power can be found in the GT-Suite Flow Theory Manual Version 7.4 [35].

All the volumes obtained from the preprocessing phase have so been discretized. This phase was done with the use of software from the GT-Suite package called GEM3D. This software allows the user to split volumes into sub-volumes and then convert them in objects like flowsplits, pipe and orifices to be imported on GT-Power model. Just as an example, considering Figure 4.4-2, part of the volume showed in Figure 4.3-3 has been divided: here the crankcase and the front cover volumes are considered. Different colors mean different sub-volumes so the software will compute all the requested scalar quantities for each of them and the vector quantities at their boundaries.

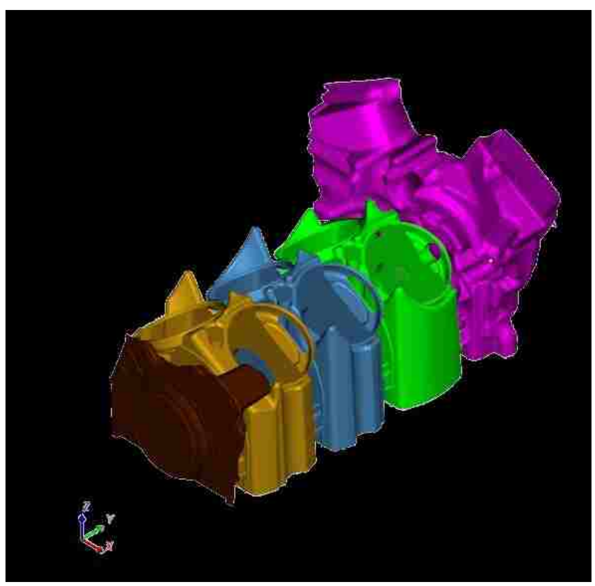

Figure 4.4-2: Crankcase and front cover volumes.

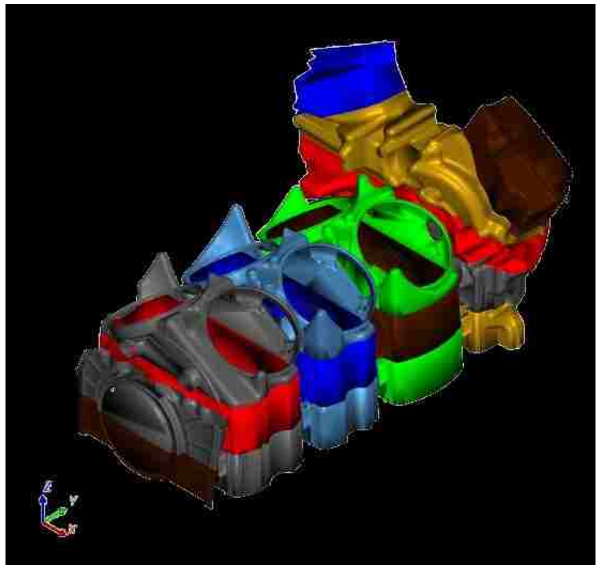

Figure 4.4-3: Crankcase and front cover volumes further split.

As shown in Figure 4.4-3 the previous mentioned sub-volumes have been split again in order to improve the accuracy of the simulation.

All the sub-volumes are then converted in flowsplits, pipes and orifices, which are the most common objects used for flow simulation in GT-Power.

Considering for example the top part of the crankcase bay of cylinder 1&2, the section of the cylinders area (1 and 2), the holes for communication with the front cover volume (3 and 4), the bay to bay holes (5 and 6) and the section of communication with the bottom part of the crankcase (7) can be noticed from Figure 4.4-4. This geometry is converted in a flowsplit which can be imported in GT-Power for the simulation.

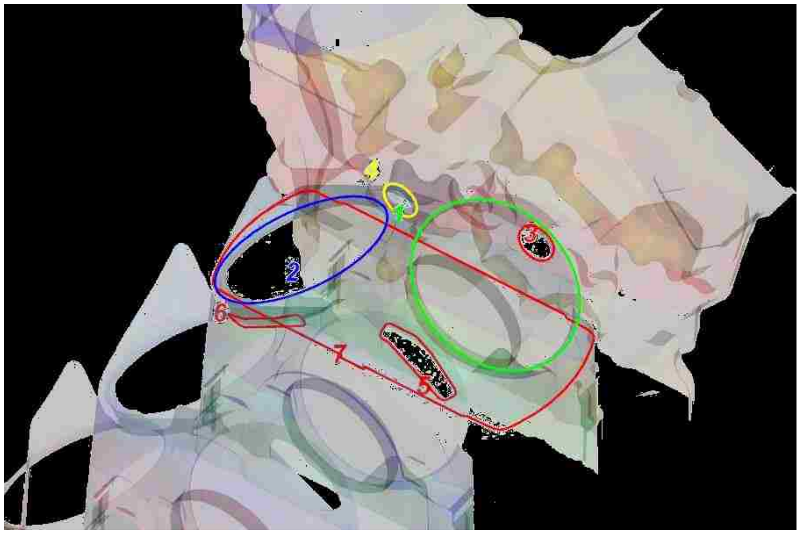

Figure 4.4-4: Flowsplit generation starting with GEM3D.

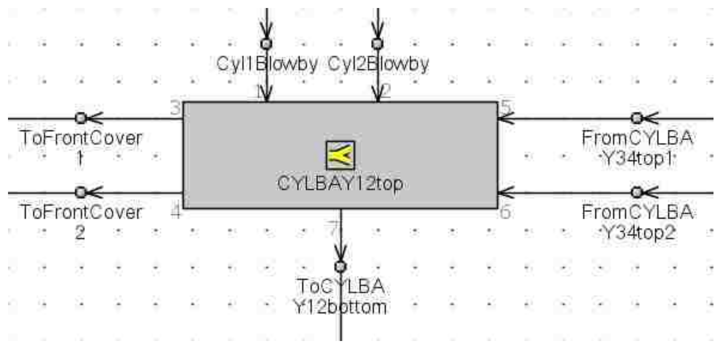

Figure 4.4-5: Flowsplit generated in GT-Power.

The volumes so computed do not consider all the organs which are inside them; for example, considering the crankcase volume, once it has been calculated with the pre-processing phase, it needs to be decreased by the amount of volume which is occupied by crankshaft, bearings and connecting rods, otherwise it will overestimate the real space available for the passage of the flow. In the same way, considering the oil pan, the volume occupied by the oil should be subtracted from the actual pan capacity because it does not constitute a space available to the flow. The volume of valve distribution chains and tensioners must be subtracted from the front cover volume; the volume of camshafts, roller rockers, valves and valve springs must be subtracted from the volume enclosed by cylinder head and cam cover. In this way the actual volumes available to the flow can be correctly estimated.

The final model is showed in Figure 4.4-6 and Figure 4.4-7.

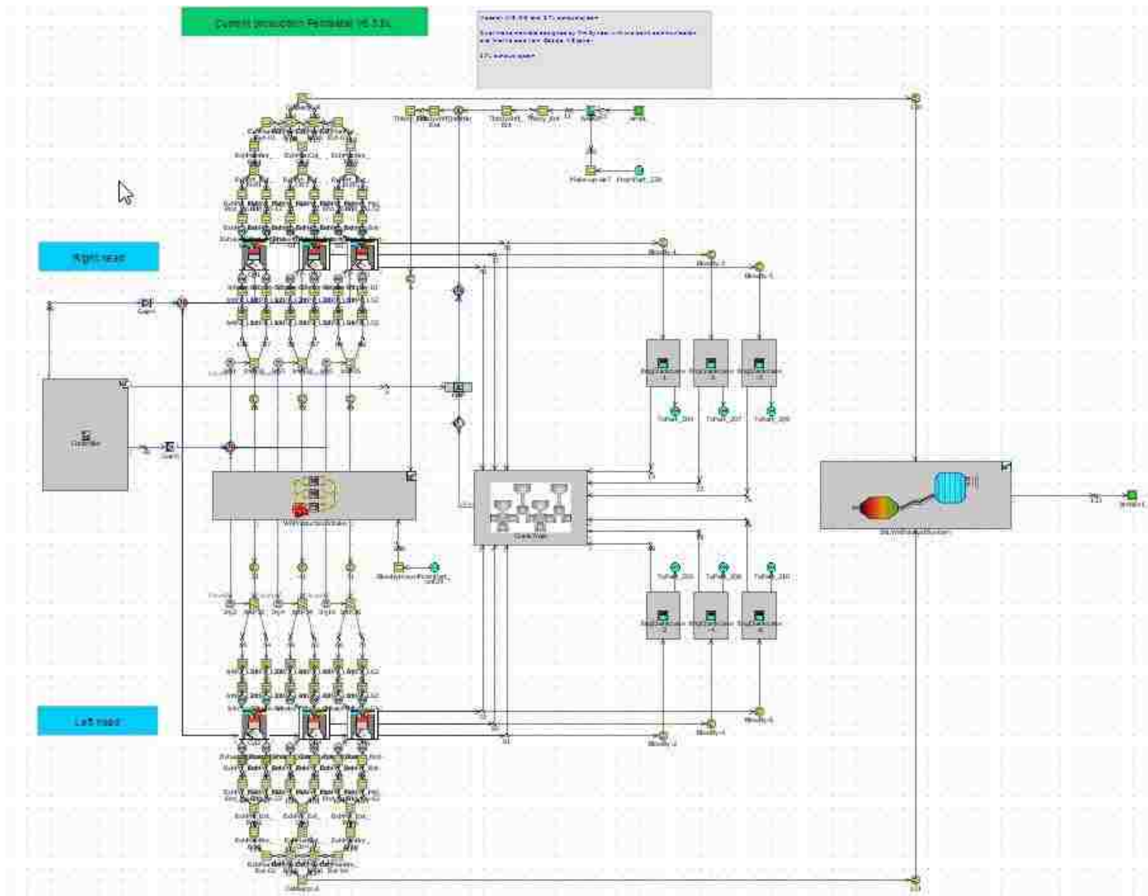

Figure 4.4-6: GT-Power model: Engine.

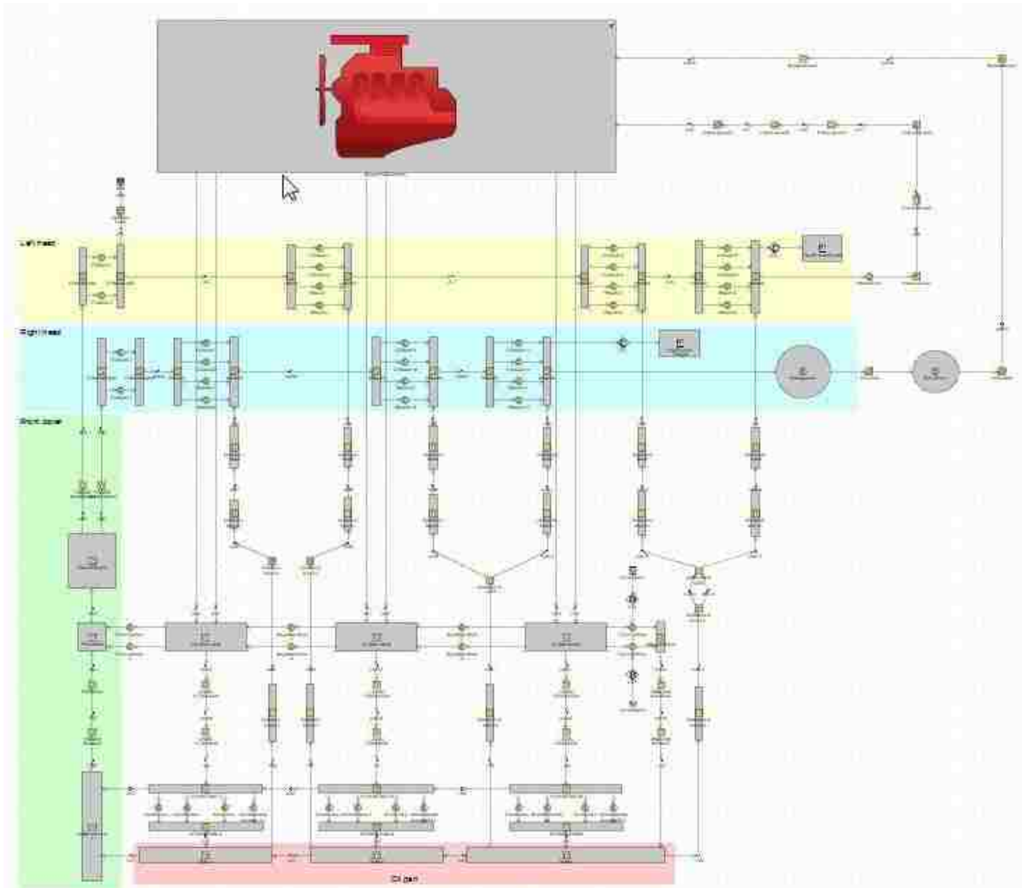

Figure 4.4-7: GT-Power model: Crankcase Ventilation System.

4.5 Model correlation

After the creation of the model, it has been correlated with some experimental data coming from a real engine.

The geometries involved were taken from the CAD file of the engine, so those values have not been changed; the correlation process basically consisted on the calibration of diameter and coefficient of discharge C_D of the "orifice" objects used: the coefficient C_D counts for the fact that when a fluid pass through an orifice the available section is lower than the reference one; it is the ratio between the effective flow area and the reference flow area, considering the isentropic velocity in the throat [35].

According to the GT-Suite Flow Theory Manual it is defined as:

$$\dot{m} = A_{eff}\rho_{is}U_{is} = C_D A_{ref}\rho_{is}U_{is} \tag{4.4}$$

and

$$\rho_{is} = \rho_o (P_r)^{1/\gamma} \tag{4.5}$$

$$U_{is} = \sqrt{RT_o}\left\{\frac{2\gamma}{\gamma - 1}\left[1 - P_r^{\frac{\gamma-1}{\gamma}}\right]\right\}^{1/2} \tag{4.6}$$

In some conditions the flow can be chocked; the flow is chocked when

$$P_r \leq \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}} \tag{4.7}$$

and the speed remains constant for any further decrease in downstream pressure.

$$\rho_{is} = \rho_o \left(\frac{2}{\gamma + 1}\right)^{\frac{1}{\gamma-1}} \tag{4.8}$$

$$U_{is} = \sqrt{\gamma RT_o}\left\{\frac{2\gamma}{\gamma - 1}\right\}^{1/2} \tag{4.9}$$

In the model, chocked condition happens in the PCV valve at part throttle, when a high differential pressure between crankcase and manifold is available.

The most difficult part of the correlation process was the calibration of the blow-by orifices; these orifices should simulate the escaping flow from the combustion chamber. The real flow area is difficult to be measured, also from a CAD file, because the available flow space is related to the piston ring axial and radial clearances; these change due to the piston movement. Moreover, the sealing characteristics of the rings are highly nonlinear, and the blow-by production can follow very unexpected trends [5] as the one depicted in Figure 4.5-1. The trend plotted is only a general one; it is really characteristic of the type of engine and it is function of many parameters, like cylinder pressure and engine speed.

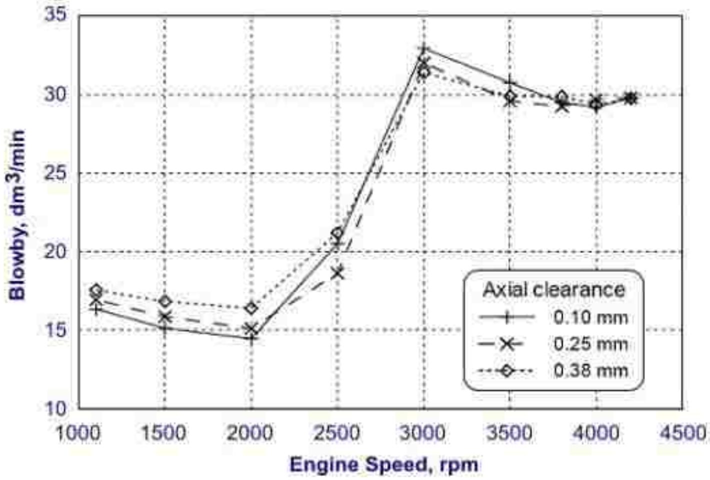

Figure 4.5-1: Blow-by production as function of rpm (general values) [5].

For this reason, the calibration of the blow-by orifices was the most time demanding part of the calibration work. In order to give some results close to reality, the coefficient of discharge of the diameter of the blow-by orifices is speed and load dependent.

As later explained the simulation has been carried for two different working conditions, at part throttle condition and at wide open throttle condition. The model has been correlated for the two cases considered.

The correlation process was done with the tool Optimizer of GT-Power: with this tool, imposing a given volume flow rate through the blow-by orifices, the software computes the coefficient of discharge which provides the required flow. The required volume flow rate is the one coming from the blow-by production of the real engine data. The diameter of the orifices was fixed at 1 mm for all the six orifices.

The next step was the calibration of the PCV valve; it was modeled as an orifice object. In the PCV valve, the passage section is changing as function of the differential pressure between crankcase and intake manifold; the orifice should behave in the same way . The flow through the valve is not engine speed dependent and it is only function of this differential pressure. For the two conditions considered, the relative differential pressure between crankcase and manifold has been computed. For the first case (part throttle) the differential pressure is close to 60 kPa, nearly constant for the speed range considered and the available section is minimum; the opposite case is at WOT condition, where the differential pressure is much smaller (5 or 6 kPa) and the section in the orifice is the maximum available. The correct diameter has been set in the PCV valve orifice object for these two cases, so that the flow through the PCV orifice matches the real flow through the valve, coming from experimental data. The Optimizer tool of GT-Power has been used for this purpose.

5 RESULTS AND DISCUSSION

5.1 Running the model

The next step has been the setup of the cases studies to be considered in the simulation. As mentioned before, for the present job, two cases have been analyzed:

- Case 1: part throttle condition.
- Case 2: wide open throttle condition.

The two cases are described in the following table:

	Case 1	Case 2
Throttle	Partially open	Fully open
bmep (bar)	2.2	max.
RPM	1200 to 6800	1200 to 6800
T coolant initial (K)	372	372
T oil initial (K)	377	377
External pressure (bar)	0.9829	0.9829
External Temperature (K)	303	303

Table 5.1-1: Case setup.

The first case represents a cruising condition when a small amount of power is requested from the engine; the throttle is nearly closed and so the intake manifold's pressure is very low. The second case is the opposite and represents the best performance which can be pulled out from the engine. The throttle is fully opened at 90°.

Both the cases have been evaluated at 8 different engine speeds, from 1200 rpm to 6800 rpm, with an interval of 800 rpm.

The model was run and the results are shown in the following plots.

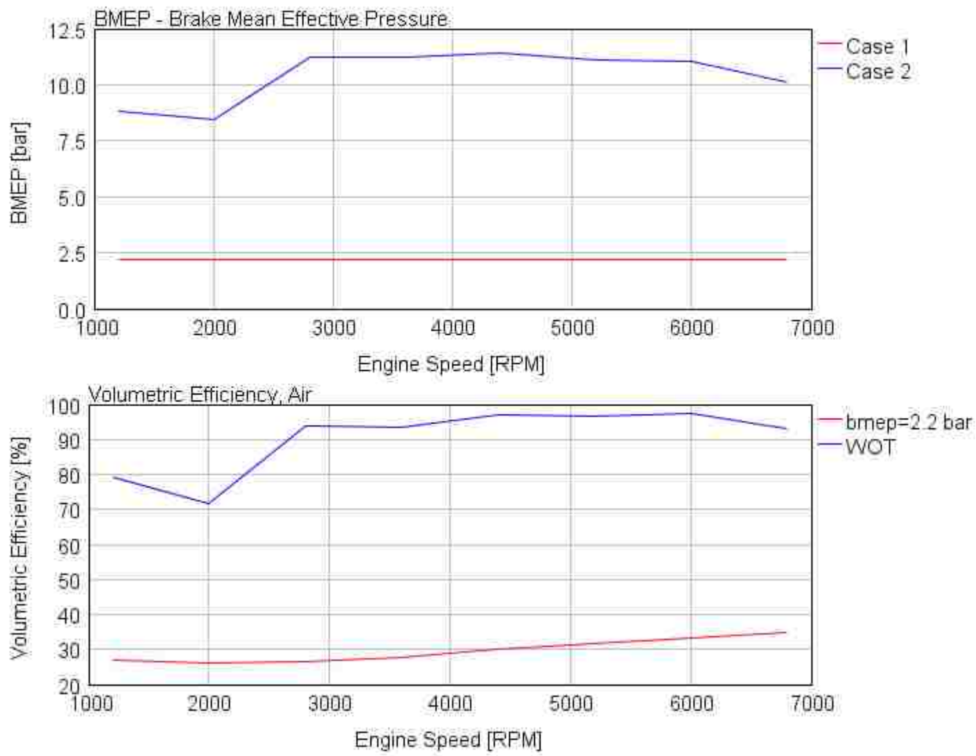

Figure 5.1-1: Bmep and Volumetric Efficiency.

The *bmep* and volumetric efficiency for the two cases is shown in Figure 5.1-1.

The volumetric efficiency follows a trend very similar to the one of the *bmep.* They are related by the following equation:

$$bmep = \lambda_v \eta_o \frac{CV}{AFR} \rho_{air} \tag{5.7}$$

In the simulation, calorific value *CV* and air density ρ_{air} are constant; the air fuel ratio *AFR* is quite constant around stoichiometric value for Case 1, while a little enrichment is present at high speed in Case 2. In any case, the *AFR* changes are small and the brake mean effective pressure is mainly function of volumetric efficiency and overall engine efficiency.

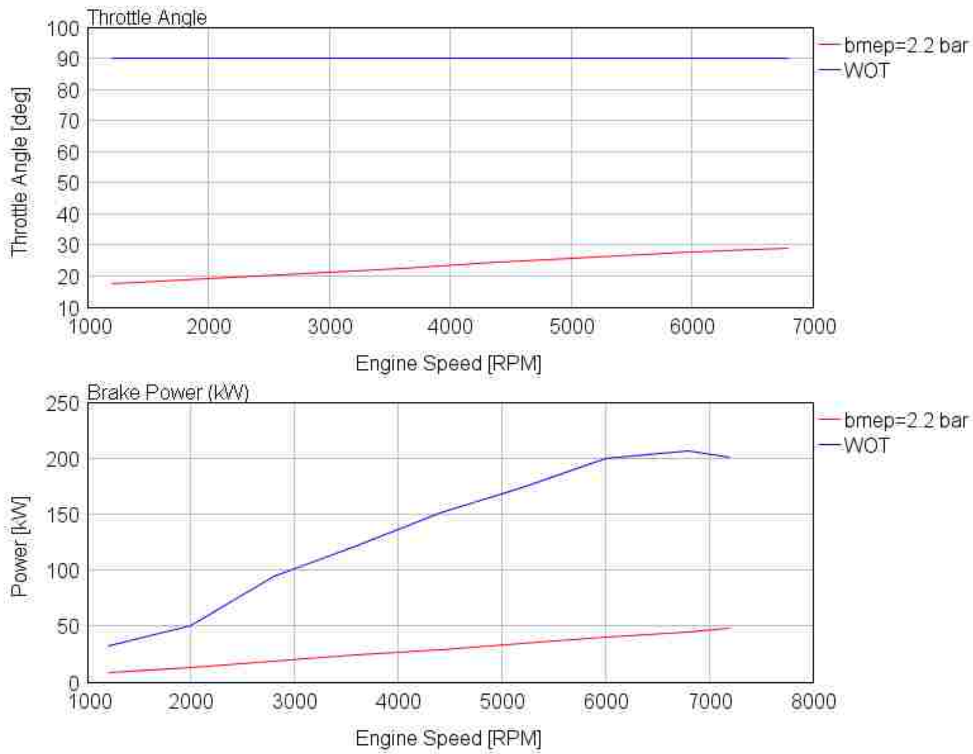

Figure 5.1-2: Throttle Angle and Brake Power.

Considering Case 1 (red line), in order to keep a constant output (*bmep=2.2 bar*) the throttle needs to be opened more; this happens because the overall efficiency is decreasing. At high speed the mechanical losses are greater and the mechanical efficiency is lower, thus, the volumetric efficiency has to increase if constant output (*bmep*) is requested. As a consequence the throttle angle needs to be slightly increased.

For Case 2 the volumetric efficiency is the maximum possible because the maximum performances are requested to the system.

The volumetric efficiency is related to the pressure inside the intake manifold; if the throttle is partly opened (Case 1) the volumetric efficiency is very low and so the manifold pressure. For Case 1 the average manifold absolute pressure was around 40 kPa, with small variations in the engine speed range. For Case 2 the manifold absolute pressure was close to the atmospheric pressure because the throttle was fully open.

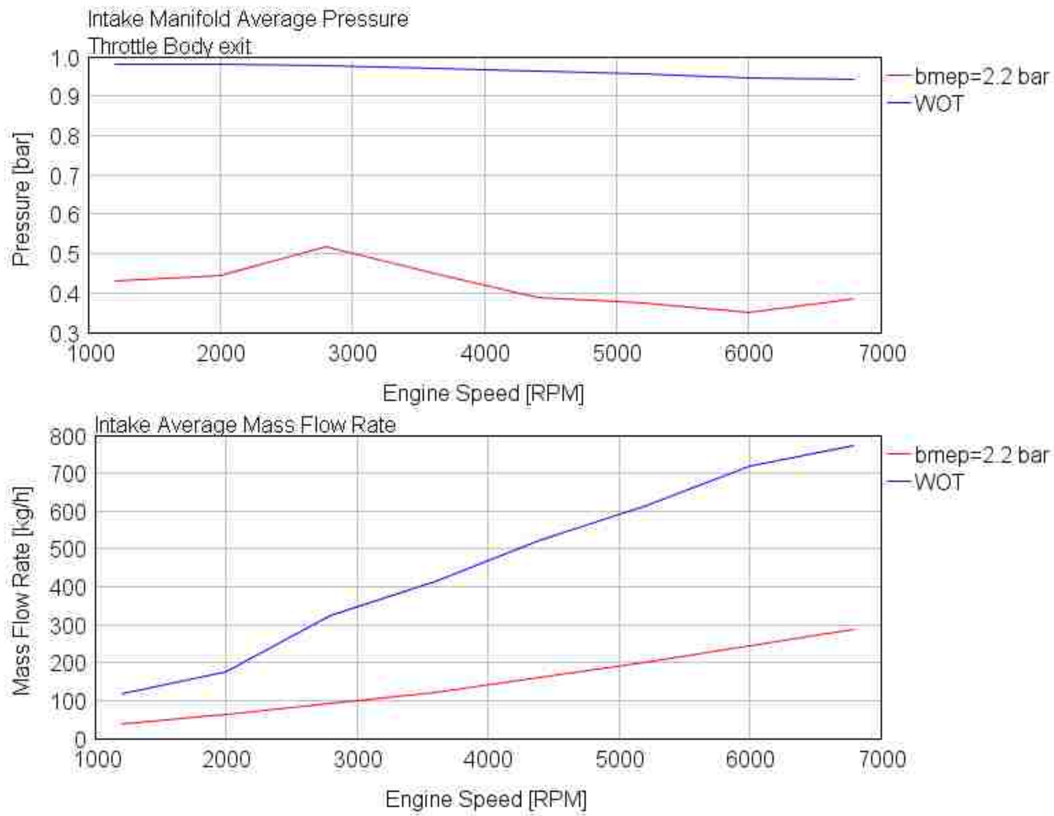

Figure 5.1-3: Intake Manifold Average Pressure and Mass Flow Rate.

The pressure level in the manifold dictates the differential pressure available to scavenge the crankcase gases; considering that the crankcase pressure is close to the atmospheric value, there will be an high differential pressure in Case 1 and a very small in Case 2. In WOT condition the PCV valve should be fully opened, as a consequence.

A greater intake manifold pressure also means greater mass flow rate inside the cylinders and a greater mass of fuel can be burned; greater energy can be released in the combustion chamber and the peak pressure is higher. This is shown in Figure 5.1-4.

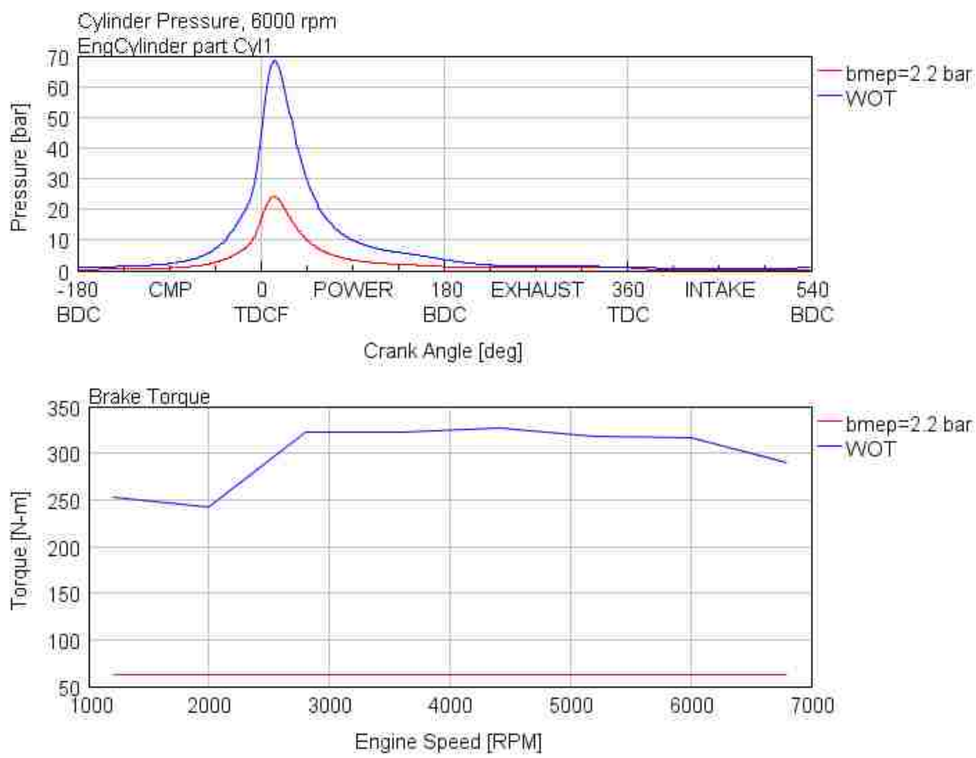

Figure 5.1-4: Cylinder Pressure (6000 rpm) and Brake Torque.

Case 2 has a much greater cylinder pressure than Case 1; as a consequence the differential pressure between cylinder and crankcase is greater and the blow-by generation is expected to be greater; this is not true at all the engine rotational speed because, as said before, it is difficult to forecast the blow-by generation. In some cases, at high rotational speed, it has been observed a greater blow-by generation in part throttle respect to WOT conditions.

The crankcase fluid considered has been modeled with a FluidMixtureBurned object; the inputs to this object are the mass fraction of fuel and air, so basically the air fuel ratio; the object does not use the inputs directly for the initialization but it calculates the products of combustion. As starting point, the fluid in the crankcase was initialized only with burned gas, but during the simulation fresh air enters the crankcase through the make-up –air circuit.

The brake power and torque curve for the WOT condition are shown in Figure 5.1-5: it can be noticed that the trace and the values are quite similar to the published values, depicted in Figure 4.1-1, with only small variations. The results from the model are slightly lower than the published data: the main reason for that is related to the external conditions which were set for the model (see Table 5.1-1). The external pressure was imposed to be much lower than the atmospheric value to simulate a mountain driving condition and the external temperature was quite high (303 K), typical of summer season. This makes the air density in the intake manifold to be lower than the one at standard condition and so the power output is less than the published data. From the model, the maximum engine power at 6350 rpm was 278 HP, while the published value is 283 HP.

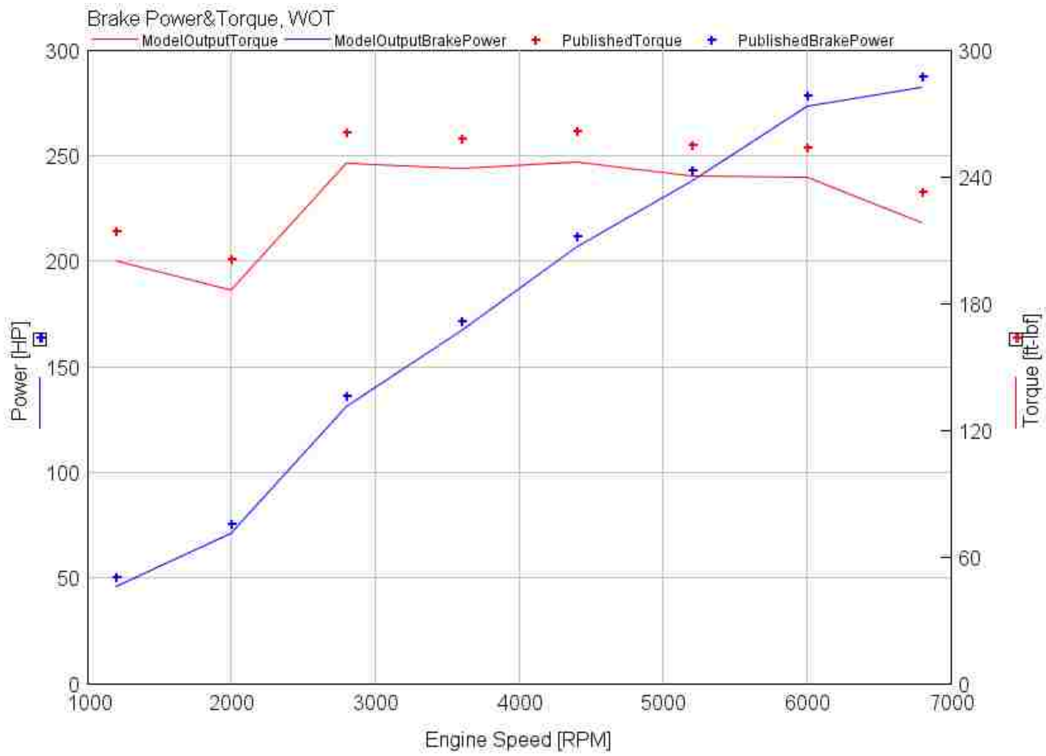

Figure 5.1-5: Brake Torque and Brake Power, WOT.

5.2 Crankcase pressure control

In this and in the following sections the most important parameters affecting the pumping losses inside the crankcase are evaluated with the results provided from the GT-Power model.

In an engine with a V configuration each crankcase bay has 2 cylinders facing it; this means that for a V6 there will be 3 crankcase bays and 4 for a V8. The crankcase bay pressure is influenced by the motion of 2 pistons; depending on the phase shift between them and on the instantaneous crankangle, the pistons can move in the same direction (e.g. both toward the BDC) or in opposite direction (e.g. one toward the TDC and the other toward the BDC).

The pressure inside the crankcase is function of the crankangle. As a rule of thumb, when the piston is moving downward the pressure is increasing because the volume is reducing and the opposite happens when the piston is moving upward. This generates pumping losses which reflect in power losses at the shaft. In a V engine the pressure in the crankcase is influenced by both the pistons; moreover, the flow though the breathing area, through the windage tray and through the oildrains influence the pressure level.

The cylinder volume under the piston is described in the GT-Power model with the object EngCrankcase. The lower part of the crankcase instead has been discretized with flowsplit objects and named CylBays.

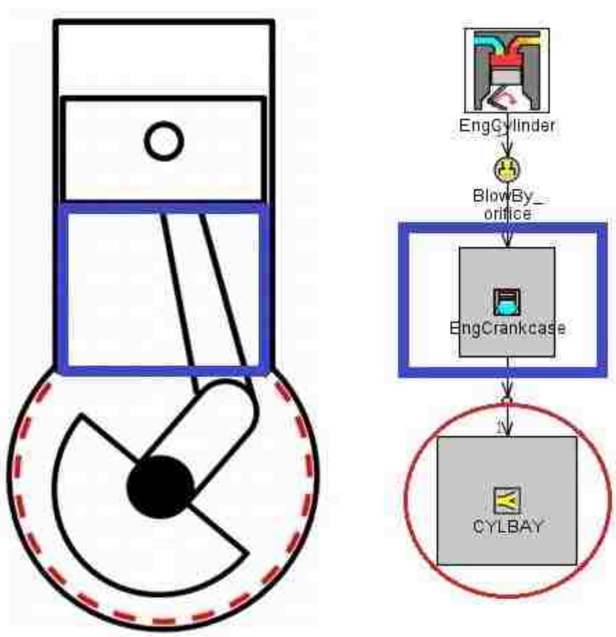

Figure 5.2-1: EngCrankcase and CylBay objects.

For the V6 configuration considered there are 2 EngCrankcase objects for every CylBay, and 3 CylBays; each CylBay is then further split in three smaller volumes for accuracy's improvement. In the following paragraphs, the volume under the piston included in the cylinder volume will be called EngCrankcase, referring to the EngCrankcase object; the crankcase bay (housing of the crankshaft) is instead called cylinder bay or crankcase bay or crankcase, referring to the CylBay object.

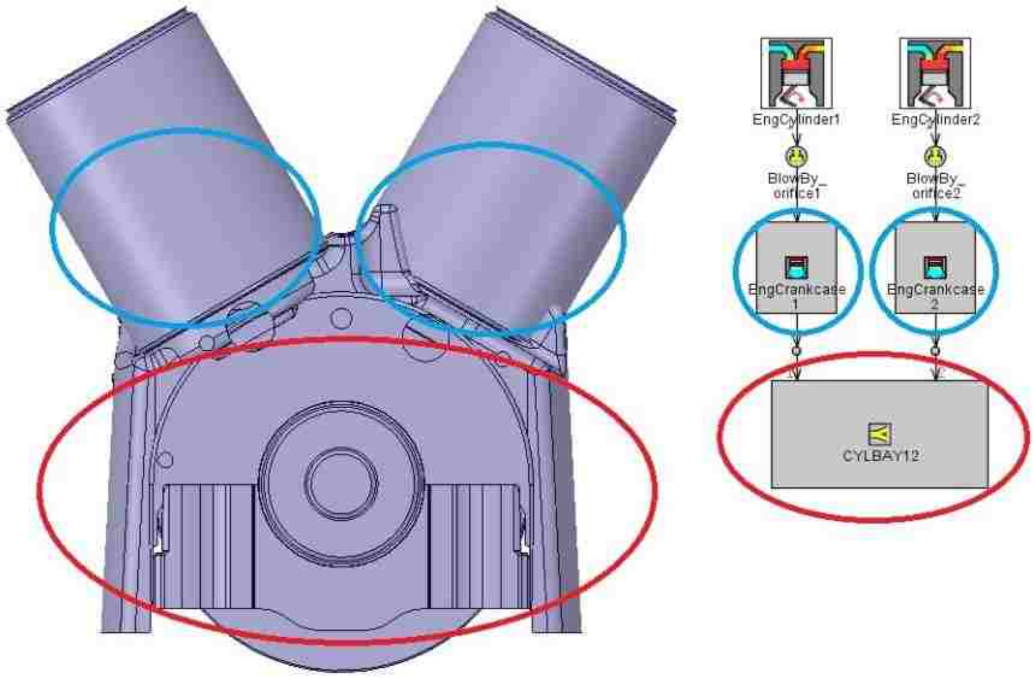

Figure 5.2-2: V6 configuration.

The results coming from the model run at 6000 rpm, in Case 1 (part throttle) show the pressure trends for all the EngCrankcase objects; the pressure trends are plotted below.

The plots in Figure 5.2-3 are results from the model of the real engine; the geometries, the size of the whole breathing area and all the other parameters are exactly the same of the engine considered; the fluid used is only a mixture of combustion products, without considering oil vapor or oil droplets.

In the next paragraphs, some geometries and fluid parameters will be changed, to see the related effect.

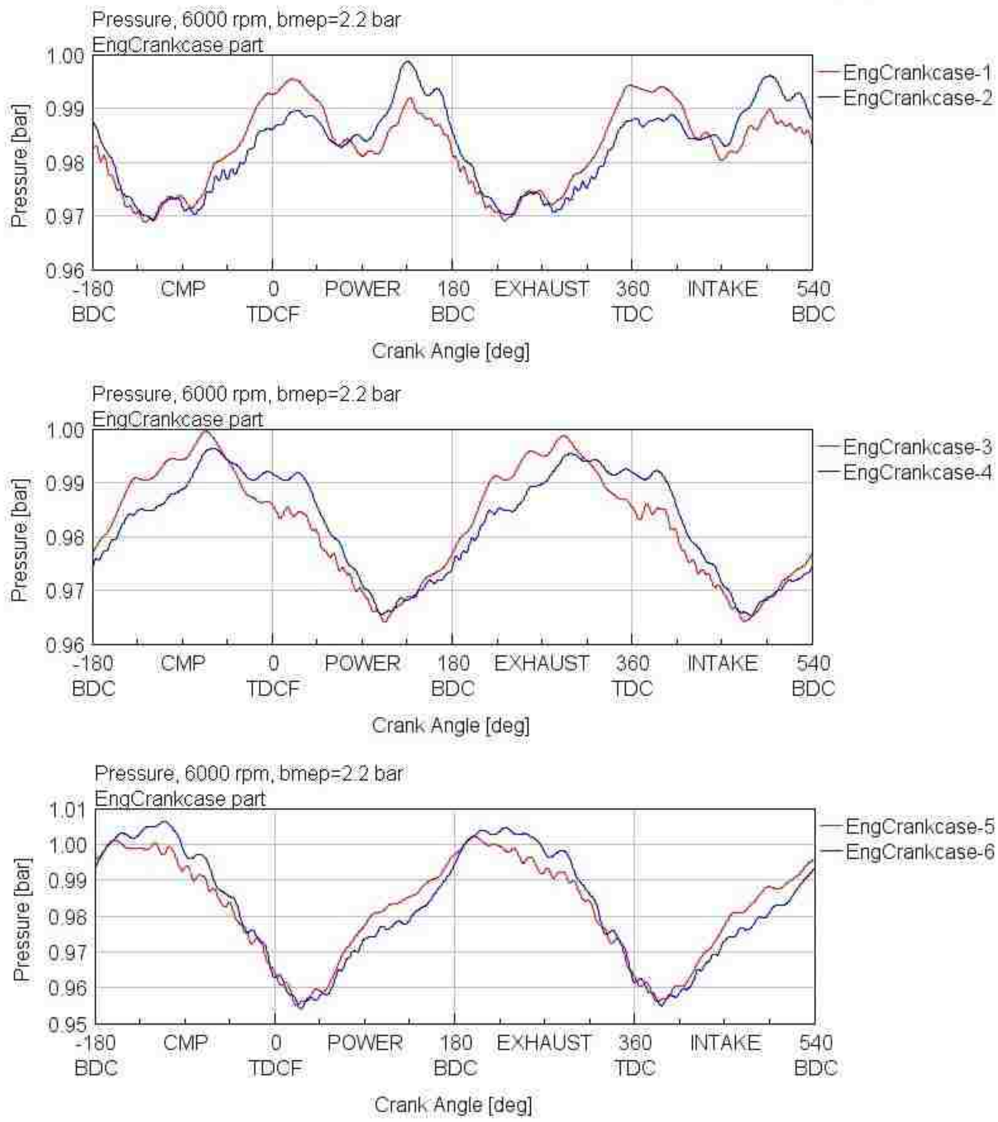

Figure 5.2-3: Pressure trace for EngCrankcase objects.

It can be seen that pressure variation as function of crankangle has a general periodic behavior, caused by the reciprocating motion of the pistons, but quite different from the ideal sinusoidal trace. As said before, the pressure is also influenced by the gas flow through the breathing area (bay to bay holes, windage tray holes, area under main bearing caps and windage tray side areas…) and through the front cover volume and oildrains; this can explain the difference from the ideal pressure profile showed in Figure 3.3-1. In particular, cylinders 5&6 have the pressure variation which is closer to a sinusoidal function respect to the other EngCrankcase objects; these objects also have the greatest pressure amplitude of variation due to a very small breathing area, so the gas flow is greatly hindered and pressure amplitude of variation is high. Cylinders 3&4 have a greater breathing area and the pressure variation is smaller. Cylinders 1&2 are the closest to the front cover volume and have a large breathing area so the pressure oscillations are lower. The pressure trace is quite different from a sinusoidal function.

Pressure amplitude of variation for the three CylBay objects as function of engine speed is plotted in Figure 5.2-4.

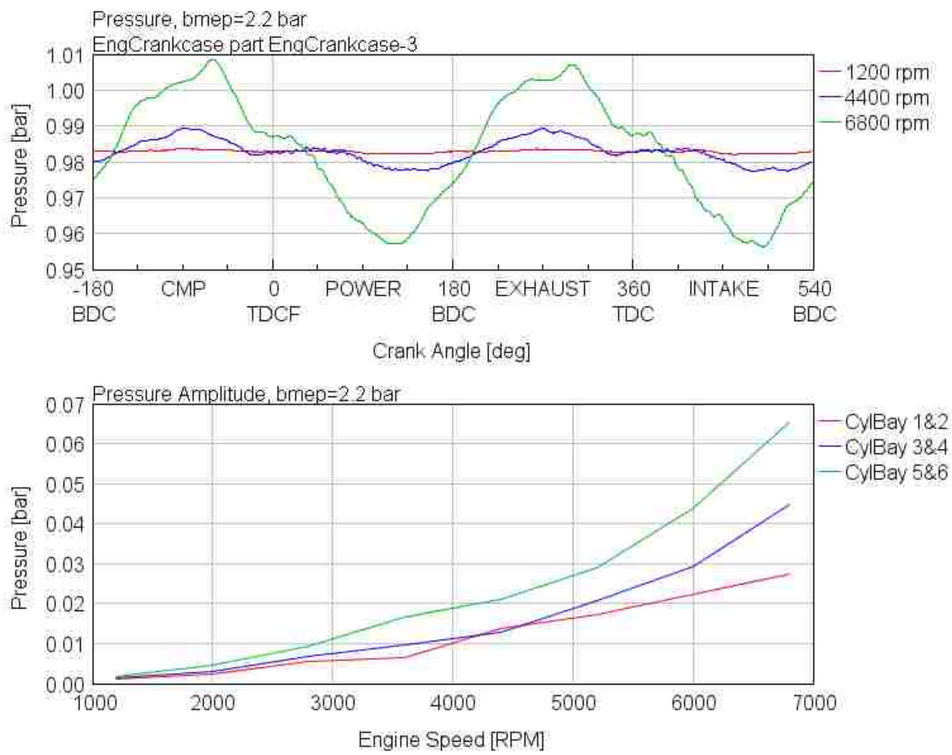

Figure 5.2-4: Pressure in the EngCrankcase 3 and Pressure Amplitude in the CylBays.

The pressure variation is increasing with the speed of the engine because there is less time for the gas flow movements. Pressure variation in the crankcase is essentially released by the bay to bay flow and at high speed the time available for the gas to enter or exit the CylBay is lower, resulting in greater pressure amplitude.

Just as an example, consider cylinder 3; in Figure 5.2-5 the EngCrankcase3 pressure and volume are plotted together. The volume (red curve) follows a sinusoidal trend, from a maximum value when the piston is at TDC and a minimum value when it is at BDC. Instead, the pressure (blue curve) has a more irregular shape, even if roughly sinusoidal, and seems to be a bit shifted respect to the volume curve.

The pressure in the volume under the piston is not only influenced by the piston over it, but also by the motion of the piston on the other engine bank and by the gas flow through the breather area. This can generate peak pressure also when the volume is maximum (as can be seen from the first graph) and vice versa; the situation is the opposite respect to the one described in Figure 3.3-1 in which the pressure was maximum at minimum volume and vice versa. That resulted in zero pumping work because all the work lost in compression was recovered in expansion (Figure 3.3-2).

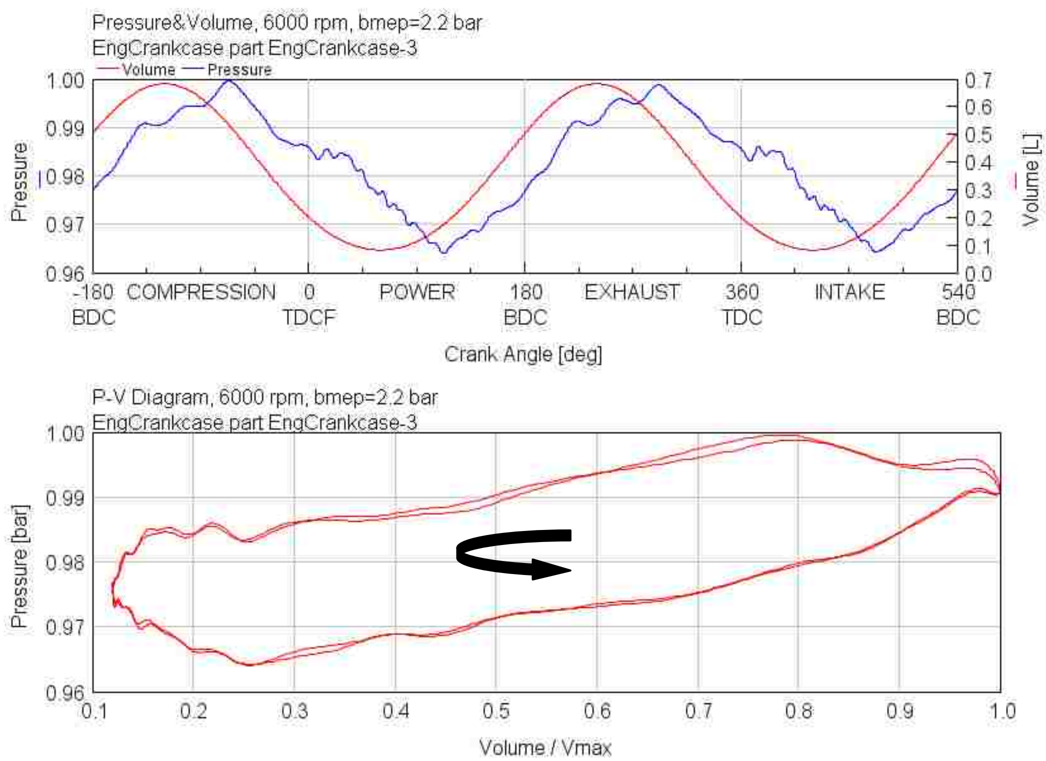

Figure 5.2-5: Pressure and Volume diagrams for EngCrankcase3.

Instead, in real applications, pumping losses are present and the CPMEP can be computed.

In the P-V diagram the trace moves in counterclockwise direction and so the area inside the trace represents the pumping work losses. The work is negative so it is work subtracted to the crankshaft output.

In GT-Power, the work in the crankcase is not computed with integration over a whole engine cycle like Equation 3.16 (from 0° to 720°) but only with integration over one revolution (from 180° after TDC to 180° before the next TDC). This makes the result to be half the one computed for a whole engine cycle. Moreover, differently form Equation 3.16, the software computes the CPMEP for a single cylinder dividing the crankcase work by the EngCrankcase displacement instead of the cylinder displacement. For cylinder 3 the CPMEP becomes:

$$W_{EngCrankcase3} = \int_{180}^{-180} p_{EngCrankcase3} \cdot dV_{EngCrankcase3} \tag{5.8}$$

$$\text{CPMEP}_3 = \frac{W_{EngCrankcase3}}{V_{EngCrankcase3}} \tag{5.9}$$

In the previous equation, $V_{EngCrankcase3}$ is the volume of the EngCrankcase3 object when the piston is at TDC.

The CPMEP is negative, because the work is negative and this means losses.

The trend of the CPMEP, computed for cylinder 3 is displayed in Figure 5.2-6.

As expected from Equation 3.42, the CPMEP is proportional to the square value of the engine speed; this explains the parabolic trend of Figure 5.2-6. Maximum pumping losses are generated at maximum speed.

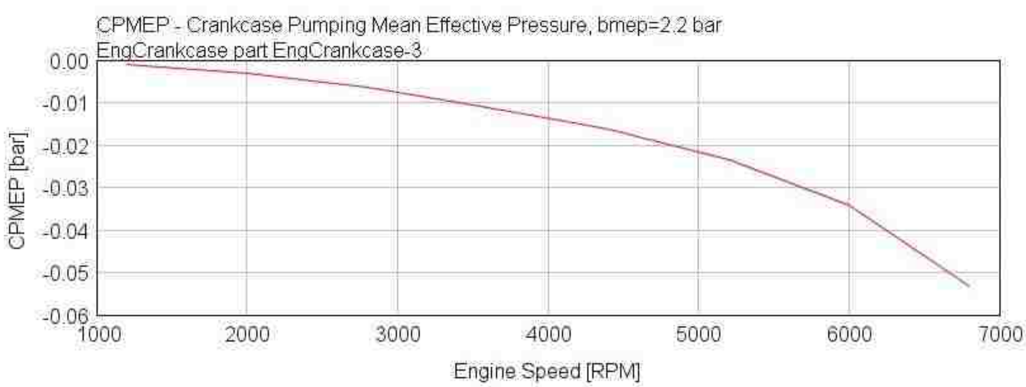

Figure 5.2-6: CPMEP for cylinder 3.

For the considered cylinder, at 6800 rpm, the CPMEP is close to 5500 Pa which is 2.5% of the considered *bmep*.

The crankcase pumping losses are work which is subtracted from7 the shaft. For the energy conservation, this energy is conserved and the work is converted into heat; the heat transfer to the cylinder walls and engine block rises the temperature of these components [29], [32].

In particular, if the oil pan temperature increases, also the oil temperature is greater and the oil cooling requirements of the engine are greater. As a consequence, greater effort is requested from the cooling system.

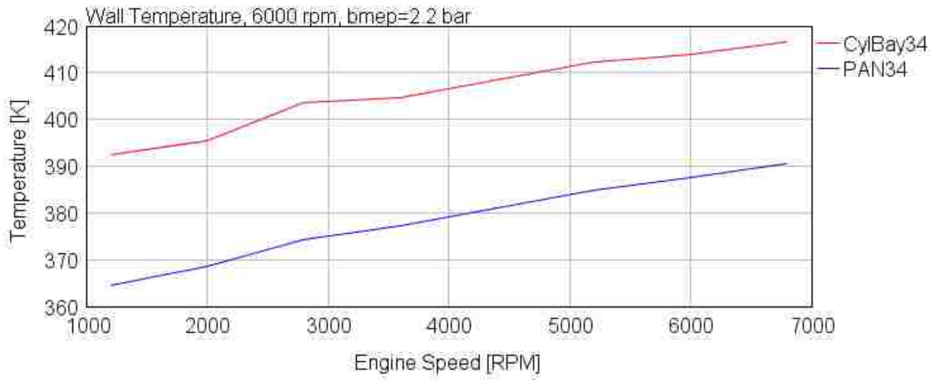

Figure 5.2-7: Wall Temperature.

Considering just a single cylinder does not explain what really happens in the whole crankcase, especially in an engine with a V configuration. In such a type of engine, for every cylinder there is one common portion of CylBay and the pressure here is influenced by the motion of both pistons (together with the gas flow through the breathing area). This means that, while one piston is subject to resistance in its motion due to the crankcase fluid, the other can effectively gain work. In the considered engine the phase shift between one cylinder and the neighbor is 120° and the pressure generated by the downward motion of one piston can positively help the piston in the other bank. The flow through the breather area, generated by the motion of the pistons can also produce a positive work on the pistons on the next bay.

Previously EngCrankcase3 has been considered; looking now at the EngCrankcase4, which shares the same CylBay34, the situation is a bit different.

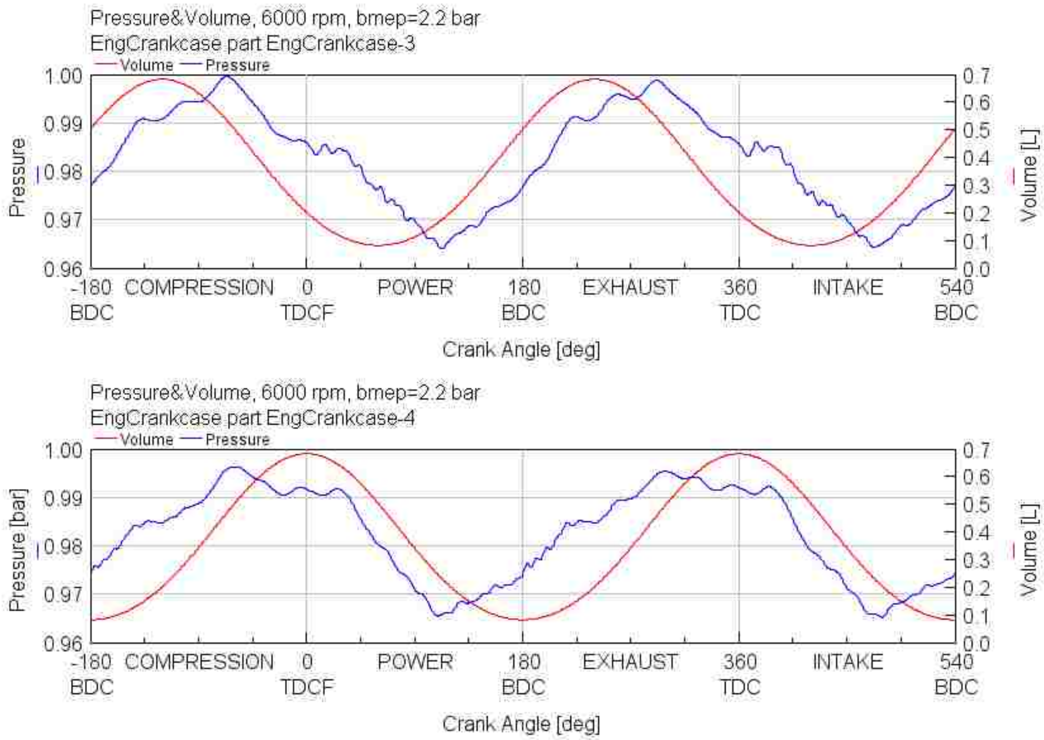

Figure 5.2-8: EngCrankcase3&4, Pressure&Volume.

As can be seen the volumes (red curves) follow a sinusoidal trend, from a maximum value when the piston is at TDC and a minimum value when it is at BDC. The second red curve is shifted of 120° respect to the first one, because that is the phase shift between the two pistons.

The pressures (blue curves) are very similar because both of them communicate with the large CylBay34 under them but pressures do not follow exactly the expected sinusoidal trend. The P-V diagrams for the two cylinders are different due to the phase shift and for the same reason cylinder 4 presents different pumping characteristics; even if, in a given time instant, the pressure values are comparable with the cylinder 3 values, the volume is different. This can be seen in the P-V diagrams below.

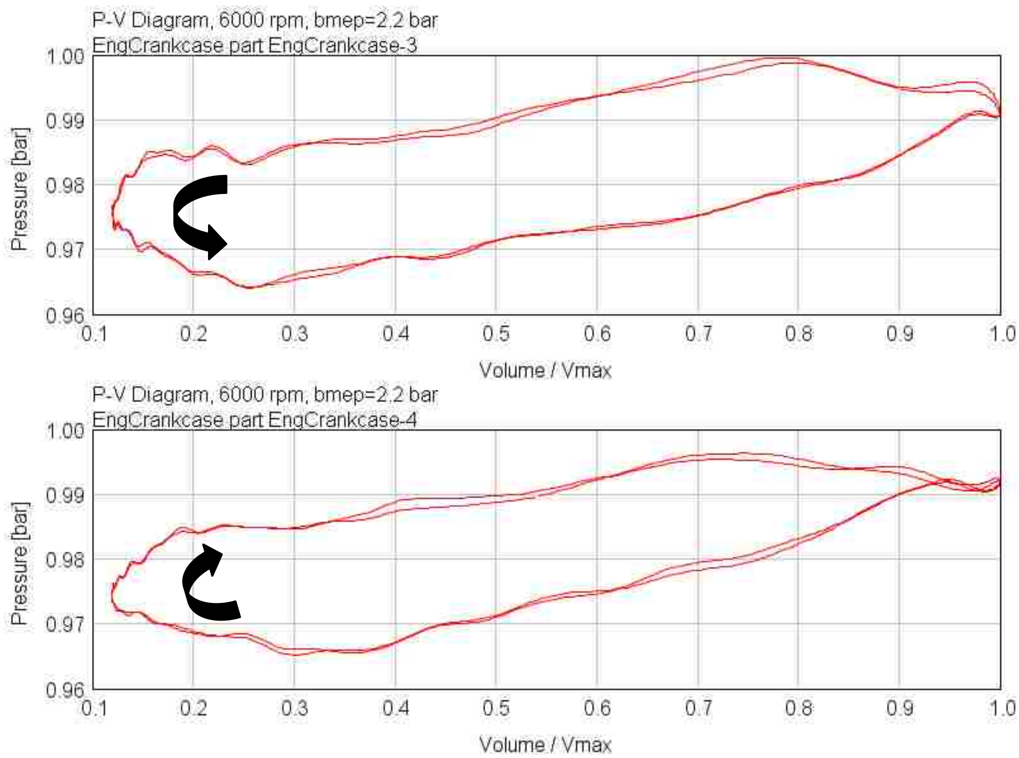

Figure 5.2-9: P-V Diagram for EngCrankcase3&4.

The first diagram is relative to the cylinder 3; the trace moves in counterclockwise direction and so the area inside the trace represents the pumping work losses. The work is negative so it is work subtracted to the crankshaft output. Instead, the P-V trace relative to the cylinder 4 moves in clockwise direction and this means that the work is positive (green). Figure 5.2-10 shows the pumping work for the 2 cylinders.

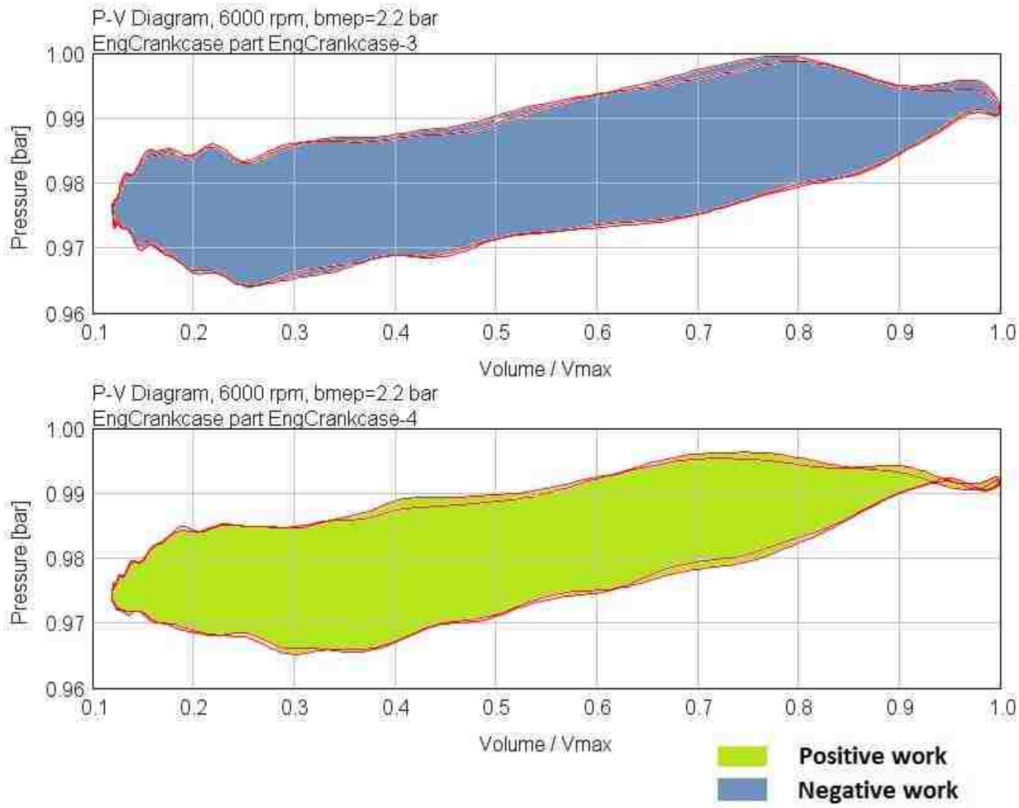

Figure 5.2-10: Pumping works.

Now, recovering Equation 5.8 and 5.9 the crankcase pumping mean effective pressure can be computed. Again, the software computes the work only for one crankshaft revolution and dividing the work by the EngCrankcase displacement. For cylinder 4 the equations are:

$$W_{EngCrankcase4} = \int_{180}^{-180} p_{EngCrankcase4} \cdot dV_{EngCrankcase4} \tag{5.10}$$

$$\text{CPMEP}_4 = \frac{W_{EngCrankcase4}}{V_{EngCrankcase4}} \tag{5.11}$$

Considering cylinder 3, it is clear that the CPMEP is negative, because the work is negative and this means losses; this is not true for cylinder 4, where the crankcase work is positive, as shown in Figure 5.2-10. This can be explained considering the reciprocating motion of the pistons, which are shifted of 120°, and with the flow through the breathing area which can positively help the piston motion.

The trend of the CPMEP is displayed in Figure 5.2-11. As can be seen, EngCrankcase3 is subjected to power losses, while EngCrankcase4 is subjected to power gains.

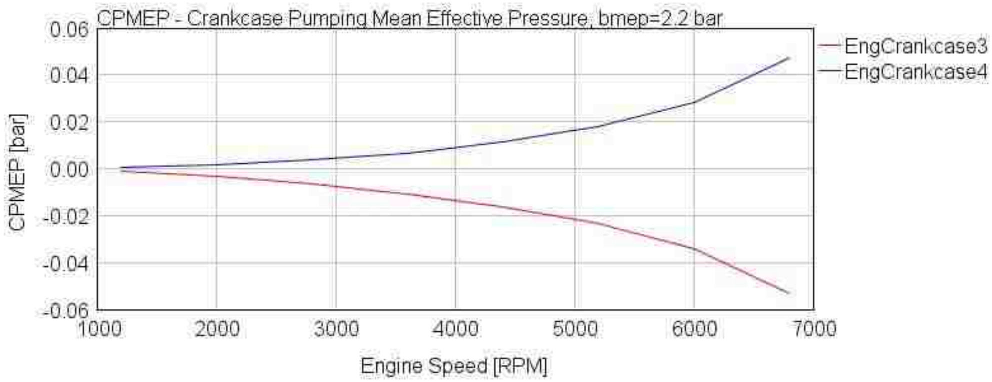

Figure 5.2-11: CPMEP for cylinder 3&4.

The plot of Figure 5.2-11 can be redrawn considering all the cylinders and their average; Figure 5.2-12 shows how some cylinders have positive crankcase work, while others have negative.

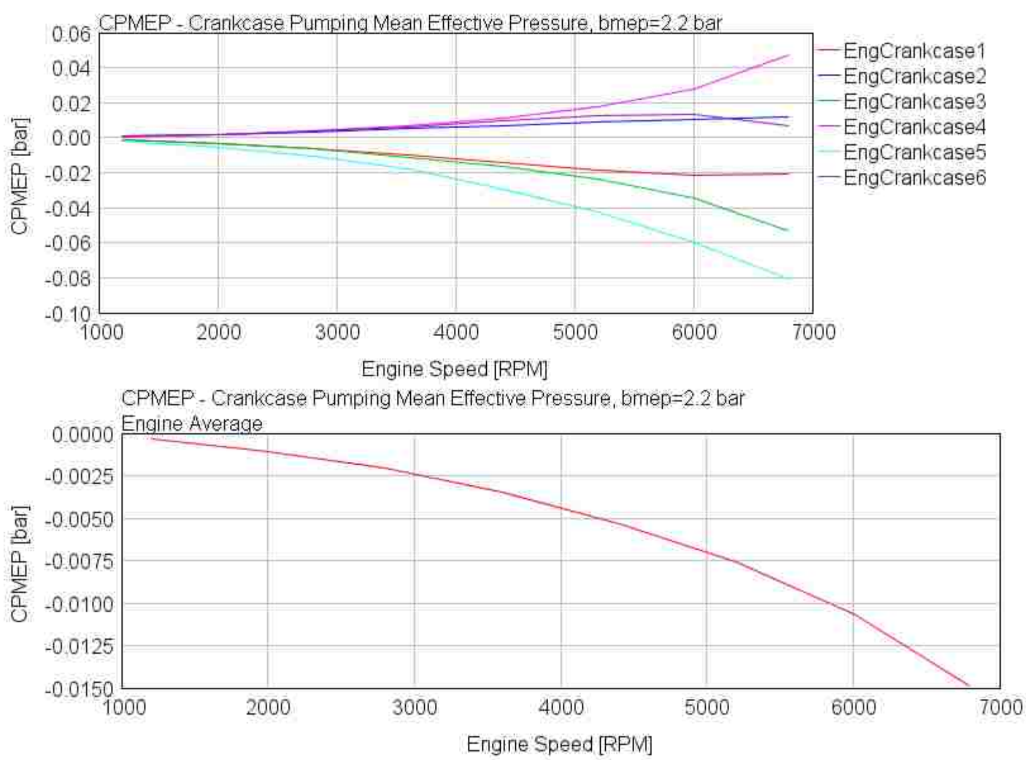

Figure 5.2-12: CPMEP for all the cylinders and Engine Average.

On average the negative work is greater, as expected, due to flow losses. The average value of CPMEP is plotted in the second graph of Figure 5.2-12; it follows the parabolic trend predicted by Equation 3.42. The values of CPMEP are very low, due to the fact that only exhaust gases were considered in this simulation and the fluid density is very low.

From the CPMEP, the power lost in the crankcase flow motion can be computed with Equation 5.12.

$$\dot{W}_{crankcase} = \frac{W_{crankcase}}{t_{rev}} \tag{5.12}$$

Differently from Equation 3.34 and 3.37, here the time required for one crankshaft revolution t_{rev} is considered instead of the cycle time, because the crankcase work $W_{crankcase}$ was previously integrated only for 360° instead of 720° (see Equation 5.8 and 5.10). The final results, considering the case at 6800 rpm is:

$$\dot{W}_{crankcase} = \text{CPMEP} \cdot \frac{V_{crankcase}}{V_u} \cdot V \cdot \frac{rpm}{60} =$$

$$= 1500\ Pa \cdot \frac{0.683\ l}{0.6\ l} \cdot 3.6\ l\ \cdot \frac{6800}{60} = 0.651\ kW = 0.872\ HP \tag{5.13}$$

The computed power loss is quite small because the CPMEP is small when only exhaust gases are considered as crankcase fluid. In paragraph 5.5 it will be shown how the CPMEP deeply changes considering also oil vapor and liquid oil in the crankcase.

Moreover, in a 1D simulation, the rotation of the crankshaft and counterweights is not considered so the flow field generated by their rotation and the drag on that elements is not taken into account, leading to a low value of power losses.

5.3 Effect of load

The load effect on the crankcase ventilation system has been evaluated comparing Case 1 and Case 2. The basic difference between those two cases is the intake manifold pressure and the cylinder pressure. The first one is important because it determines the pressure difference which is available in order to scavenge the crankcase. The second one has its main effect on blow-by production.

The flow through the PCV valve is related to the differential pressure between crankcase and manifold. In Case 1 the valve is nearly closed because the mass flow rate should be small and the differential pressure is high; in Case 2 there is low differential pressure to scavenge the great blow-by mass flow rate, so it is fully opened.

The system was correlated with blow-by experimental data coming from a real engine; according to them, the blow-by production characteristics were implemented in the model, imposing a fixed diameter of the blow-by orifices and a coefficient of discharge which was speed and load dependent. The blow-by volume flow rate coming from experimental data is shown in Figure 5.3-1; the scale on the vertical axis is not present because that is confidential material from Chrysler LLC.

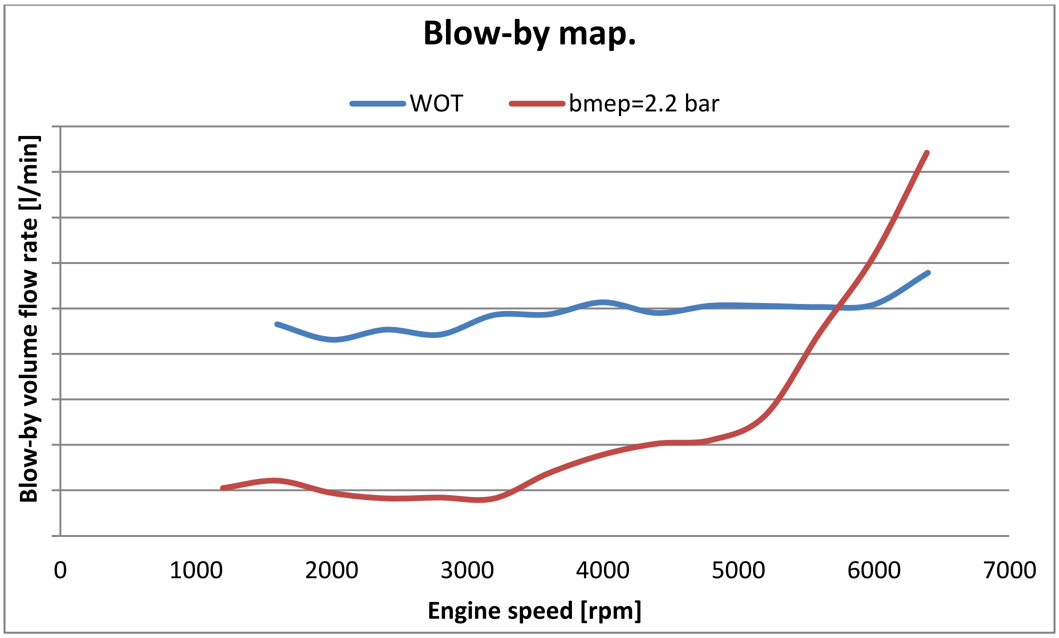

Figure 5.3-1: Blow-by map.

In Figure 5.3-2 there is the result from the model. On that graph and on the following two there are no numbers on the vertical axis, because these are confidential information. The traces are very close to the traces coming from experimental data, so the model is correctly representing the ventilation characteristics of the engine from the blow-by point of view. The difference between model output and experimental data is less than 3% of the experimental value.

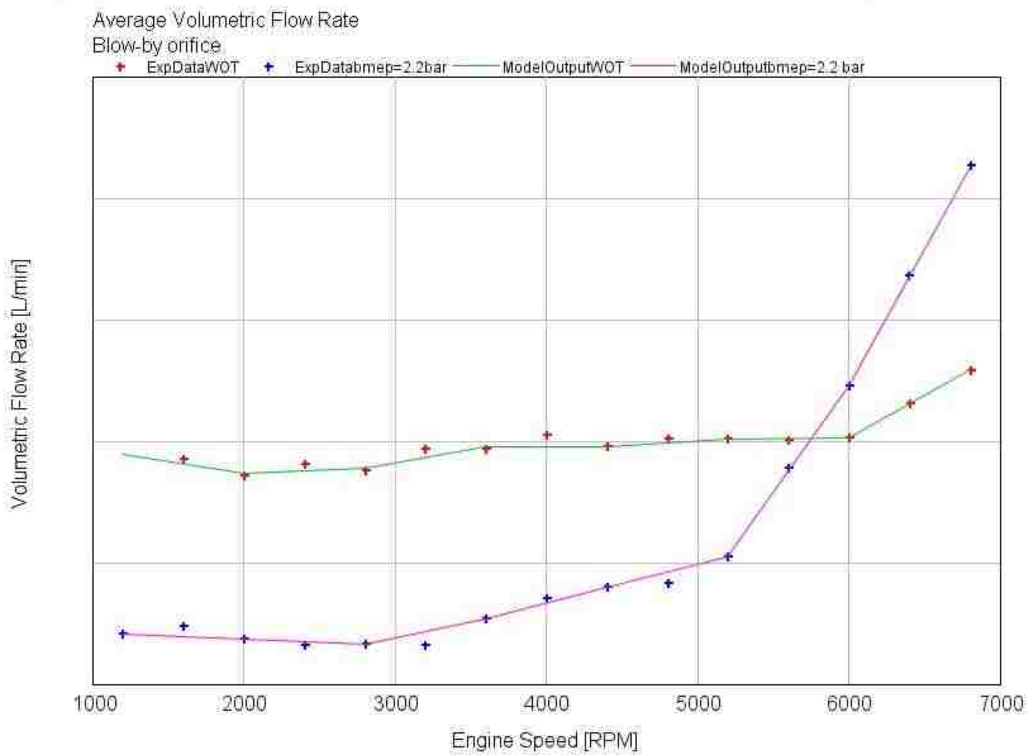

Figure 5.3-2: Blow-by volume flow rate for the two cases.

It can be seen that the blow-by production has a trend which is very difficult to be predicted; as a rule of thumb, greater in cylinder pressure generates greater differential pressure between cylinder and crankcase and so greater blow-by flow. Looking at Figure 5.3-2 it is not always true, especially at high engine speed and low load, where piston rings instability is greater and the sealing characteristics are lower; at WOT and high speed, the higher cylinder pressure improves the sealing of the piston rings compared to part throttle condition.

The flow through the PCV valve is only function of the differential pressure between intake manifold and crankcase, so it is independent of engine speed. This flow reflects the real flow in the PCV valve because it matches the experimental data. Considering that the crankcase pressure is always close to atmospheric pressure, there are two different PCV flows, one for Case 1 (part throttle, manifold absolute pressure close to 40 kPa) and one for Case 2 (wide open throttle, manifold absolute pressure close to atmospheric value).

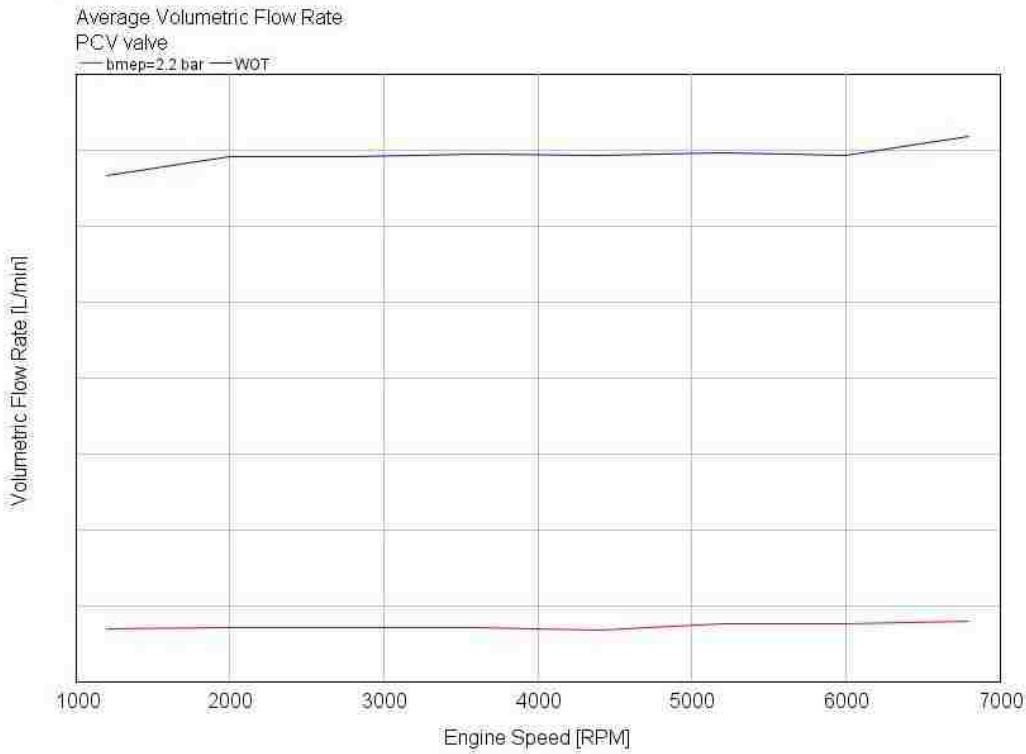

Figure 5.3-3: Flow through the PCV circuit for the two cases.

The PCV flow for a given case is constant in the engine speed range; considering the blow-by production characteristics of the engine, the flow through the make-up air circuit can be predicted. For the conservation of the mass, the mass flow rate entering the system should be equal to the one which exits. The majority of the mass is entering the system through the blow-by orifices, while, for a given case, a constant flow is exiting from the PCV circuit. The make-up-air circuit is a two-way circuit, so mass can enter or exit the system depending on engine speed.

In Figure 5.3-4 the flow through the make-up-air circuit is depicted: negative values mean fresh air which enters the system, so from the air box to the cylinder head, while positive values mean the opposite flow, blow-by gases which exit the crankcase and reach the air filter. At low speed fresh air is flowing into the crankcase, because there is a low blow-by production and the flow through the PCV circuit is constant. Increasing the engine speed the blow-by production increases and lower fresh air can enter. At engine speed round 4400 rpm the make-up air flow reverses (i.e. blow-by gases go from valve cover to air box, not air from air box to valve cover) in order to vent the excess blow-by.

At the highest rpm the blow-by flow is vented in the intake side on both the make-up air and PCV circuits.

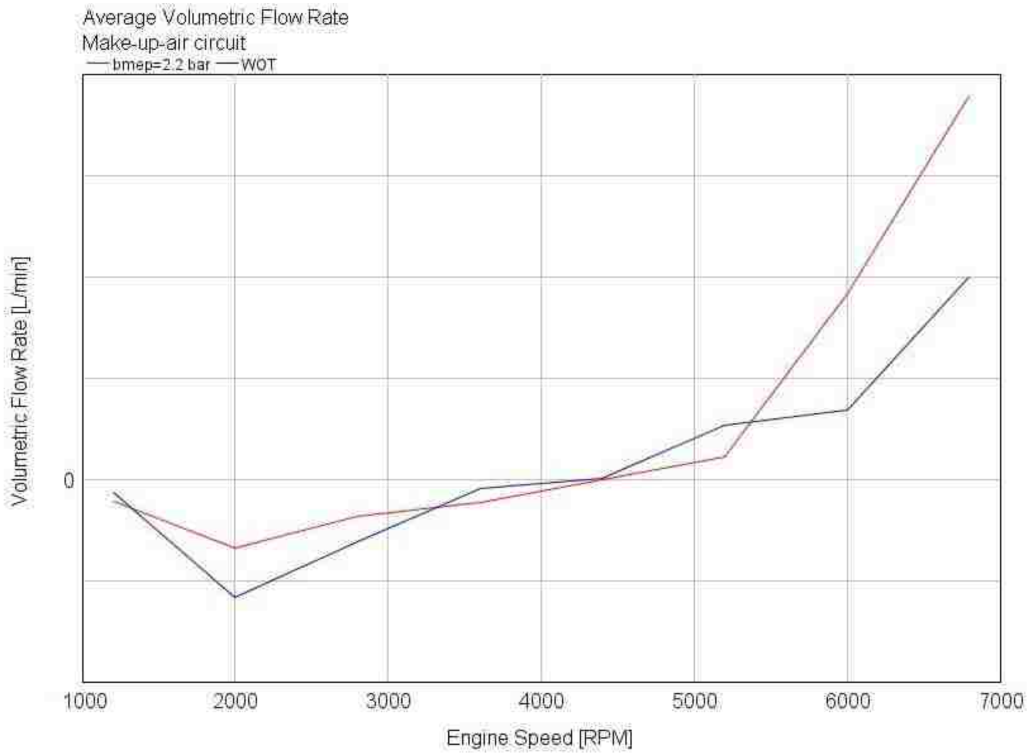

Figure 5.3-4: Flow through the make-up-air circuit for the two cases.

The results from the simulation show that there are not many differences in the crankcase pumping mean effective pressure for the two cases. The two traces, depicted in Figure 5.3-5, are very similar and this means that the load does not have a great effect on the pumping losses inside the crankcase. During part throttle operation the CPMEP is slightly greater than WOT condition because the PCV valve is nearly closed and the restriction increases the flow frictional losses.

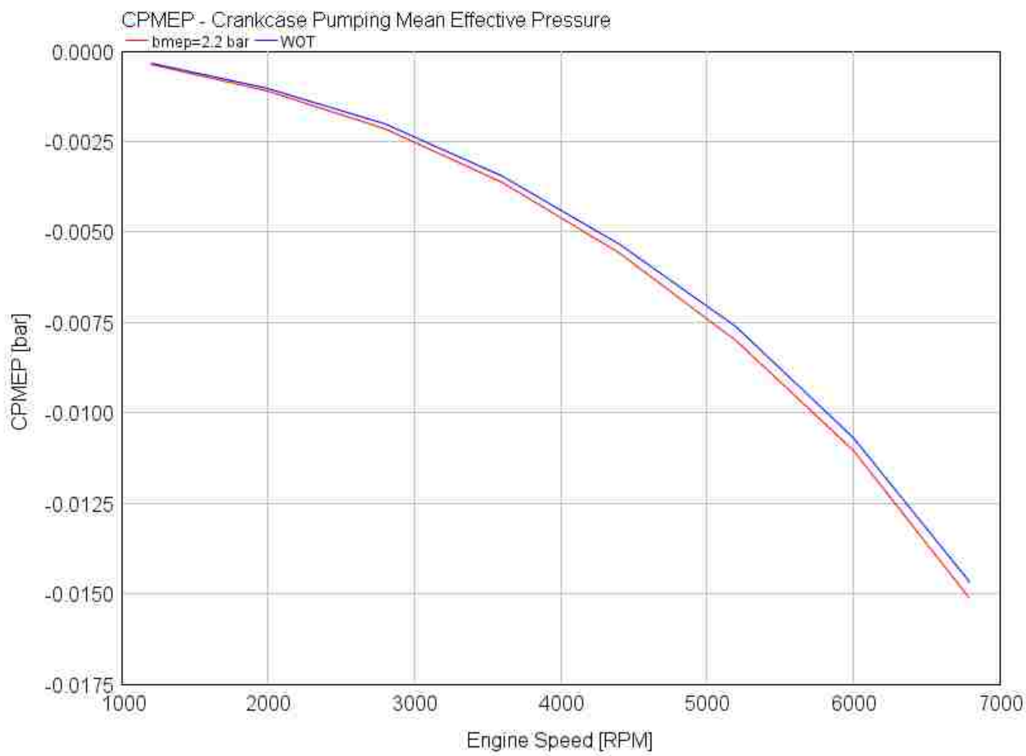

Figure 5.3-5: CPMEP for the two considered cases.

The developed model can be used in the design phase of the ventilation system, in order to design the correct size of PCV and make-up-air circuits for a given blow-by production.

5.4 Breather area

The breathing area (also called ventilating area or breather area) is the available section for gas movements inside the crankcase. It consists of the bay to bay holes, the windage tray holes, the side area around the windage tray and the area under the main bearing caps. These areas allow gases to flow from a high pressure region to a lower pressure region inside the crankcase, thus reducing pressure amplitude.

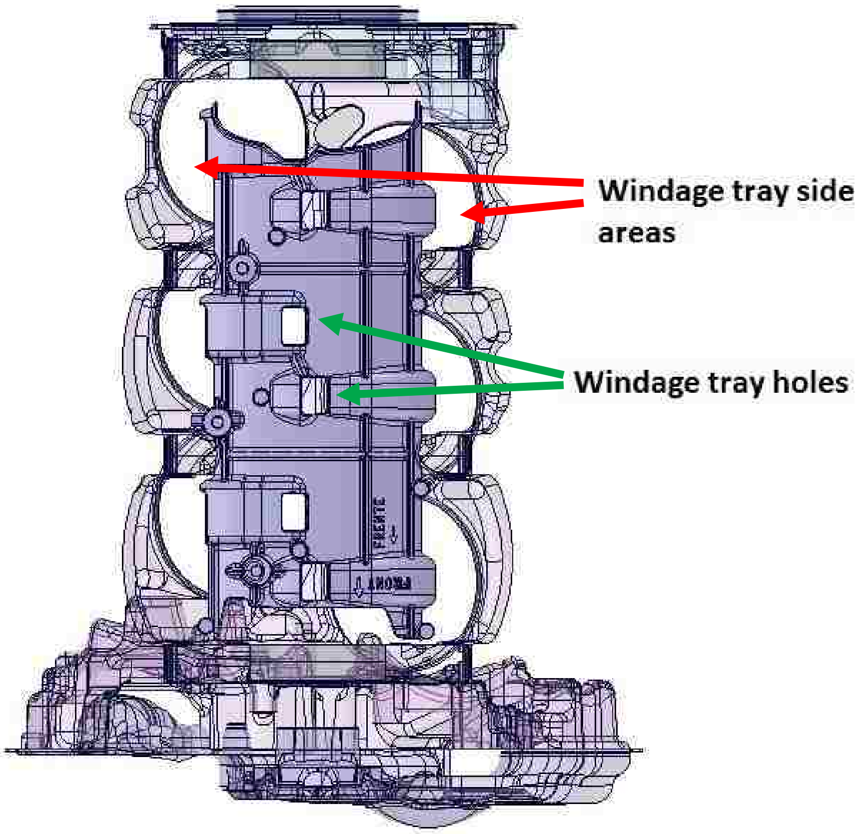

Figure 5.4-1: Windage tray holes.

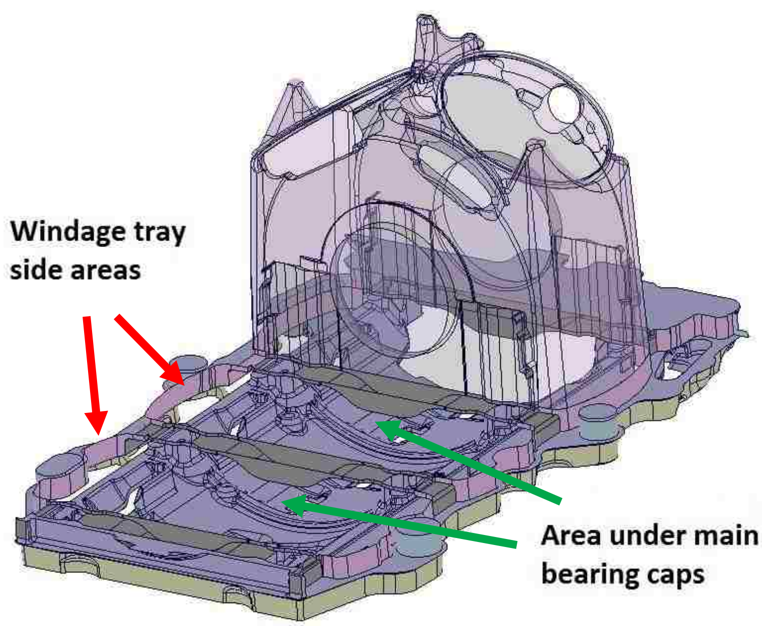

Figure 5.4-2: Area under main bearing caps.

Gas flow means power losses; as a consequence a zero breathing area would ideally result in zero power losses. In order to evaluate the effect of the breathing area, the model has been modified, completely closing the bay to bay holes, the windage tray holes and side areas and the area below the main bearing caps. The closure of the windage tray holes isolates the oil sump volume, which, in this way, is not affected by the reciprocating motion of the pistons. The model was then run again; in this case, considering one single CylBay volume, the gas is compressed and it expands due to the reciprocating motion of the pistons. No gas exchange is possible because the volume is isolated: for EngCrankcase3 and EngCrankcase4 objects, the volume variation as function of crankangle is plotted below.

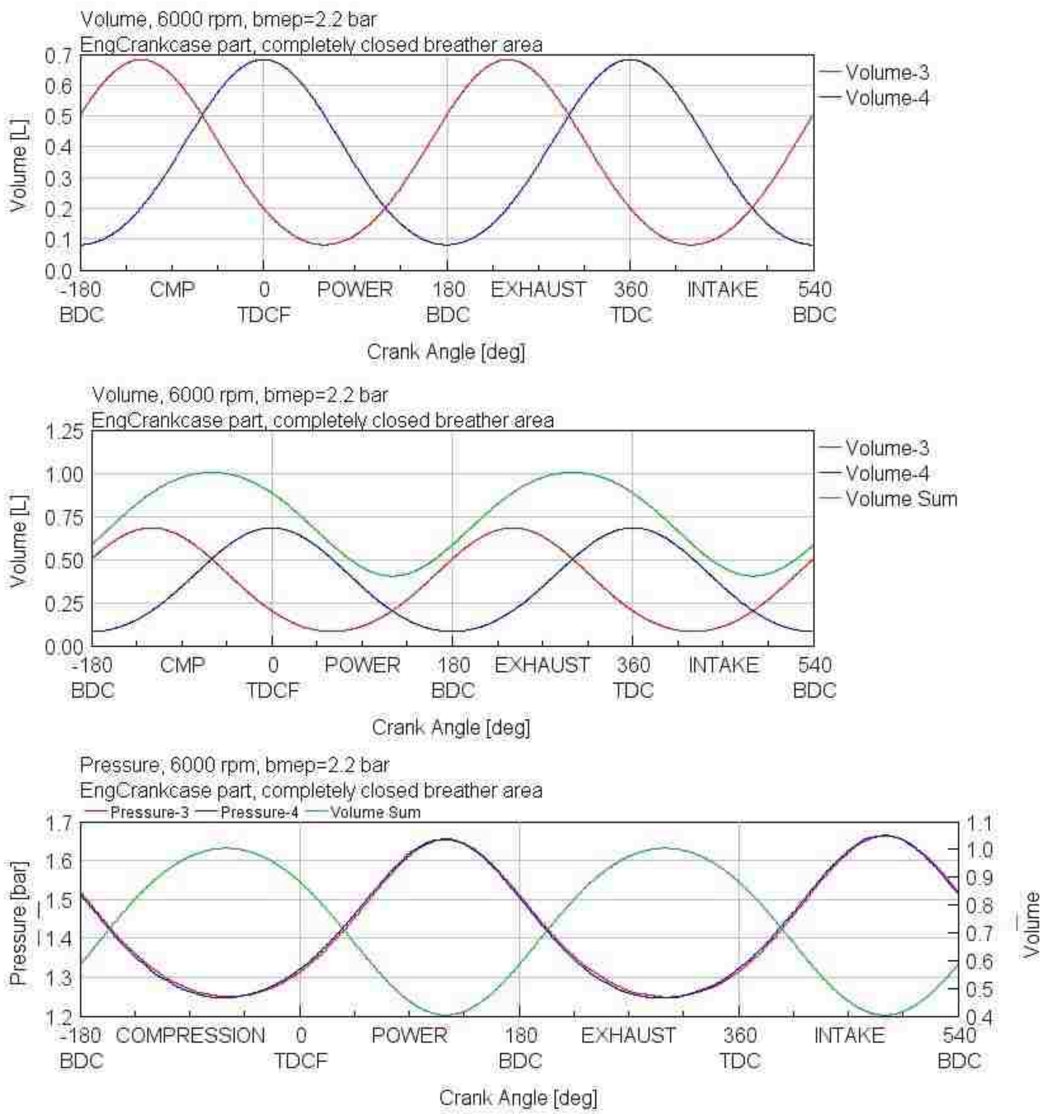

Figure 5.4-3: Volume and Pressure plot for EnCrankcase3 & EngCrankcase4, closed breather area.

The first picture shows the phase shift of 120° between cylinder 3 and cylinder 4 volumes. The red line refers to cylinder 3 while the blue line to cylinder 4. The sum of these two instantaneous volumes is plotted in green in the second graph: it clearly shows how the whole volume under the two considered cylinders changes as function of crankangle. The CylBay34 volume (around 2 liters) under the two EngCrankcase volumes is not considered, because it does not change. The considered volume variation gives rise to two pressure traces which are in anti-phase with respect to the volume trace. The two pressure traces are related to the EngCrankcase objects 3 and 4 and they overlap because they both communicate with the CylBay34 object under them. The last graph is similar to the one showed in Figure 3.3-1, which was analytically predicted; here the simulation result matches the theoretical prediction.

Comparing Figure 5.4-3 with Figure 5.2-3 big differences can be seen in the trace; these are entirely due to the exchange gas flow in the breather area which can shift the point of peak pressure and generate a more irregular trace.

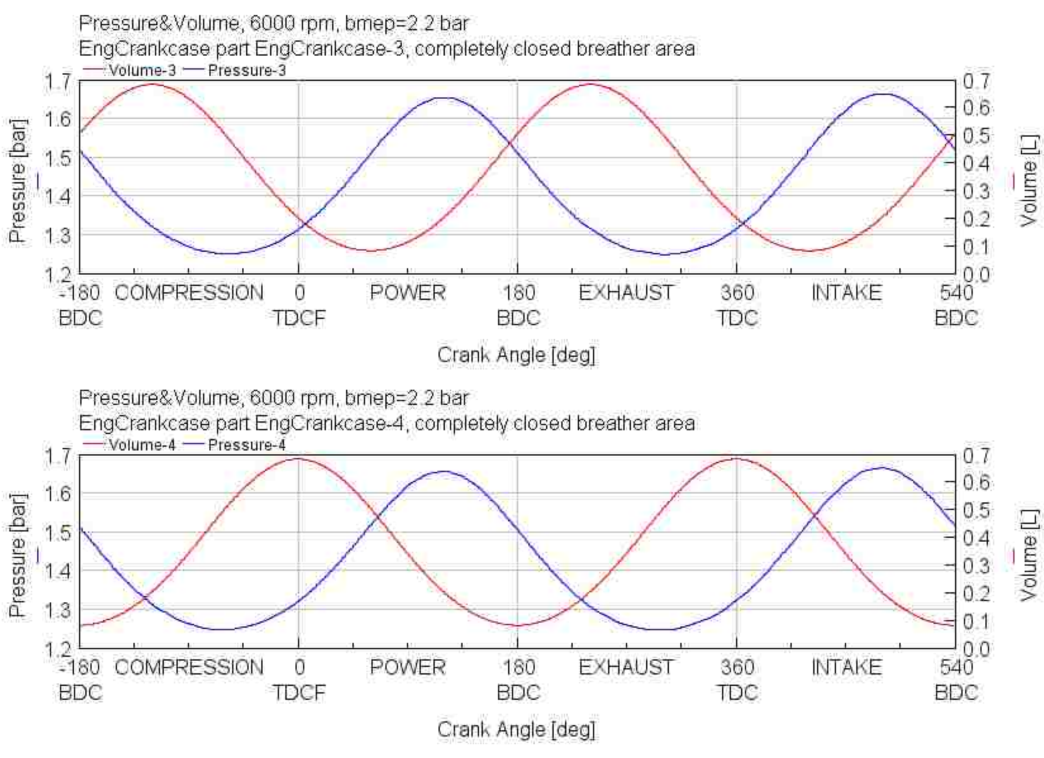

Figure 5.4-4: Pressure&Volume for EngCrankcase3 & EngCrankcase4, closed breather area.

The P-V diagrams of cylinder 3 and cylinder 4 for completely closed breather area are depicted in Figure 5.4-5. Compared to the P-V diagram of the real case (see Figure 5.2-9) with considerable breather area, the trace is much more roundish and regular. Furthermore, EngCrankcase3 is exposed to pumping gains because the trace is moving in clockwise direction, while EngCrankcase4 has pumping losses due to the fact that the trace is moving in counter-clockwise direction. This is exactly the opposite of what happens in Figure 5.2-9: this shows that the breathing inside the crankcase deeply influences its pumping characteristics.

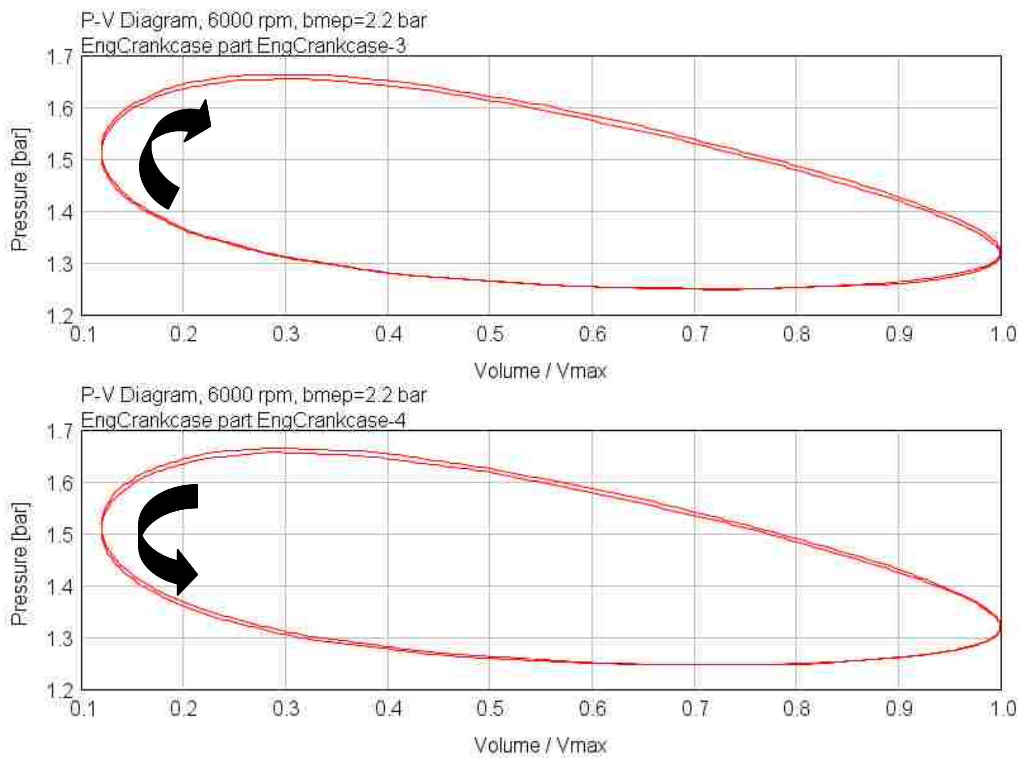

Figure 5.4-5: P-V diagram for EngCrankcase3 and EngCrankcase4, closed breather area.

Figure 5.4-6 shows the CPMEP as function of engine speed both for the EngCrankcase objects of the two considered cylinders and for the whole engine. As can be seen, the CPMEP - Engine Average is very close to zero.

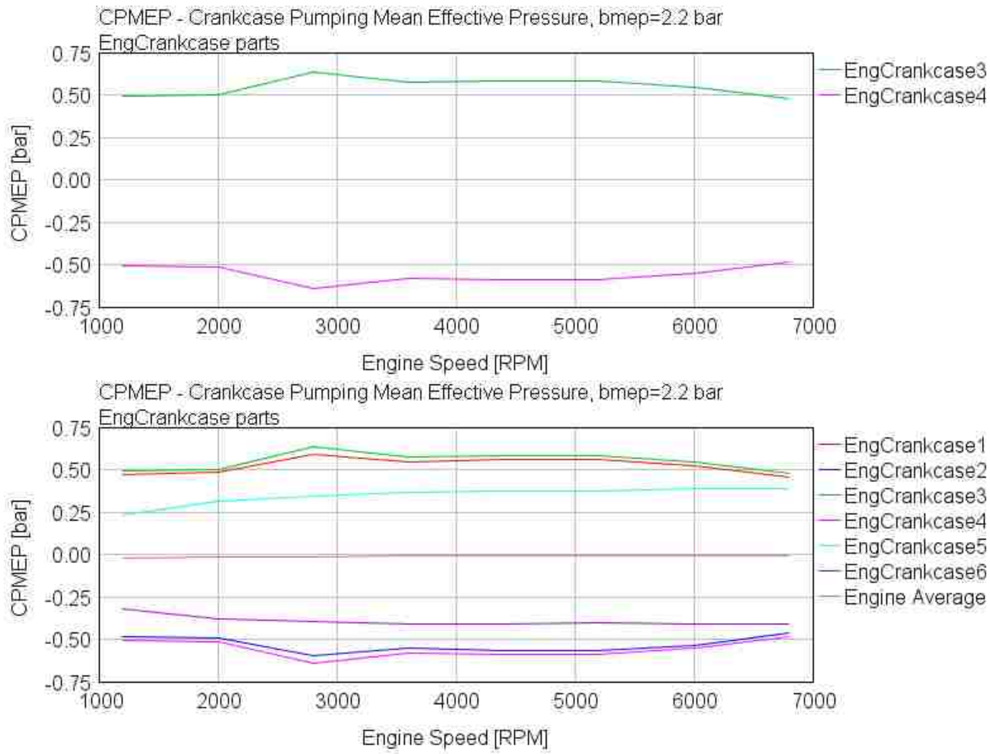

Figure 5.4-6: CPMEP for completely closed breather area.

Theoretically, on engine average and with an isentropic process, the pumping losses are reduced to zero because all the work lost in compression is recovered in expansion. With zero breather area the level of flow exchange between one CylBay and the neighbors is zero; as a consequence, flow losses are very low and all the work done on crankcase compression is recovered upon expansion (piston moving toward TDC). For this reason, the Engine Average (brown line) is close to zero in all the speed range. The CPMEP is not zero in any case; this is due to the blow-by flow between the cylinder and the crankcase, which is small and negligible if compared to the bay to bay or windage tray flow but it has been considered in the model. Moreover, the higher pressure amplitude of variation generates more internal friction and high heat transfer to the wall, so not all the compression work is recovered upon expansion (the real process is not isentropic).

From this model, which reflects the analytical results predicted in paragraph 3.3, it can be stated that, **theoretically, with an isentropic process, zero breathing area will minimize the pumping losses and thus maximize the engine performance.**

Unfortunately, this cannot be done for at least 2 reasons. The first one is that a zero ventilating area will generate a too high pressure variation inside the crankcase; this can be understood considering Figure 5.4-7 , where the pressure trace as function of crankangle is plotted for both zero ventilating area and for the real ventilating area (the one really implemented in the engine considered, same of Figure 5.2-8). The pressure amplitude is much greater in the first case, reaching 45000 Pa, which is a value which cannot be tolerated by the engine block's gasket. This level of pressure will blow out the engine sealing.

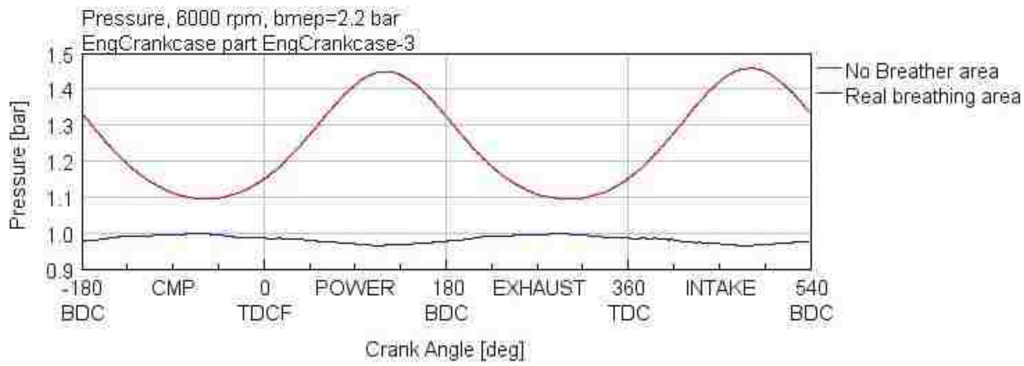

Figure 5.4-7: Pressure for real and closed breather area.

The second reason is related to the blow-by flow; this flow needs to be evacuated from the crankcase, to prevent the pressure to build up, and this cannot be done without a ventilating area.

For these reasons, the ventilating area is necessary and should be correctly designed to guarantee high crankcase scavenge efficiency and low pumping losses.

To release the pressure amplitude in the crankcase, the bay to bay holes and the windage tray holes are created; also the space over the oil in the oil pan contributes to gas flow exchange. In any case, these areas generate power losses.

In Figure 5.4-8 the CPMEP is plotted as function of different breathing areas and one single engine regime (6000 rpm), for the 6 cylinders and for their average. As can be noticed from the second plot, for a given engine speed there is a given area for which the CPMEP is a maximum. Considering 6000 rpm, for the considered engine, the diameter which generates the greater pumping losses is close to 20 mm; this critical diameter is responsible for a critical breathing area in which the losses are maximized. This is also confirmed by the literature [29], [32].

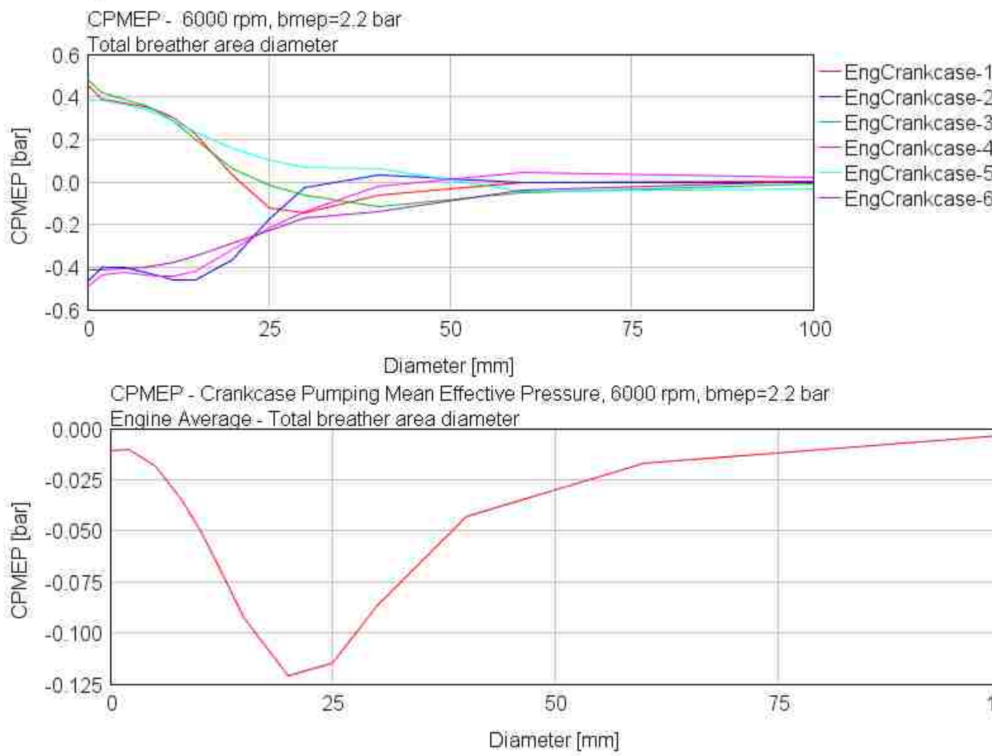

Figure 5.4-8: CPMEP as function of total breather area diameter.

On the left side of the critical breathing area, for diameter smaller than the critical one, the CPMEP decreases if the total breathing area decreases; this happens because the available section for gas flow is very small and if it further decreases the gas flow exchange is reduced and so the related losses. For zero breather area the CPMEP is close to zero, with small pumping losses due to heat transfer from the fluid to the hardware of the crankcase, which is great due to the high pressure variation (no isentropic process).

On the right side of the critical area, for diameters larger than the critical one, an increment of the breathing area leads to a reduction of the CPMEP, because the resistance to inter-bay flow and bay to oil pan flow is greatly reduced.

The critical area is not constant, but changes with the engine speed. Increasing the engine speed, the critical area increases and the maximum negative peak of CPMEP is greater. This can be seen in Figure 5.4-9.

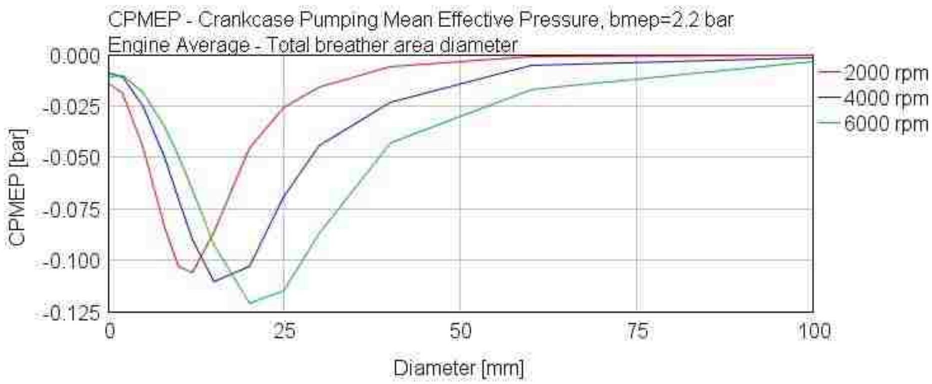

Figure 5.4-9: CPMEP as function of breather area for different engine speeds.

It can be noticed that if the breather area is very great (over 80-90 mm) the CPMEP is even lower than the case with closed breather area, due to lower heat transfer losses. In a normal wet sump engine very small pressure variations can be tolerated and the crankcase pressure should be as constant as possible; a very large ventilating area, over the critical ventilating area is so mandatory. The size of bay to bay holes and windage tray holes should be sized accordingly.

The critical area is very important in case a dry sump system is adopted: in this case there is no breather area between the bays and the only area opened for pressure release from the crankcase is the scavenge section on the sump which communicates with the suction pump (see Figure 2.7-4). The diameter of this section is likely to be close to the critical diameter; this critical section should be carefully avoided in order to reduce the CPMEP.

5.4.1 Effect of changing the bay to bay holes diameter

The crankcase pressure can be controlled with the size of the bay to bay holes; in the considered engine, each bay can communicate with the neighboring bays with two holes. Figure 5.4-10 shows the mass flow rate through one communication hole between CylBay12 and CylBay34. The mass flow rate is function of the crankangle, with a general sinusoidal trace. Positive values mean flow from CylBay34 toward CylBay12; negative values mean the reverse flow. The maximum mass flow rate is relatively great, reaching a peak of over 85 kg/h. This flow releases the pressure in regions where the pressure is high and increments the pressure where it is low. The pressure amplitude can be reduced increasing the size of these holes. A greater hole allows a greater gas mass to flow in the adjacent bay, reducing the pressure oscillations. This can be seen in Figure 5.4-11.

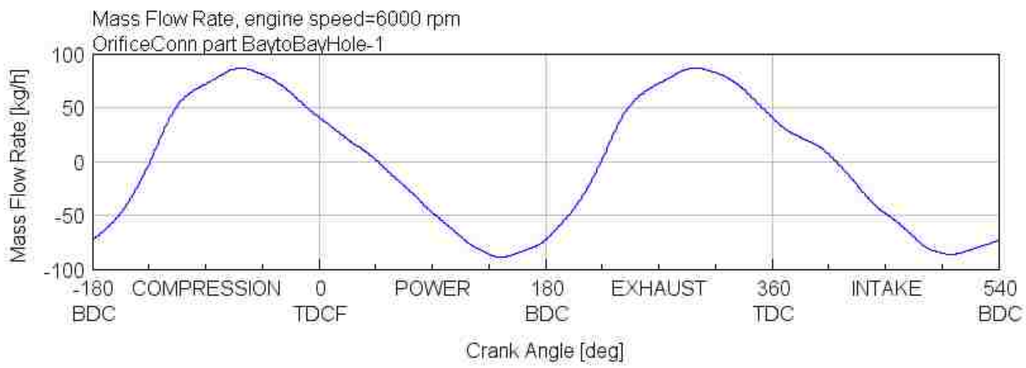

Figure 5.4-10: Mass Flow Rate through the bay to bay hole.

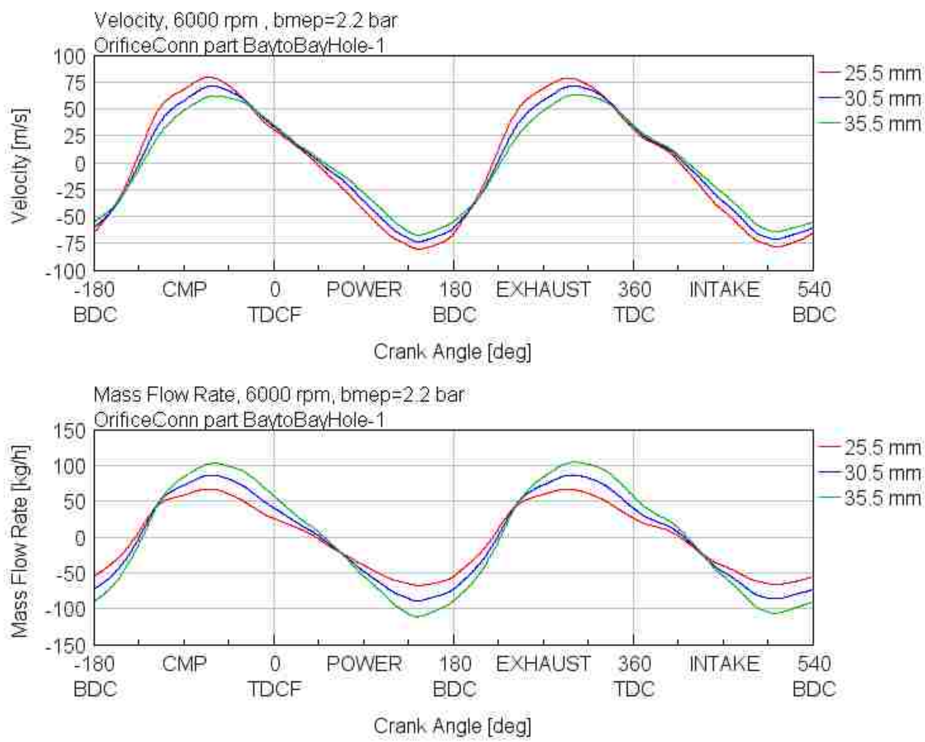

Figure 5.4-11: Mass flow rate and Velocity through the bay to bay hole.

The reference breather hole diameter was set to 30.5 mm (blue line). The red line refers to a smaller hole (diameter 25.5 mm), the green to a bigger one (diameter 35.5 mm).

The greater hole allows greater mass to flow, as expected; the opposite happens for the smaller hole. The velocities follow the opposite trend; maximum speed is reached with the smallest hole and it decreases increasing the hole diameter. As shown in Equation

3.29 the velocity through the breathing hole is linearly related to the piston speed. This can be clearly seen in Figure 5.4-12. If the hole is smaller the speed increases and thus the related frictional losses.

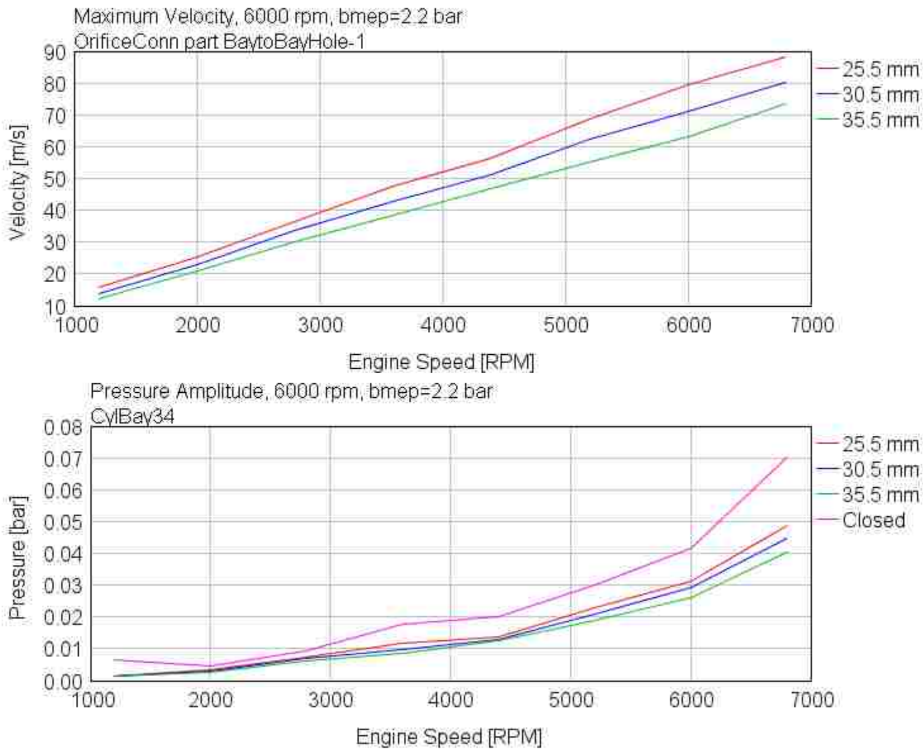

Figure 5.4-12: Maximum Velocity through the bay to bay hole and Pressure Amplitude in the cylinder bay.

Considering the second graph of Figure 5.4-12, it is clear that pressure amplitude is minimum with a large size of the holes; the case in which the holes are closed (no bay to bay communication) has the highest pressure oscillation (magenta line), highlighting the benefit of the breathing holes.

The effect of the breathing area on the crankcase pumping mean effective pressure is shown in Figure 5.4-13. The CPMEP decreases if the breathing area increases. Completely closed holes correspond to maximum CPMEP, because the pressure cannot be released and the piston has to face a greater resistance. In the considered engine the total breather area is much greater than the critical one, so considering Figure 5.4-8, it is on the right side of the critical breather area. On the right side a reduction of the breather diameter (and so of the breather area) leads to an increment of the CPMEP.

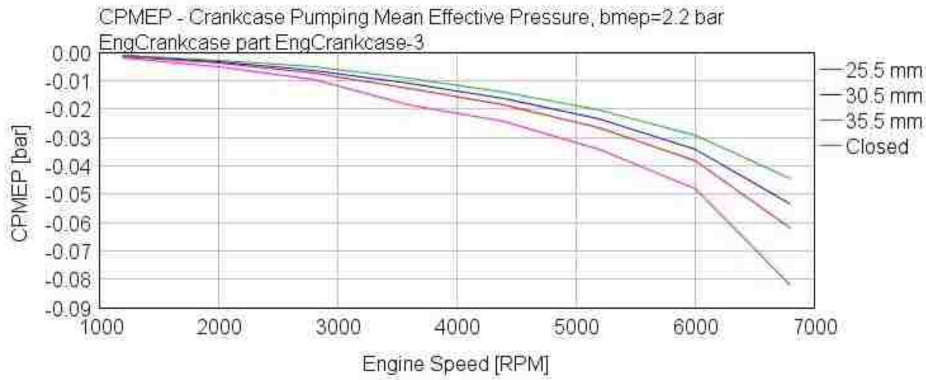

Figure 5.4-13: CPMEP for different breathing hole diameter.

If the bay to bay holes are closed, a greater mass flow rate is forced to pass through the windage tray holes, increasing flow losses. For this reason, closing completely the bay to bay holes does not improve the work at the shaft if the total breather area is not enlarged accordingly.

In the first graph of Figure 5.4-14 the pressure variation in the bottom part of the cylinder liner for two different cases is plotted. The red line corresponds to the case in which the bay to bay holes diameter is 30.5 mm; the green line is the configuration with closed bay to bay holes. As expected, with closed holes, pressure amplitude of variation is greater, resulting in greater risk of blowing out the engine sealing. An increase in pressure amplitude means greater flow through the windage tray and consequently greater losses. This is shown in the second graph, where the mass flow rate through the windage tray has been plotted as function of crankangle. A greater mass flow rate goes toward the oil pan in the green case, so greater power losses are generated. Moreover, greater gas flow toward the oil pan further increases oil motion and splashing, aerating it and reducing its service life; the risk of greater oil quantity trapped in the windage cloud is also increased.

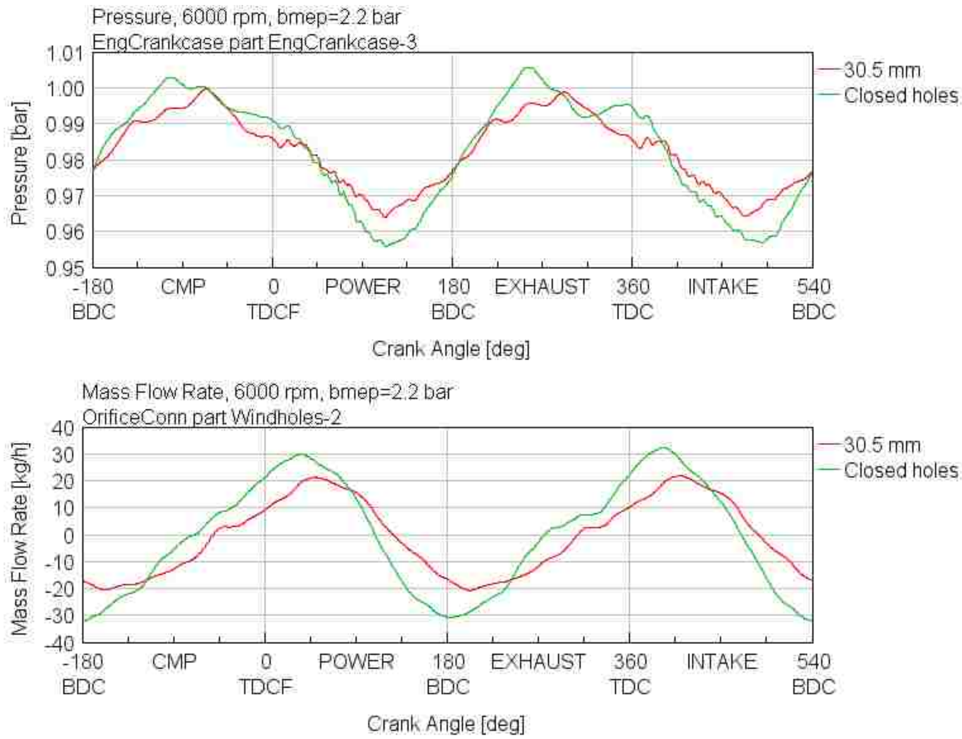

Figure 5.4-14: EngCrankcase3 Pressure for open and closed bay to bay holes area and related Mass Flow Rate through the windage tray holes.

5.4.2 Effect of windage tray holes diameter

The effect of changing the diameter of the holes in the windage tray is similar to the effect of changing the bay to bay holes; the increment of the diameter allows a greater quantity of gas to flow in the oil pan, releasing the pressure. Smaller holes in the windage tray make a greater fluid mass flow rate passing through the bay to bay holes and they should be enlarged to reduce pumping losses; smaller windage tray holes also reduce the oil aeration, stretching its service life.

The velocity of the flow through the windage tray holes is showed in Figure 5.4-15.

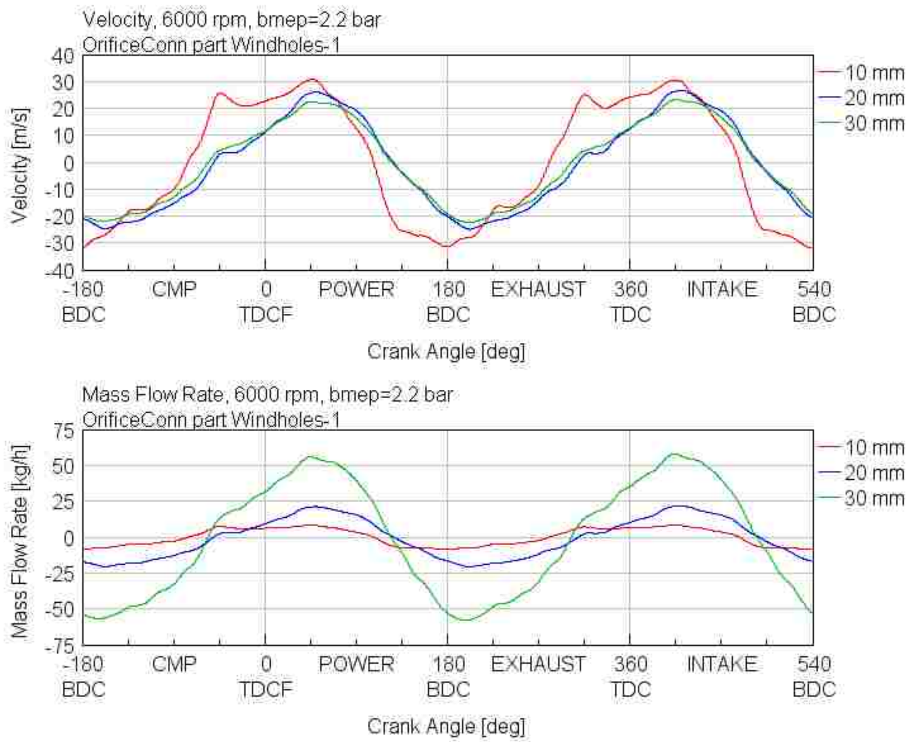

Figure 5.4-15: Velocity and Mass Flow Rate through the windage tray hole.

As expected, the maximum velocity is greater for smaller holes and decreases increasing the diameter. The mass flow rate follows the opposite trend; smaller holes allow a smaller gas mass to flow and it is more difficult to release the high peaks of pressure in the crankcase. This reflects in a greater effort on the back piston surface and so greater pumping losses.

From the result of the model, the effect of the windage holes on the CPMEP is lower compared to the bay to bay holes, because they are farther from the high peak pressure regions close to the back surface of the piston. In any case, these are results provided by a simplified 1D model. This model does not consider the rotation of crankshaft and counterweights and so their influence is not taken into account, diminishing the effect of the windage tray holes.

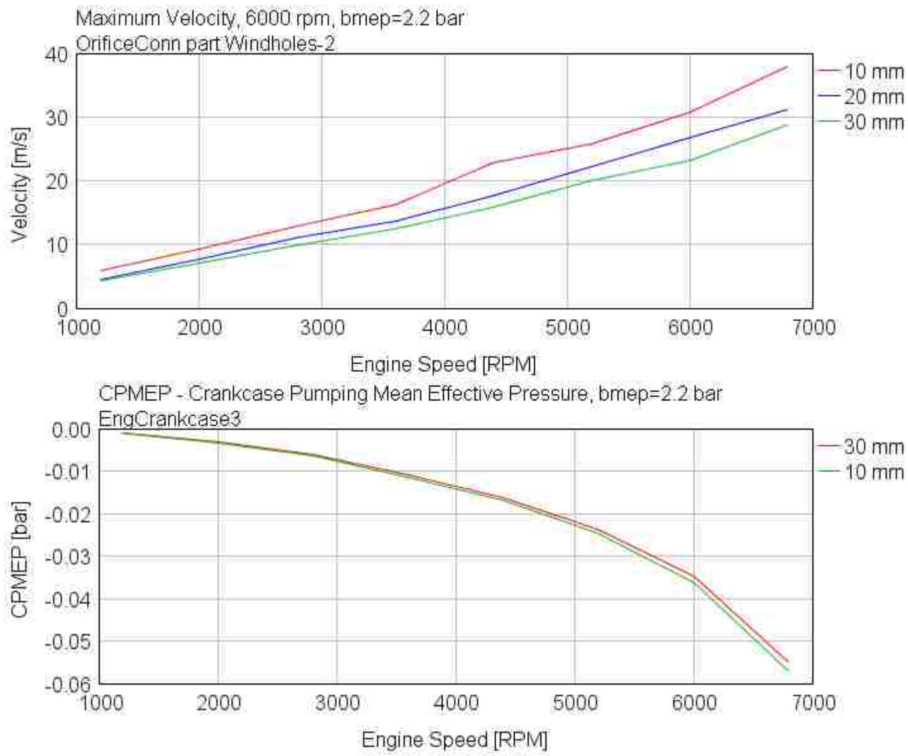

Figure 5.4-16: Maximum velocity through the windage tray holes and related effect on CPMEP of EngCrankcase3.

There is a limit on the size of the windage tray holes. Too great holes facilitate the risk of whipping up the oil from the pan, with an increment of the pumping losses due to the greater fluid density. If a greater flow reaches the sump, the oil splashes, is aerated and can be damaged. The diameter of the holes is usually a tradeoff between the requirement of pressure release and low oil motion in the pan.

5.5 Effect of fluid density

The first run of the model was initialized considering just burned gases in the crankcase; imposing a given air fuel ratio, the software computes the combustion products which are present in the crankcase. When the model is running, the software computes the burned gas composition, also considering that fresh air is entering the system. Burned gases can enter the crankcase through the blow-by orifices, while fresh air enters from the make-up-air circuit. A mixture of burned gases and fresh air reaches the intake manifold from the

PCV circuit; in some operating conditions, when there is high blow-by production, part of these gases can exit the crankcase through the make-up-air circuit.

Burned gases and air are not the only fluids present in the crankcase; usually a not small percentage of volume is occupied by oil droplets and oil vapors; the crankshaft rotation whip up the oil from the oil pan, and generally oil is wildly flying in the crankcase due to crankshaft, counterweights and connecting rods motion. The oil coming from the bearings of the crankshaft is also trapped in the crankshaft rotation and the common lubrication of the bottom part of the piston through oil squirters does not improve the situation. The result is a greater fluid density compared to burned gases only and greater density means greater power losses.

In order to evaluate this effect, the initial crankcase fluid was modified considering other two fluid compositions for Case 1; the first one considers 70% mass fraction of burned gases and 30% of oil vapor, the second 50% of burned gases and 50% of oil vapor. In Figure 5.5-1 there is a comparison between the three considered fluids.

Initialization	Fluid 1 (red)	Fluid 2 (blue)	Fluid 3 (green)
Exhaust gases (mass fraction)	1	0.70	0.50
Oil vapor (mass fraction)	-	0.30	0.50

Table 5.5-1: Fluid initialization.

During engine operation, unburned air is entering the system through the make-up air circuit, changing the mass fractions described above.

In Figure 5.5-1 there is a comparison between the densities of the three considered fluid and their effect on the CPMEP as well.

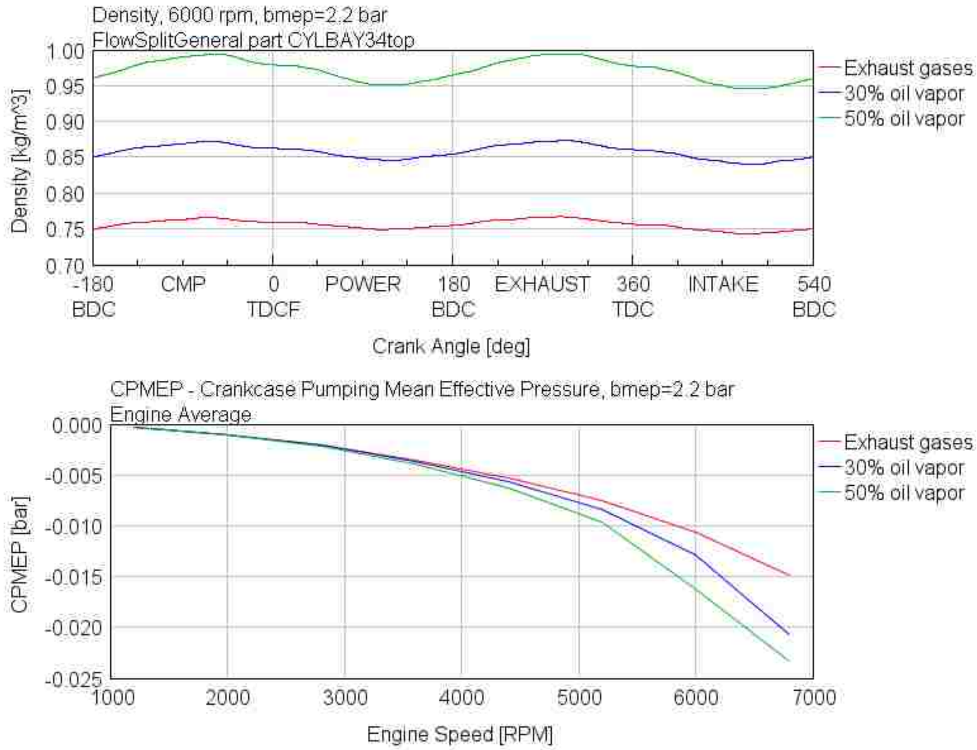

Figure 5.5-1: Density and CPMEP for different fluid compositions.

According to Equation 3.42 the crankcase pumping mean effective pressure is directly related to the fluid's density. This can explain the trend in the Figure 5.5-1. It can be noticed that the greater effect is present at high engine speed, where there is a high fluid speed through the breather area and so high mass flow rate, which generates power losses. The power lost, considering maximum engine speed and the fluid with the greater density is:

$$\dot{W}_{crankcase} = \text{CPMEP} \cdot \frac{V_{crankcase}}{V_u} \cdot V \cdot \frac{rpm}{60} =$$

$$= 2400\ Pa \cdot \frac{0.683\ l}{0.6\ l} \cdot 3.6\ l\ \cdot \frac{6800}{60} = 1.041\ kW = 1.396\ HP \qquad (5.13)$$

It is a greater value than the case considering just exhaust gases, because a great quantity of oil vapor is mixed with the exhaust gases in the crankcase.

The last case was run considering a mixture of exhaust gases and liquid oil 5W-30.

The volume fractions considered are depicted in Table 5.5-2.

Fluids	Volume fractions [37]
Exhaust gases	0.997
Oil 5W-30	0.003

Table 5.5-2: Fluids composition, Oil 5W-30 and Exhaust gases.

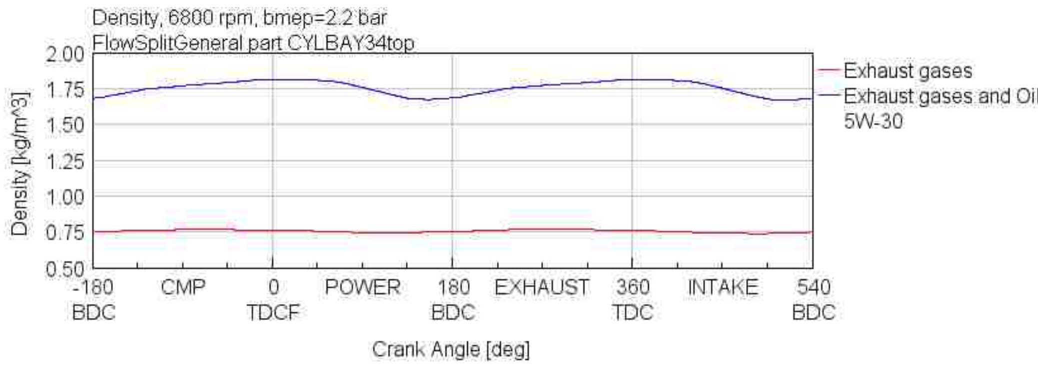

Figure 5.5-2: Density and CPMEP for Oil 5W-30 and Exhaust gases.

According to the authors of [37], this mixture can be considered the closest to the real fluid. The average density is very high because a liquid is considered in the mixture and this leads to high power losses. The CPMEP value for engine average, considering the case at 6800 rpm was 4539 Pa. The power lost is:

$$\dot{W}_{crankcase} = \text{CPMEP} \cdot \frac{V_{crankcase}}{V_u} \cdot V \cdot \frac{rpm}{60} =$$

$$= 4539\ Pa \cdot \frac{0.683\ l}{0.6\ l} \cdot 3.6\ l\ \cdot \frac{6800}{60} = 2.108\ kW = 2.826\ HP \tag{5.14}$$

It is a much greater value compared to the previous one, and also much more realistic. Smaller windage tray holes and deeper oil pan can decrease the amount of oil whipped up by the crankshaft rotation, reducing the power sapped by the oil.

5.6 Effect of oil pan volume

In this section the effect of changing the oil pan volume has been evaluated; in the considered model only the volume not occupied by the oil has been considered. From the 3D CAD of the oil pan, the volume occupied by the oil has been subtracted to the total pan volume. In any case, the oil level in the pan is not constant but it changes as function of engine speed; at high speed, more oil is sucked by the pump and the oil level is reducing, increasing the space available for gas movements.

At low speed the space for the gases is lower because lots of oil is in the pan. In the model, a linear variation of the oil pan volume as function of the engine speed has been considered.

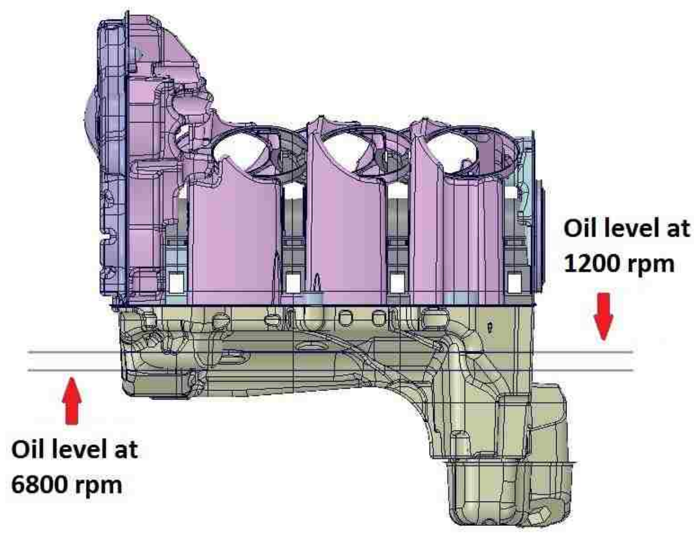

Figure 5.6-1: Oil level.

At high speed the crankshaft rotates faster and so greater turbulence is generated which increases oil splashing; however, the crankshaft rotation is not taken into account by the simplified 1D model. The model does not consider the interaction between gas and lubricant in the oil pan and so it is not able to provide sensible differences in the final output. This can be seen from Figure 5.6-2 where the CPMEP is similar for two different oil pans, one with the volume double than the original and the other half of the original.

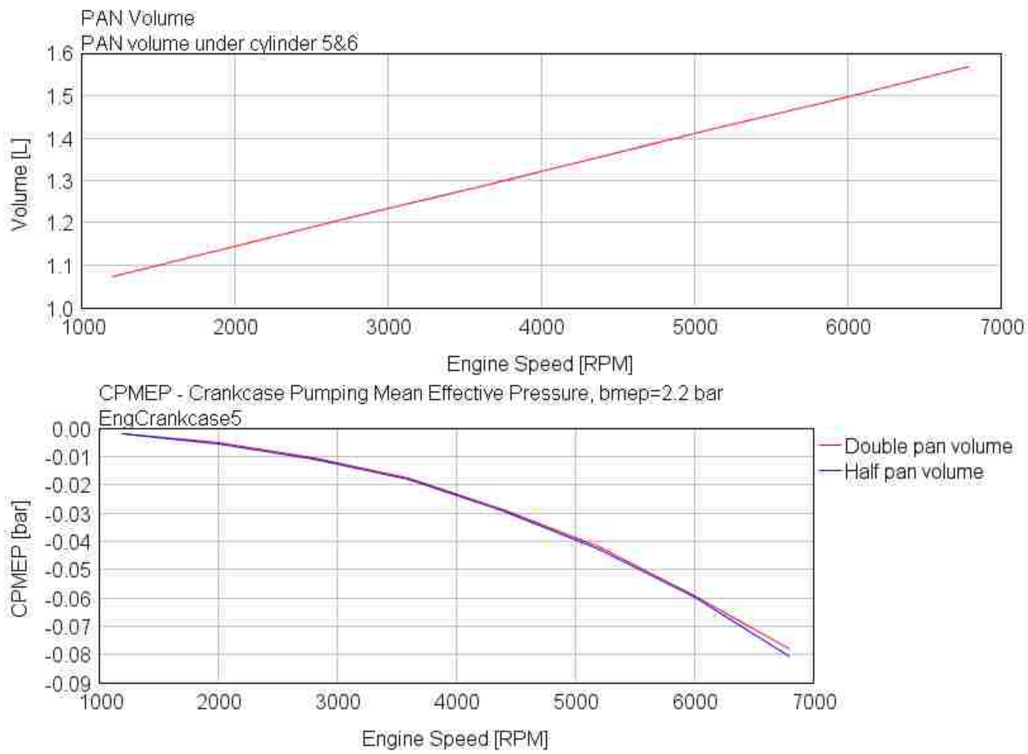

Figure 5.6-2: PAN Volume variation and CPMEP for cylinder 5.

Real benefits can be achieved with a deeper oil pan. A deeper oil pan reduces the risk of whipping up the oil in the crankshaft rotation, providing additional benefits especially in engines without a windage tray. The opposite happens for a reduced oil pan volume. Unfortunately, overall engine height and CG position requirements limit the size of the oil pan in order to get a sufficient ground clearance.

5.7 Dry sump system

As said before, for an isentropic process, the lowest pumping losses can be obtained only with completely closed breather area; in this way there is no gas flow and no power losses. The weak points are the requirements of low pressure amplitude of variation (to prevent gasket damages) and blow-by gases scavenge. A dry sump system jumps over these weak points; in this system there are no bay to bay holes, minimizing the flow related losses. A vacuum pump sucks the blow-by gases, scavenging the crankcase and providing low pressure inside the crankcase; this also decreases the crankcase fluid density which offers less resistance to the crankshaft motion.

In order to evaluate the effect of the dry sump system, the original model was modified closing the breather area in the CylBays. The pan volumes were connected to three EndEnvironment objects, which simulate the low pressure regions generated by the vacuum pump, one for each of the CylBays. The pressure in this environment was set to a general value of 0.35 bar, typical of motorsport applications [3], [32]. The PCV circuit and the make-up-air circuit were disconnected from their connections with the intake manifold. Gas can now enter the system only through the blow-by orifices and, for mass conservation; the same mass flow rate must exit from the dry sump system (sucked by the vacuum pump). In Figure 5.7-1 there is a comparison between the wet sump system considering the exhaust gases and oil mixture of Table 5.5-2 and the dry sump model.

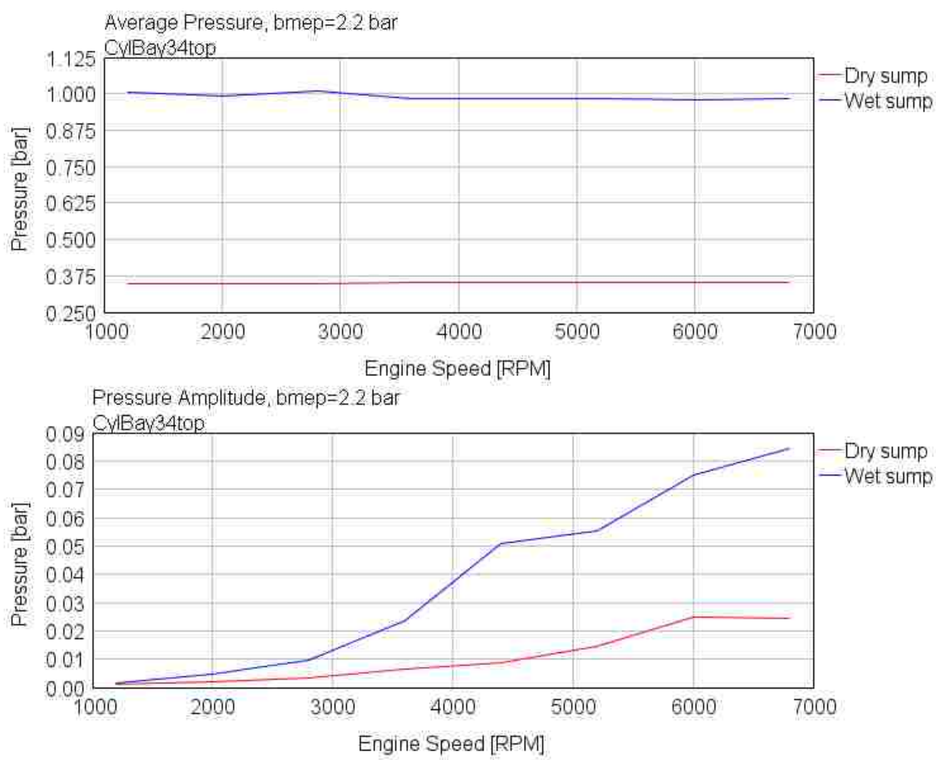

Figure 5.7-1: Average Pressure and Pressure Amplitude in CylBay34.

The average pressure is low everywhere in the dry sump circuit and close to the vacuum provided by the scavenge pump. The pressure amplitude of variation is also greatly reduced.

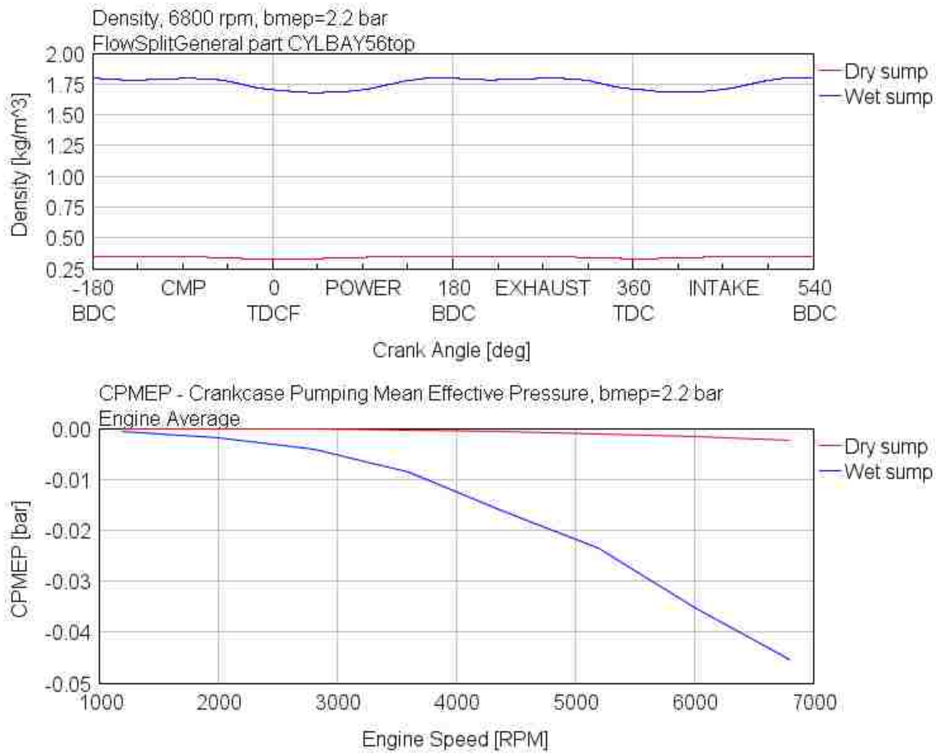

Figure 5.7-2: Density and CPMEP for wet and dry sump.

Effective advantages can be obtained thanks to the lower pumping losses. As can be seen from Figure 5.7-2, the fluid density is greatly reduced compared to the wet sump solution, due to the lower fluid pressure. As a consequence, the CPMEP was decreased from 4539 Pa to 229 Pa at 6800 rpm adopting a dry sump solution.

Generally, the power required to drive a vacuum pump in a dry sump system is around 3-5 HP [38]; in any case the internal oil pump that robs horsepower as well is removed so the loss is minimal from the accessories point of view.

From the results of the model, the power gain adopting the dry sump system is 2.68 HP at 6800 rpm, which is a big improvement at that engine speed. At 2800 rpm, which is an engine speed more likely for road engines, the power gain is only 0.25 HP because the CPMEP as a negative parabolic trend.

The adoption of the dry sump system for efficiency gains is hardly justified for a road engine, because it usually operates at very low speed were the power gains are minimum.

A dry sump system is much more suitable to be applied to high speed engines like racecar engines and motorcycle engines than common road car engines. The CPMEP is proportional to the squared value of engine speed; this means that the engines which work at high engine speed have the greatest crankcase pumping losses. Moreover, these engines operate for the majority of their life at high speed, unlike road car engines which most likely operate at low speed. Race cars and motorcycles are also subjected to great longitudinal and lateral forces which can create high oil motion in the wet sump and create lubrication problems; this situation is less likely in road cars. A dry sump system deletes the problems related to oil motion in the pan because the oil is stored in a separated reservoir. The dry sump system improves the weight distribution in the car, because the engine height is lower and the engine can be fitted lower in the engine compartment; the oil reservoir can be positioned in a place close to the center of gravity of the car.

Furthermore, a dry sump system is much more expensive than a traditional wet sump.

For all of these reasons, the dry sump system is difficult to be found on a road car engine.

6 CONCLUSIONS & RECOMMENDATIONS

6.1 Conclusions

The basic conclusions which can be stated from this work are:

1. The crankcase ventilation system is an important part of the engine and should be further studied and optimized to improve the efficiency of the engine.
2. The CPMEP for the considered engine is very small and in the order of few thousands of Pa; even if it is low, it is always an indicator of work subtracted from the crankshaft output, hence every improvement is directly reflected in efficiency improvement. The low level of CPMEP can be explained with the fact that in the 1D model the flow field is activated only by the motions of pistons and by the flow through blow-by orifices and PCV/make-up-air circuit; in reality the crankshaft and counterweights rotation has a great impact on the flow inside the crankcase and on the power losses. The 1D/CFD model is not able to consider the drag on the crankshaft, thus the overall losses are underestimated.
3. Assuming an isentropic evolution in the crankcase, minimum CPMEP is reached with completely closed breather area; this results in minimum flow losses in the crankcase due to the null flow exchange between the bays. Unfortunately, the requirements of low pressure variation inside the crankcase and blow-by gases scavenge do not allow this solution to be adopted.
4. In order to minimize pressure variations and clean the crankcase from blow-by gases, the breather area is necessary if a wet sump solution is adopted. There is a critical non zero area for which the pumping losses are maximized, hence this area should be carefully avoided.
5. The CPMEP is mainly function of engine speed, breather area, fluid density and crankcase volume. The trace of CPMEP vs engine speed has a negative parabolic trend.
6. Maximizing the breather area will reduce the crankcase pumping losses. The best approach is to increase as much as possible the size of the bay to bay holes, which can be done as long as the requirement of engine block stiffness is satisfied.

7. Smoother hole entrance and exit will also improve the coefficient of discharge for the considered bay to bay hole section, reducing the power losses.
8. Increasing the size of the holes in the windage tray will increase the size of the breather area. If these holes are too big, then the risk of oil splashing in the crankcase is greater and this means greater pumping losses due to greater fluid density. The size of the breather area should be set to the maximum without enlarging too much the windage tray holes. A completely closed windage tray minimizes the power sapped by the oil motion.
9. Minimum fluid density in the crankcase minimizes the crankcase losses because lower resistance is applied on flow exchange between the bays and crankshaft rotation.
10. The 1D model does not show effective improvements increasing the size of the oil pan. The reason for this is that only the volume over the oil level in the pan has been considered. For practical applications, a deeper oil pan greatly reduces the risk of having oil wildly flying in the crankcase, thus reducing the friction at the shaft. The oil pan should be as low as possible as long as the requirement of minimum ground clearance is satisfied.
11. The dry sump system provides a great reduction of the pumping losses inside the crankcase, because the density of the fluid is greatly reduced. The dry sump system should have closed breather area to reduce the losses. In any case, the higher cost and complexity of a dry sump system does not offset the small improvements gained with its use in a road car engine because it usually operates at low engine speed where the pumping losses are low.

6.2 Recommendations

The present model is based on the geometry of Chrysler Pentastar V6 3.6L engine and provides useful indications on the behaviour of the fluid in the crankcase; it can also be used in the design phase of future V6 engines, both for pumping losses evaluation and for ventilation system design. At the time of the writing, it is the first 1D/CFD model of a crankcase ventilation system ever built in both Fiat and Chrysler.

As previously said, it is a mono-dimensional model: as a consequence it does not consider the rotation of crankshaft and counterweights, which both have a big influence on the flow field in the crankcase. This can be considered the biggest limitation of this kind of analysis, because the CPMEP is likely to be underestimated.

However, the 1D/CFD model is able to provide reliable results, close enough to the 3D CFD but with much shorter computational time, hence time and money can be saved.

The present model was correlated with only a limited amount of experimental data; the only data available were the blow-by production as function of engine speed and manifold absolute pressure and the mass flow rate through the PCV valve as function of the vacuum level in the intake manifold.

More data is necessary in order to improve the accuracy of the model, such as crankcase pressure trace as function of crankangle for each of the crankcase bays. As previously explained, the pressure trace in the crankcase dictates the P-V diagrams, and so the work losses. If a detailed estimation of the CPMEP is required, the pressure behaviour as function of crankangle is necessary for different engine speeds. This information can be obtained putting pressure sensors in the crankcase. Pressure values of other parts of the system as oil pan, front cover, cylinder head and cam cover can be very useful for tweaking the model, as well as fluid temperature measurements.

A very useful piece of information could be the velocity of the flow through the bay to bay holes and windage tray holes, because they are directly related to power losses. If possible, a measurement of the density of the flow in the crankcase should be done too.

With all of these measurements the model can be greatly improved and a higher accuracy can be achieved.

Unfortunately, for this project, it was not possible to instrument an engine and record the above mentioned information.

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