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ADVANCED METHODS OF LOAD REDUCTION FOR INTERNAL COMBUSTION ENGINES

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Table of contents

ABSTRACT……..5

CHAPTER 1 INTRODUCTION…………………………………………………………………………………………………6

1.1 The problem of the efficiency…………………………………………………………………………………..……6

1.2 Causes of the efficiency decrease………………………………………………………………………………….7

1.3 Alternative methods for load reduction………………………………………………………………………….9

1.4 State of art of the Variable Valve Acting technology…………………………………………………………………10

1.5 List of contents……..12

1.6 Conclusion and motivation of study……………………………………………………………………………………13

CHAPTER 2 THEORY OF MILLER CYCLE………………………………………………………………………………………14

2.0 Overview………14

2.1 S.I. engine's working principle………………………………………………………………………………………….14

2.2 the Miller cycle………..17

2.2.1 Introduction……..17

2.2.2 Thermodynamic Transformations…………………………………………………………………………………….18

2.2.3 Thermodynamic efficiency of Miller cycle…………………………………………………………………………19

2.2.4 Implementation of Miller cycle………………………………………………………………………………………20

2.3 main applications and advantages of the Miller cycle………………………………………………………………21

2.4 comparison between Miller and Atkinson cycles…………………………………………………………………….22

2.5 Summary……23

CHAPTER 3 COMPARISON between Gasdyn and Gt Power………………………………………………………24

3.0 Overview……24

3.1 Comparison between the mean performances…………………………………………………………………………26

3.1.1 Brake torque………26

3.1.2 Brake power…….27

3.1.3 Volumetric efficiency……………………………………………………………………………………………………….28

3.1.4 Brake specific fuel consumption…………………………………………………………………………………….29

3.1.5 Bmep……….30

3.2 comparison of the instantaneous performances……………………………………………………………………….31
3.2.1 Air temperature………31
3.2.2 Air pressure……32
3.3 Conclusion……..32
CHAPTER 4 Load control on a 2000 c.c. 16 valves naturally aspirated engine obtained by the Miller cycle…33
4.0 Overview……33
4.1 The full load case………..35
4.2 Load reduced to 66% of the full one……………………………………………………………………………………..37
4.2.1 Load reduction obtained by the throttling valve…………………………………………………………………….37
4.2.1 Load reduction by means of EIVC strategy………………………………………………………………………..38
4.2. 2 Load reduction by means of LIVC strategy………………………………………………………………………….39
4.2.4 Comparison between the control strategies…………………………………………………………………………40
4.3 Load reduced to 27% of the full one……………………………………………………………………………………..43
4.3.1 Load reduction by means of the throttling valve…………………………………………………………………..44
4.3.2 Load reduction by means of the EIVC strategy……………………………………………………………………..45
4.3.3 Load reduction by means of the LIVC strategy………………………………………………………………………45
4.3.4 Comparison between the control strategies…………………………………………………………………………..46
4.4 Load reduced to 11% of the full one……………………………………………………………………………………..48
4.4.1 Load reduction by means of the throttling valve…………………………………………………………………..48
4.4.2 Load reduction by means of the EIVC strategy……………………………………………………………………..49
4.4.3 Load reduction by means of the LIVC strategy………………………………………………………………………50
4.4.4 Comparison between the control strategies…………………………………………………………………………..51
4.5 Global comparison of all the control strategies for all the load configurations…………………………………..53
4.6 Open cycle efficiency analysis………………………………………………………………………………………………57
4.7 Closed cycle efficiency analysis……………………………………………………………………………………………..58
4.8 Summary……59
CHAPTER 5 Load control in a four-cylinders turbocharged engine obtained by the Miller cycle…………………60
5.0 Overview……60
5.1 Full load configuration………………………………………………………………………………………………………..61

5.2 Engine simulated at 70% of full load……………………………………………………………………………………………..64
5.2.1 Load reduction by means of the throttling valve…………………………………………………………………..64
5.2.2 Load reduction by means of the EIVC strategy…………………………………………………………………….65
5.2.3 Load reduction by means of the LIVC strategy………………………………………………………………………66
5.2.4 Load reduction by means of the lowering of the boosting pressure……………………………………….67
5.2.5 Graphical comparison between the control strategies…………………………………………………………68
5.3 Engine simulated at 48.7% of full load……………………………………………………………………………………….70
5.3.1 Load reduction by means of the throttling valve…………………………………………………………………..70
5.3.2 Load reduction by means of the EIVC strategy…………………………………………………………………….71
5.3.3 Load reduction by means of the LIVC strategy…………………………………………………………………….72
5.3.4 Load reduction by means of the lowering of the boosting pressure……………………………………….73
5.3.5 Graphical comparison between the control strategies………………………………………………………….74
5.4 Engine simulated at 37.8% of full load………………………………………………………………………………………77
5.4.1 Load reduction by means of the throttling valve…………………………………………………………………..77
5.4.2 Load reduction by means of the EIVC strategy……………………………………………………………………78
5.4.3 Load reduction by means of the LIVC strategy……………………………………………………………………..80
5.4.4 Load reduction by means of the lowering of the boosting pressure……………………………………….81
5.4.5 Graphical comparison between the control strategies………………………………………………………….82
5.5 Global comparison of all the control strategies for all the load configurations…………………………….85
5.6 Open cycle efficiency analysis……………………………………………………………………………………………………88
5.7 Closed cycle efficiency analysis………………………………………………………………………………………………….89
5.8 Summary…….90
CHAPTER 6 CONCLUSION……….91
APPENDIX……….92
ACKNOWELEDGEMENTS……93
BIBLIOGRAPHY………94

Abstract

The aim of this thesis is to show how the variable valve acting can improve the global fuel consumption of a spark ignition engine at partial load: in a classic spark ignition engine the load is varied by means of a throttling valve which causes a pressure loss in the inlet duct and consequently a pumping work which must be spent by the piston to make the fresh charge enter in the cylinder and so reduces the efficiency.

The variable valve acting (VVA) allows to regulate the air mass entering in the cylinder by means of an early intake valve closing (EIVC) or a late one (LIVC): the first one allows to entrap into the cylinder the exact quantity of air needed to run the engine at a precise load closing the intake port before the piston reaches the bottom dead centre, the second one makes the air excess be pushed back to the intake manifold at the beginning of the compression stroke; both the solutions avoid the pressure losses generated by the throttle plate.

Both the control strategies allow to implement the Miller cycle, whose main feature is the independence between the compression and the expansion ratio which allows to have a high expansion of the burning gases with a not too high compression of the fresh charge, thanks to their capability to vary the effective compression ratio, define by the ratio between the volume swept by the cylinder at intake valve closure and the clearance volume.

The work has been performed by means of Gasdyn, a 1D simulation software developed by Politecnico di Milano which calculates all the main performances of a given engine after the setting of all its parameters; the initial phase consists in the comparison of the results given by Gasdyn and the ones provided by Gt-Power (a similar widely used software) after the analysis of the engine of the motorbike Honda CBR 250, to verify if Gasdyn is reliable: the main performance curves (torque, power, specific fuel consumption, brake mean effective pressure, efficiency) resulted to be identical.

The study of the improvements obtained by the implementation of the variable valve action is performed on two engines: a four-cylinders 2000 c.c. naturally aspirated one, called Alfa Romeo Twin Spark, and a four-cylinders 1400 c.c. turbocharged engine, which is only a numerical model based on the Fiat Multi Air.

Both the engines have been simulated at the maximum volumetric efficiency speed, three load levels have been considered and each of them has been reached by means of the throttling valve, the early intake valve closing or the late one, while on the turbocharged one the load has been reduced also by means of the lowering of the boosting pressure obtained by the opening of the wastegate valve, which allows to gain a low or medium load reduction without throttling the engine or reducing the effective intake stroke's length; the implementation of each method has been explained for all the load configuration: the valve lift profiles needed for the realization of the EIVC and LIVC strategy have been derived, as for the opening angles of the throttle plate and the boosting pressures needed to reach each target load.

The main result obtained in the whole thesis is that the early intake valve closing is the partializing strategy which ensures the best efficiency both on the naturally aspirated engine and on the turbocharged one, given its capability to generate low pumping losses without wasting useful work; the use of this method is consequently necessary to reduce the specific fuel consumption when the engine doesn't work at full load, that is the condition at which the engine itself is always used during the urban driving.

Chapter 1

Introduction and motivation of the study

1.1 The problem of the efficiency

The efficiency of the internal combustion engines for the road transportations is the first issue the designer faces, since the fuel economy and the pollutant emissions are nowadays the most important characteristics that a car-buyer looks for because of money saving and respect of emission regulations.

The conversion of the fuel's chemical energy into the mechanical work at the wheel is intrinsically characterized by a low efficiency, because of the following reasons:

- **Thermodynamic**: the internal combustion engine is a thermodynamic machine, which cannot have a unitary efficiency, since the Second Principle of Thermodynamic states that it's not possible to convert into work all the heat exchanged with a single heat surgent
- **Friction**: the internal combustion engine is an alternative machine, which basically consists in a piston which moves in a cylinder having the lateral surfaces of the piston ring and the cylinder's liner in contact: the sliding friction generates non negligible losses, and can be lowered, but not deleted, by the oil film on the liner; other friction losses are also generated by the bearings a by all the gears present in the engine.
  The friction losses increase when the engine speed rises, given the quadratic dependence of the friction power by the relative speed of the involved bodies.
- **External devices**: The engine must produce power to ensure the working of some devices which are necessary for the functioning of the engine itself, like the distribution system, where some work must be spent to overcome the elastic forces provided by the springs which ensure the valve closure, or the alternator which ensures the battery charge: all these systems subtract some power to the crankshaft.

The thermodynamic and friction losses make the engine efficiency not be higher than 38% considering the best engines on the market.

The previous considerations have been made considering the engine at full load: the engine efficiency decreases with the load decreasing, because the energy spent to drive the auxiliaries becomes more important; this loss is strongly worsened if the device used to reduce the quantity of air-fuel mixture leads to a pressure drop in the working fluid, as it will be seen later.

This is actually the efficiency loss which is more important to reduce, since the highest part of the vehicle produced in the World circulate mostly in the urban roads, where the requested engine load is low, given the presence of traffic and speed limits: the improvement of the efficiency of the urban vehicle's engines must be concentrated on the lowest part of the engine map, at low load and low speed.

1.2 Causes of efficiency decrease

In an S.I. engine the load reduction is obtained by the lowering of the air-fuel mixture mass entering in the cylinder, which decreases the quantity of energy which can be provided by the engine itself, remembering that the effective power is given by the following formula:

$$P_e = \eta_g \frac{H_i}{\alpha} \lambda_v V \rho_a \frac{n}{\varepsilon}$$

Where H_i is the fuel's lower heating value, α is the air to fuel ratio, ρ_a is the air density, V is the engine's total displacement, n is the engine speed, ε is equal to 2 for the four-stroke engine and equal to 1 in case of two-stroke engines, η_g is the engine's global efficiency and $\lambda_v = \frac{m_{aspirated}}{V \rho_a}$ is the filling coefficient, which is reduced to decrease the engine load.

In the traditional S.I. engines, the reduction of the air mass entrapped by the cylinder is obtained by a throttling valve which reduces the effective cross section through which the air passes: this system causes a pressure loss across the valve, since the more the valve is closed, the more the C_d coefficient is decreased.

Given the presence of this concentrated loss and other ones (like the air filter) and distributed ones (on the ducts' surfaces), when the piston runs the intake stroke, he must spend some work to make the air overcome these losses and enter in the cylinder; this work is subtracted to the crankshaft, so the useful work produced by the engine is reduced: the more the throttling valve is closed, the more the pumping work is increased, and consequently the useful work is decreased and then the engine's efficiency is reduced.

The previous concepts can be better understood looking to the comparison between the Pressure-Volume diagram provided by the engine at full load with the ones resulting from the same engine throttled with different opening angles of the throttling valve: the work exchanged from the working fluid to the piston is the integral closed on the whole cycle of the pressure multiplied by the volume's differential:

$$L_i = \oint p dV$$

The integral's sign is given by the verse of the P-V curve: if is clockwise the work is positive, otherwise the verse is negative.

The pumping work is the anti-clockwise area in the lower part of the diagram, and its shape is due to the intake pressure which is lower than the atmospheric one because of the pressure losses: during the intake stroke, the volume of the combustion chamber increases while the air pressure is constant and lower than the atmospheric one.

That area increases at the increase of the throttling's pressure loss, so the useful work at the crankshaft is more and more reduced.

Recalling the formulas of indicated power:

$$P_i = \frac{L_i n}{\varepsilon}$$

and indicated efficiency:

$$\eta_i = \frac{P_i}{H_i \dot{m}_c}$$

Where H_i is the fuel's lower heating value and $\dot{m}_c$ is the fuel's mass flow rate, it can be inferred that the throttling's pressure loss reduces the useful work at the crankshaft, so it reduces the useful power and then the engine efficiency; the higher the throttling, the harder the loss.

The P-V diagram has been plotted in a logarithmic scale, which allows to clearly show the pumping zone.

The curves compared in the graph are taken at full load, at 30° of throttle valve opening and at 19° (where the load is the 15% of full load); it can be easily noticed how the pumping area increases at the increase of throttling.

The throttle valve angle equal to 30° is a small load reduction, so the P-V diagram of this case is close to the full load's one.

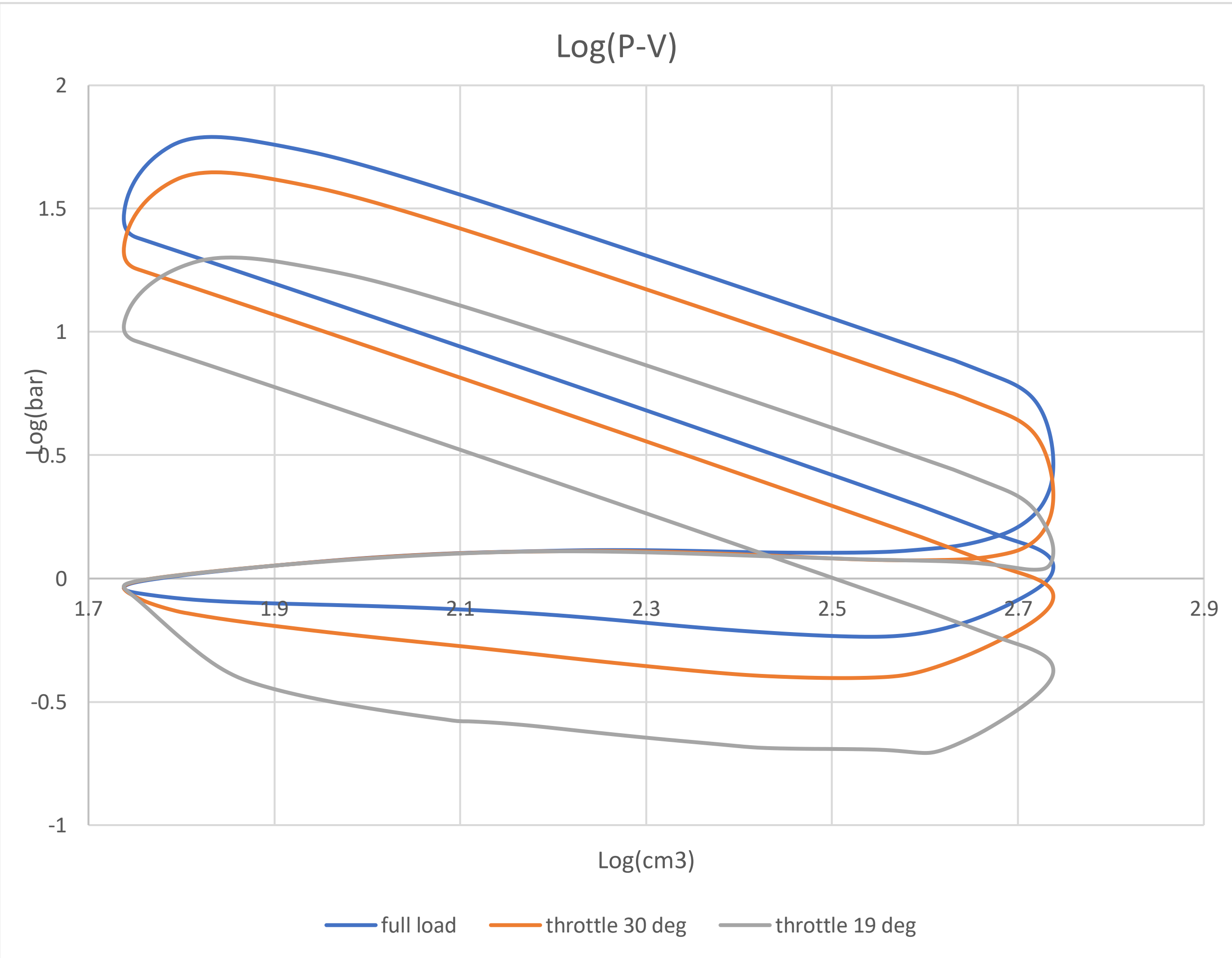

Figure 1.1 logarithmic P-V diagrams at different position of the throttle plate

1.3 Alternative methods for load reduction

The action really needed to reduce the load in an S.I. engine is the reduction of the air-fuel mixture's mass entering in the combustion chamber: the objective is to reach this reduction avoiding the pressure losses typical of the throttling valve.

This result can be achieved with a variable valve acting system, which allows to close the intake valve before the intake stroke is completed in order to entrap in the cylinder only the necessary mass of air, or to retard its closure with respect to the bottom dead centre, to eject back in the intake duct the excess of mixture; the first strategy is called Early Intake Valve Closure (EIVC), the second one is called Late Inlet Valve Closure (LIVC).

Those systems allow to eliminate the throttling valve (which is however mounted on the engine to ensure the load reduction in case of failure of the variable valve train system), whose absence avoids the pressure losses in the intake ducts; the pumping work is strongly reduced, since the air which enters in the combustion chamber has a pressure which is much closer to the atmospheric one than the previous case (considering a Naturally Aspirated engine).

The reduction of the pumping work increases the net useful one, increasing consequently the indicated efficiency of the engine, which is given by the ratio between the indicated work, that is the integral of the P-V diagram, and the engine displacement.

In the image below there is an example of EIVC and LIVC in comparison with the original valve lift of a naturally aspirated 2000 cc 16V engine, which will be analysed in a dedicated chapter in the work.

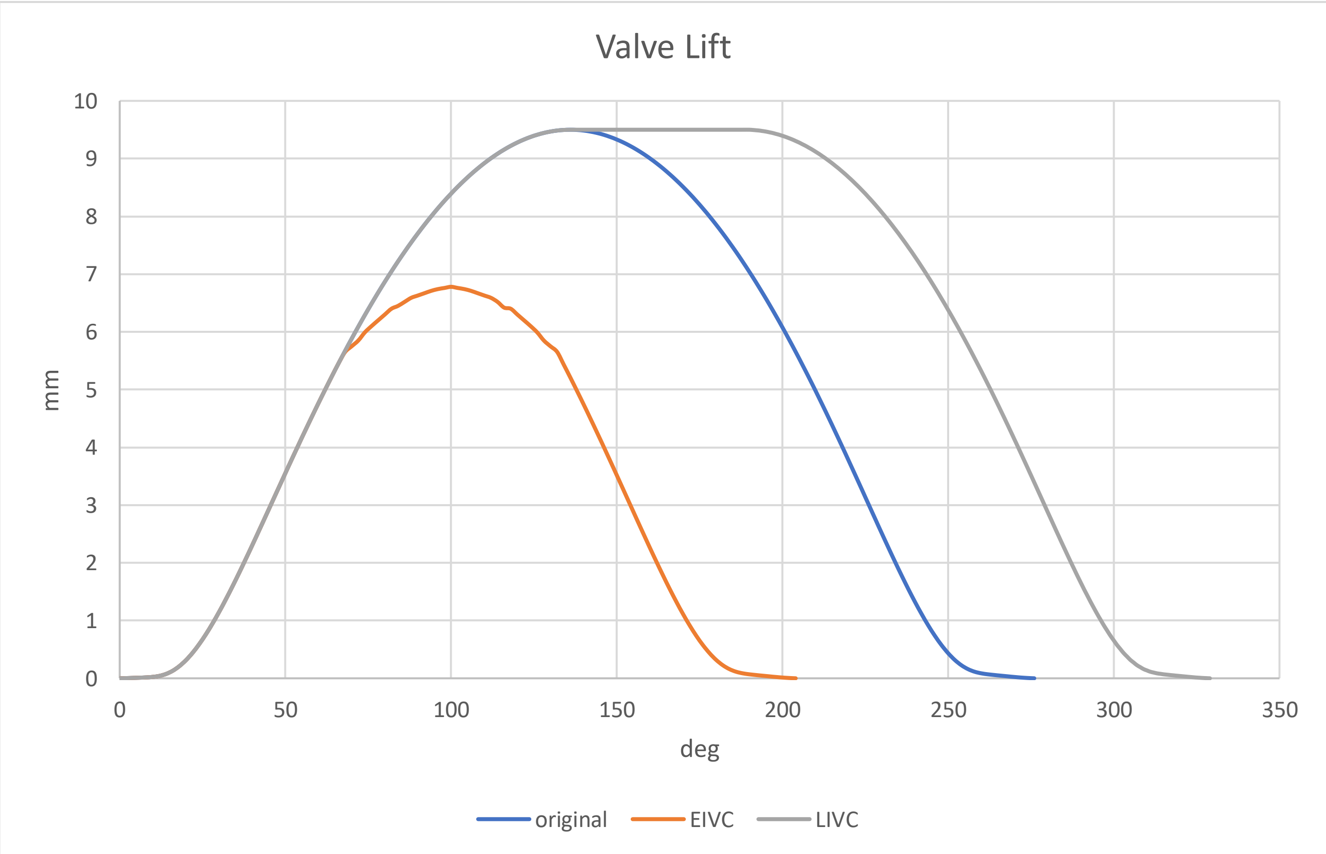

Figure 1.2 comparison between standard, EIVC and LIVC profiles

1.4 State of art of the Variable Valve Acting technology

The load control methods previously shown can be implemented only in presence of a variable valve acting (VVA) system; the idea of the VVA is not new: the first application of the variable timing on a car was on the Alfa Romeo Spider model 1980, whose system allowed to "rigidly move" the intake valve lift profile on the crank angle axis, to vary the overlap period.

The evolution of the variable valve acting system had three main steps:

- **1st generation**: the first VVA system allowed to vary only the valve timing and not to vary the valve lift, since it consisted in a union sleeve with external helical spline and internal straight spline, acted by pressurized oil, which rotates the camshaft up to 25°

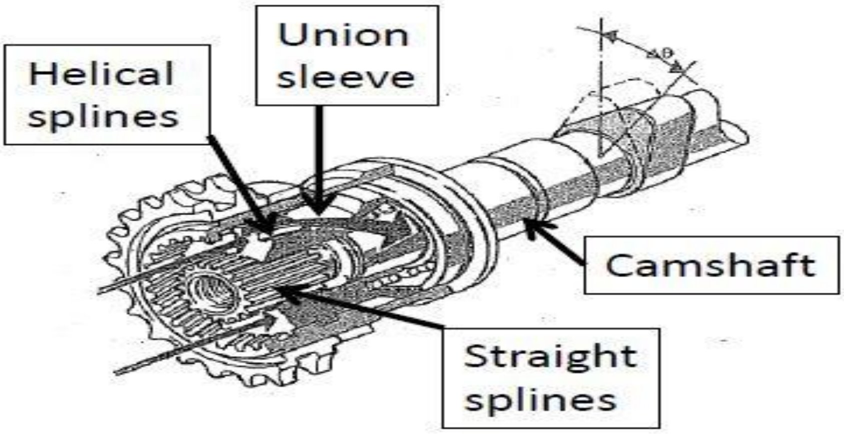

Figure 1.3 1st generation of VVA

- **2nd generation:** The second generation of VVA allows to vary in a discrete way both the timing and the lift of the valve; the mechanism which varies the lift consists in a set of three cams for each intake valve: the central one has a low lift and acts on the internal follower at low engine speed, while the lateral ones have an higher lift act on the external follower at high speed, where the valve lift have to be increased.
  If the two followers are not locked together, the high lift cams push the external one without causing the valve motion, so the valve lift is regulated only by the low lift cam; at high regimes, a hydraulic or an electro-magnetic system blocks the followers together, so the valve lift is regulated by the high lift cams.
  This system was implemented for the first time in 1989 on the engine Honda V-TEC

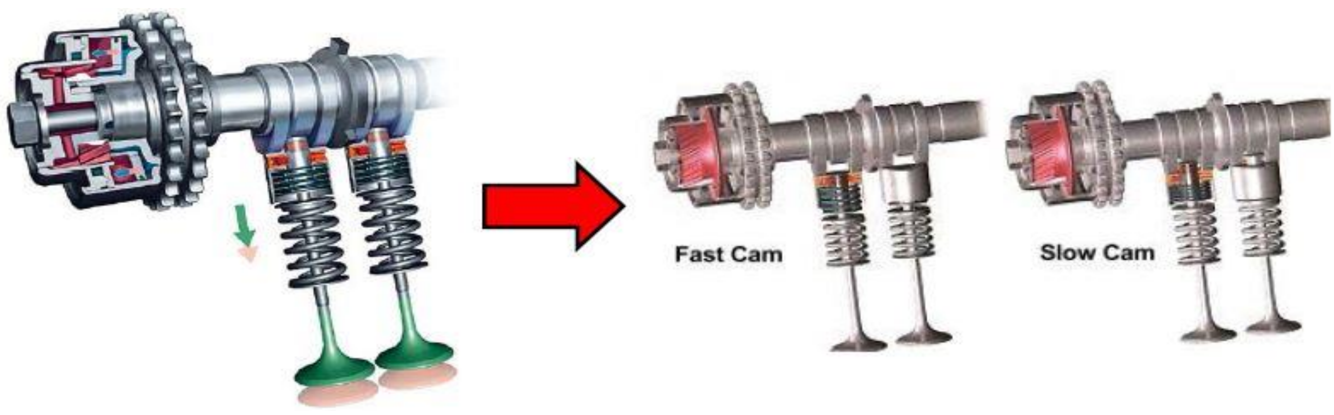

Figure 1.4 2nd generation of VVA

- **3rd generation:** The third generation of VVA is nowadays the state of art of the engine load control: the valve timing and lift can be varied continuously and independently on each other.
The actuation can be mechanical, electro-hydraulic or electro-magnetic: the main examples are respectively the BMW VALVETRONIC and the FIAT MULTI-AIR.

BMW VALVETRONIC:
This system have been created by BMW in 2001 and its main feature is that the cam doesn't act directly on the valve's rocker arm but moves it through a follower whose position with respect to the cam can be varied by an electric motor: this change of relative position allows to transmit a different lift to the valve, at the end of the kinematic chain; the maximum lift transmitted to the stem can be reduced up to one tenth from its full load value.
A spring ensures the contact between the cam and the follower.
The rocker's position can be continuously varied, so the valve lift can vary in a continuous way; all the clearances are recovered by means of a hydraulic group.

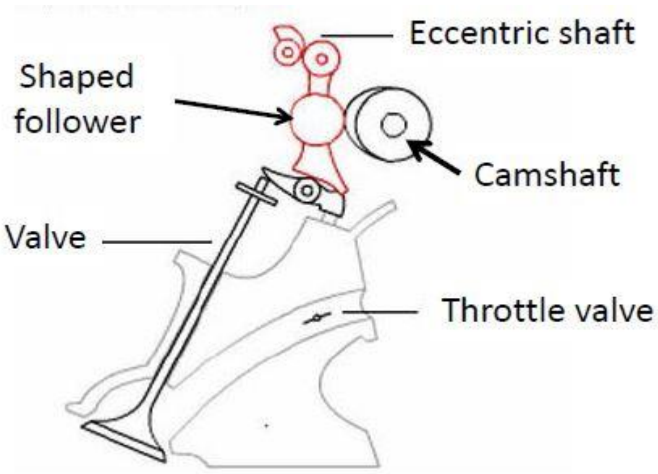

Figure 1.5 3rd generation of VVA

FIAT MULTI-AIR:
This system has been patented by FIAT in 2009, and its main innovation is the use of pressurized oil to transmit the motion in the valve's kinematic chain.
The intake valve's cam acts on a hydraulic piston which pressurizes the oil to move a second plunger which effectively acts the valve.
The oil is incompressible, so it behaves like a rigid body: the lift imposed by the cam to the first plunger is fully transmitted to the valve by the second one.
When the valve lift has to be reduced, the oil is sent to an oil chamber by the opening of a solenoid valve controlled by the ECU, so the cam's motion is no more transmitted to the secondary plunger or is transmitted with a lower lift.
The anticipated or delayed opening of the solenoid valve ensures the early or late inlet valve opening.
This system allows to implement several strategies of valve acting control and load reduction: EIVC, LIVC, LIVO (Late Inlet Valve Closing) and Multi Lift (the intake valve opens more than one time during the intake stroke, to improve air-fuel mixing, it's obtained by a multiple opening of the solenoid valve).

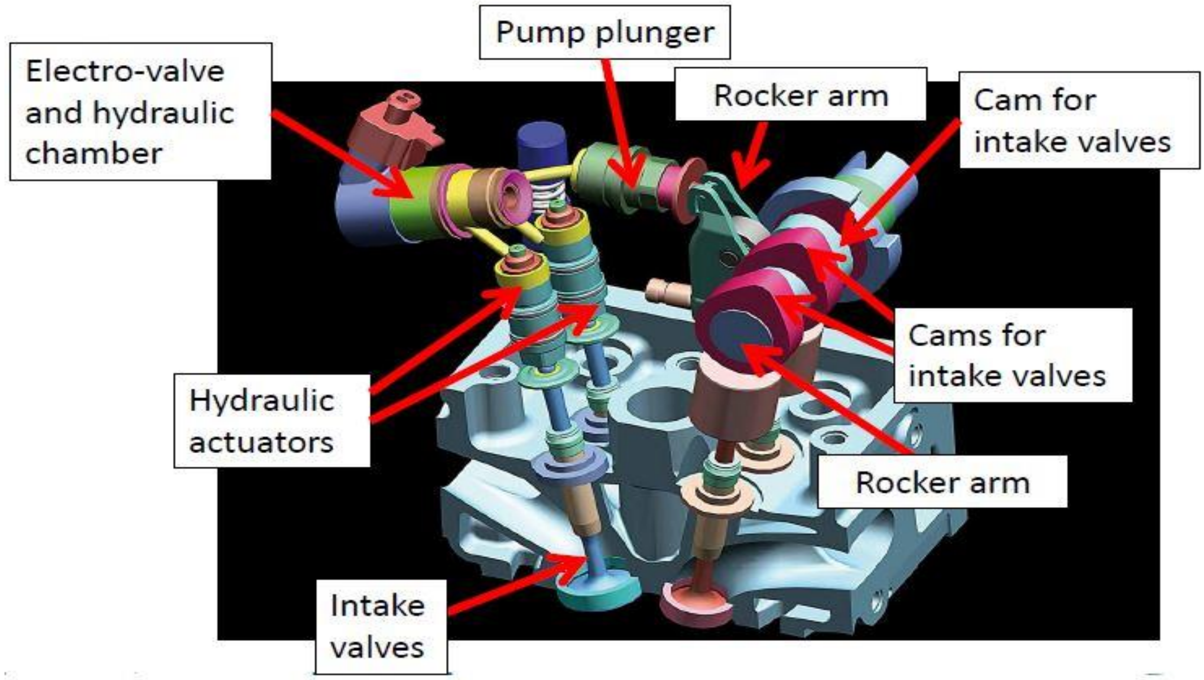

Figure 1.6 Fiat Multi Air system,

1.5 List of contents:

This first chapter regards the introduction to the whole work, with a focus on the importance of the reduction of the specific fuel consumption of the internal combustion engines and the explanation of the alternative partializing methods, completed by the description of the current state of art about their implementation.

The second chapter of this work is focused on the theory of Miller cycle, a thermodynamic cycle whose efficiency is higher than the one of an Otto cycle with the same input parameters and whose main feature is to have the intake stroke shorter than the expansion one: the early intake valve closing allows to implement it, since the intake stroke effectively ends when the intake valve is closed and so becomes shorter than the expansion stroke.

The third chapter is related to the description of the software used to carry out the whole analysis and the demonstration of its reliability, obtained by means of the comparison of the results it provides after the analysis of a simple single-cylinder engine and the ones given by a similar software largely used for the 1-D simulation after the study of the same engine.

The fourth chapter regards the analysis of the naturally aspirated engine controlled by means of the throttling valve, the EIVC and the LIVC strategy at high, medium and low load, aimed to the comparison between the volumetric efficiency, the indicated one and the brake specific fuel consumption generated by each method, needed to understand which is the best one.

The fifth chapter performs the same analysis carried out in the fourth one on the turbocharged engine, with the addition of another method of load control which consists in the lowering of the boosting pressure; even in this chapter the four strategies are implemented to obtain three levels of power reduction and the comparison of the obtained results is used to understand which system is the best one.

The sixth chapter is the final one and contains the synthesis of the main results obtained in the whole work.

1.6 Conclusion and motivation of study

This introduction shows the importance of improving the efficiency of S.I. engines, in particular at low load, and then explains the main reason of the efficiency losses at full and partial load, showing in detail the role of the pumping losses.

After a short overview of the EIVC and LIVC techniques of variable valve acting, the evolution of the mechanical devices needed to implement them is briefly shown, up to the state of art represented by the FIAT MULTI-AIR.

The aim of this thesis is to analyse the benefit given by variable valve acting on two different engines: the first is a four-cylinder 2000 cc 16V naturally aspirated engine, the second is a four-cylinder turbo-charged engine.

On both the engines, in correspondence of one or two precise engine speed, a comparison of the BSFC and indicated efficiency between the case of load reduction obtained by throttling valve and the ones obtained with EIVC and LIVC will be made, for different levels of load reduction: the objective is to find the intake lift law for each case under analysis and to demonstrate that the load control made by means of variable valve acting is more efficient than the one done with the throttling valve.

Chapter 2

Theory of Miller Cycle

2.0 Overview:

In this chapter the Miller cycle will be shown in terms of thermodynamic and in terms of modes of realization. The chapter starts with a recall of the main working principle of a S.I. engine, in order to study the principal reason of its low thermodynamic efficiency, i.e. the incomplete exploiting of the burned gases' energy during the expansion stroke; the principle of Miller Cycle will be introduced starting from this problem.

In the second part, the strategy which can be used to realise the Miller Cycle starting from an Otto engine using a variable valve action will be shown, with a focus on advantages and disadvantages of EIVC and LIVC strategies.

2.1 S.I. engine's working principle

The S.I. engine is based on the Otto cycle, which is a power cycle invented by Nikolaus Otto in 1869 and composed by four thermodynamic transformations:

1. 1→2 Isentropic compression of the fresh charge
2. 2→3 Isochoric heat addiction given by the combustion
3. 3→4 Isentropic expansion of the burned gas
4. 4→1 Isochoric heat rejection (blowdown after exhaust valve opening)

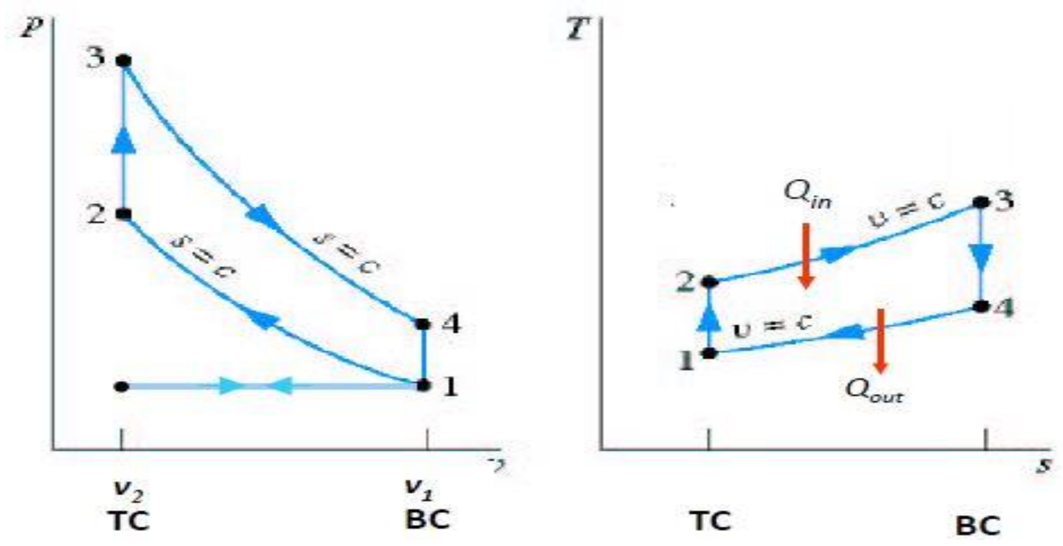

Figure 2.1 Otto cycle's P-V and T-S diagrams

This cycle is implemented by means of a piston which travels alternatively inside a cylinder in order to draw the air- fuel mixture inside the cylinder itself, compress it, expands the burned gases and draw them out.

The Otto cycle can be realised in four-stroke of the piston (four-stroke engine) or in two strokes (two stroke engines); the following discussion will be based only on four stroke engines.

The four-stroke engine performs an Otto cycle with these four phases:

- **Intake stroke:**
  The piston moves from the TDC to the BDC, and the intake valve is open, to fill the cylinder with fresh air-fuel mixture.

- **Compression stroke:**
  The piston moves from BDC to TDC, and both the intake and exhaust valves are closed: the fresh charge is compressed in an adiabatic way (the time to exchange heat with the ambient is too low).
  Towards the end of the compression stroke, the combustion is started by the spark, and the pressure in the combustion chamber rapidly rises at constant volume.

- **Expansion stroke:**
  The piston is pushed to the BDC by the pressure, and the charge expands in an adiabatic way; the expansion work done on the piston, which is almost five times higher than the one spent by the piston itself to compress the air-fuel mixture, is exchanged with the crankshaft which transmits the motion, the torque and the power to the vehicle's wheel by means of the driveline.
  The exhaust valve opens 40-60° c.a. before the piston reaches the BDC, when the burned gases' pressure is still higher than the atmospheric one, to start the gas exhaust process thanks to the blowdown (the pressure is reduced at constant volume) and reduce the pressure into the cylinder, avoiding an excessive work to the piston during the exhaust stroke.
  The incomplete exploiting of the expansion work is a source of efficiency loss, as it will be seen later.

- **Exhaust stroke:**
  The piston moves to the TDC to push the burned gas out of the cylinder; the exhaust valve remains open 10-30° c.a. after the TDC and the intake one opens 10-40° c.a. before the TDC: the period when both the valves are open is called overlap period, in which the inertia of the burned gas exiting from the cylinder is used to draw the fresh charge into the cylinder itself.

The efficiency of the Otto cycle can be calculated dividing the extracted work by the given heat:

$$\eta = \frac{W}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} = 1 - \frac{T_1}{T_2} = 1 - \frac{1}{r^{k-1}}$$

Because the cycle is symmetric and so the property according to which $T_1T_3 = T_2T_4$ is valid (it can be demonstrated applying the equation of the politropic transformation to each couple of transformations) and the compression from 1 to 2 is isentropic, so is possible to use the politropic equation to describe that transformation:

$$P_1V_1^k = P_2V_2^k \rightarrow \frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^k \rightarrow \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{1-k} = r^{1-k}$$

The formula states that the engine efficiency increases at the increasing of the compression ratio, which is equal to the expansion ratio because of the engine's constructive scheme: the higher is the expansion ratio, the higher is the work done on the piston by the burning gases, which can expand more and so convert a larger part of their internal energy into mechanical work.

The more the expansion ratio is increased, the more the combustion's energy is exploited and consequently the higher is the efficiency.

As previously said, because of the constructive scheme of the internal combustion engine, the increase of the expansion ratio implies an increase of the compression ratio, which cannot be increased over 14 to avoid the risk of auto-ignition and knocking, two phaenomena which decrease the performance and damage the engine

The main limit of the internal combustion engine is that the expansion work generated by the expanding burned gas during the expansion stroke is not fully exploited, since the exhaust phase starts before the pressure reaches the atmospheric value.

This problem is generated by the constructive scheme of the engine itself: since the combustion generates a pressure increase from the end of the intake stroke to the beginning of the expansion one, in order to have the atmospheric pressure when the piston is at the BDC, the expansion stroke must be longer than the intake one, but this is made unfeasible by the ordinary centred crank-conrod mechanism which imposes the equality of direct and reverse stroke.

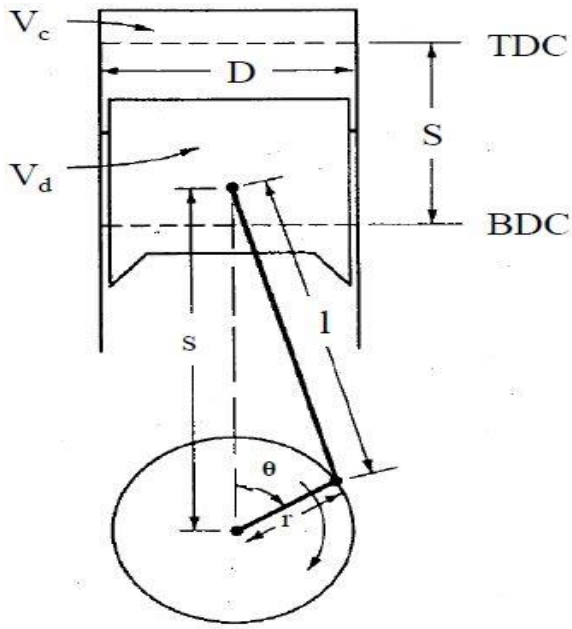

Figure 2.2 scheme of a crank-conrod mechanism [1]

The only way to improve the thermodynamic efficiency of the S.I. engine is to make the compression ratio be lower than the expansion one: this allows to increase the expansion ratio, and consequently the efficiency, without risk the knocking.

This can be done varying the geometry of the engine or using a variable valve train.

2.2 The Miller cycle

2.2.1 Introduction

The first example of engine characterized by an intake stroke lower than the expansion one was created by Atkinson, who in 1889 invented his cycle adding a two-bar linkage between the crankshaft and the connecting rod, which allowed the piston to travel through four unequal strokes in every crankshaft revolution.

The disadvantage of this system consisted in the increased mechanical complexity and friction losses (due to the longer expansion stroke).

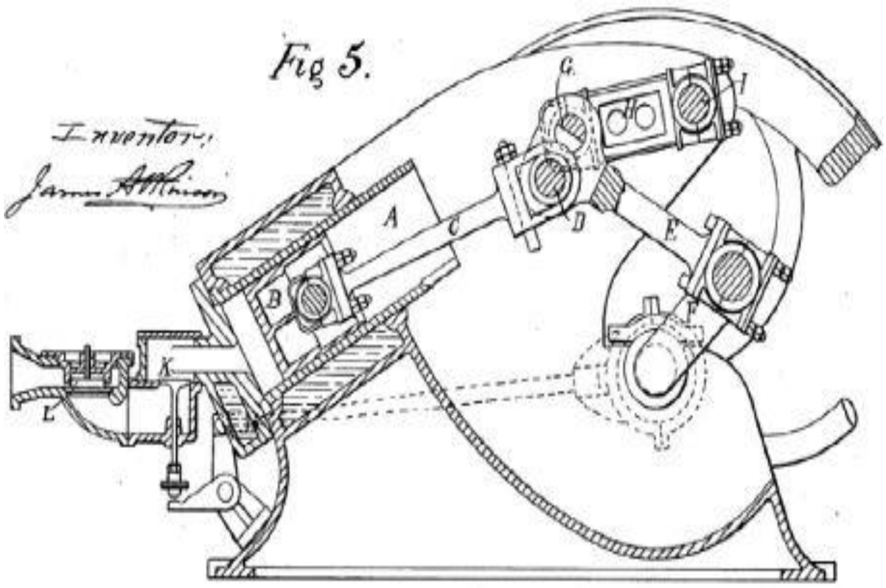

Figure 2.3 Atkinson engine [3]

The simplest way to obtain an expansion stroke longer than the intake one is to reduce the effective intake stroke, that is reduce its portion effectively used to draw the fresh charge into the cylinder: the intake valve may be closed early to draw a lower amount of charge into the cylinder (EIVC strategy) or late, to push the charge in excess back to the intake manifold (LIVC strategy).

The first strategy results in a downscaled lift profile, the second one in the traslation of the lift curve's descendant part in the rigth direction (higher crank angle).

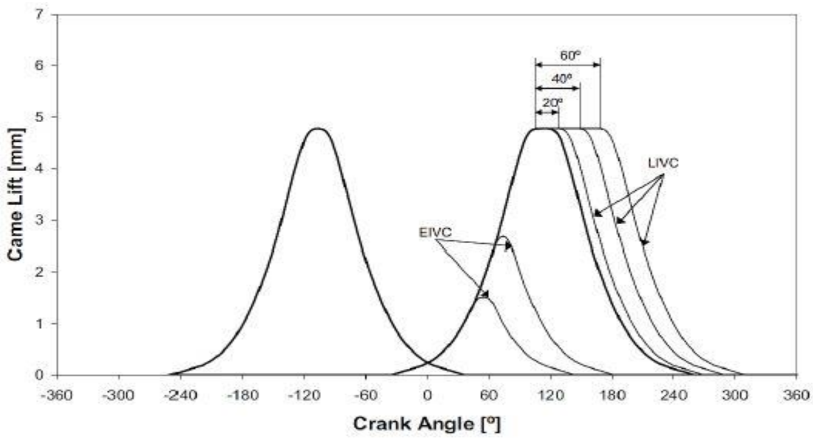

Figure 2.4 comparison between EIVC and LIVC valve profiles [2]

This management of the valvetrain allows to realise the Miller cycle, as it will be seen below.

2.2.2 Thermodynamic transformations

The Miller cycle is almost equal to the Otto cycle, but the compression of the fresh charge starts from a pressure lower than the atmospheric one, because the volume swept during the effective intake stroke is lower than the one swept during the exhaust stroke, so the compression ratio is lower than the expansion one, and consequently the Miller cycle is an over expanded cycle: the higher efficiency with respect to the Otto cycle is given by the better use of the expansion work which can be made by the burning gases, since they can expand until the atmospheric pressure is reached.

The P-V and the T-S diagrams have one more point with respect to the respective diagrams of the Otto cycle, since the compression of the fresh mixture starts at a volume lower than the one at which the expansion of the burned gases ends.

The Miller cycle is composed by the same transformation of the Otto cycle with the addition of a secondary heat rejection which occurs during the ineffective part of the intake stroke (the initial expansion of the fresh charge when the piston approaches the BDC in case of early inlet valve close strategy), or during the rejection of the mixture in the first part of the compression stroke in case of late inlet valve closing strategy [2]:

1. 1→2 Isentropic compression of the fresh charge
2. 2→3 Isochoric heat addiction given by the combustion
3. 3→4 Isentropic expansion of the burned gas
4. 4→5 Isochoric heat rejection (blowdown after exhaust valve opening)
5. 5→1 Isobaric heat rejection
6. 1→6 isobaric exhaust

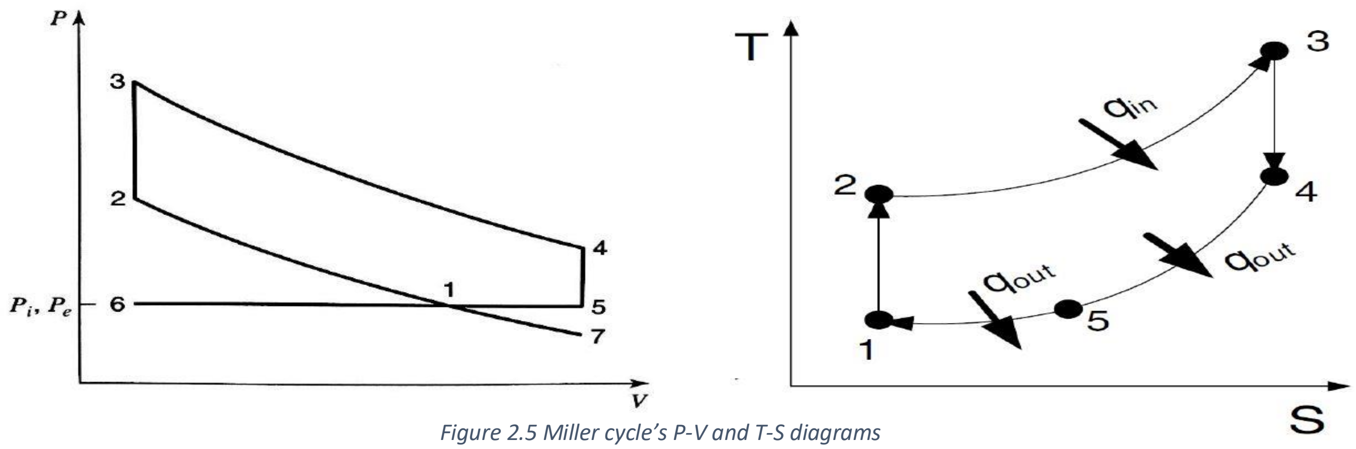

Figure 2.5 Miller cycle's P-V and T-S diagrams

As previously seen, the Miller cycle can be obtained with the EIVC strategy or with the LIVC one: in the first case, the volume increases at constant pressure from 6 to 1, then the intake valve close and the air-fuel mixture expands until the BDC is reached (point 7); the work done by the expanding fresh charge on the piston from 1 to 7 is exactly counterbalanced by the one spent by the piston to compress the fresh charge to the atmospheric pressure from 7 to 1, so the net work is null.

In the second case, the whole intake stroke is exploited to draw the air-fuel mixture into the cylinder from point 6 to point 5, but the intake valve remains open also in the first part of the compression stroke: the volume decreases from V_5 to V_1 but the pressure remains constant until the intake valve close (point 1); even in this case, the net work done in the path 1→5 and 5→1 is null.

It has been proven [4] that the EIVC strategy gives a lower specific fuel consumption with respect to the LIVC, since the second one is characterized by a pumping work which must be spent to push a portion of the fresh charge back to the intake manifold and by a loss of useful work related to the same process, as it will be confirmed in chapters 4 and 5.

2.2.3 Thermodynamic efficiency of Miller cycle

The thermodynamic efficiency of the Miller cycle is calculated dividing the useful work by the input heat; the main difference with respect to the Otto cycle is that the heat rejection is divided in two transformations, the first one is isochoric and the second one is isobaric.

$$\eta = \frac{W}{Q_{in}} = \frac{Q_{in} - Q_{out}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{c_v(T_4 - T_5) + c_p(T_5 - T_1)}{c_v(T_3 - T_2)} = 1 - \frac{T_4 - T_5}{T_3 - T_2} - \Upsilon\frac{T_5 - T_1}{T_3 - T_2}$$

$$\eta = 1 - \frac{T_4\left(1 - \frac{T_5}{T_4}\right)}{T_3\left(1 - \frac{T_2}{T_3}\right)} - \Upsilon\frac{T_1\left(\frac{T_5}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} = 1 - \frac{1}{r_e^{\Upsilon-1}}\frac{\left(1 - \frac{T_5}{T_4}\right)}{\left(1 - \frac{T_2}{T_3}\right)} - \frac{\Upsilon}{r_c^{\Upsilon-1}}\frac{\left(\frac{T_5}{T_1} - 1\right)}{\left(\frac{T_3}{T_2} - 1\right)}$$

Where $\Upsilon = \frac{c_p}{c_v}$, r_e is the expansion ratio and r_c is the compression ratio.

The formula confirms that the efficiency increases at the increasing of the expansion ratio, whose term is at denominator with an exponent higher than 1 since c_p is always higher than c_v $(c_p = c_v + R)$; this is coherent with the fact that the higher is the expansion ratio, the more the burned gases expand and consequently the more is the work extracted.

2.2.4 Implementation of Miller cycle

The Miller cycle is implemented in four strokes of the crankcase, as for the Otto one:

EIVC strategy

- **Effective intake stroke:**
  The piston moves from the TDC to the BDC, and the intake valve is open: the air is drawn into the cylinder until the valve closes (before the piston approaches the BDC)

- **Expansion of the fresh charge**:
  The piston is still moving from the TDC to the BDC, but the intake valve is now closed: the fresh charge already drawn into the cylinder expands in an adiabatic way because of the volume increase.

- **Compression stroke:**
  The piston moves from BDC to TDC, and both the intake and exhaust valves are closed: the fresh charge is compressed in an adiabatic way (the time to exchange heat with the ambient is too low). Towards the end of the compression stroke, the combustion is started by the spark, and the pressure in the combustion chamber rapidly rises at constant volume.

- E**xpansion stroke:**
  The piston is pushed to the BDC by the pressure, and the charge expands in an adiabatic way; the expansion work done on the piston is exchanged with the crankshaft which transmits the motion, the torque and the power to the vehicle's wheel by means of the driveline.
  Differently from the Otto cycle, the pressure at the end of the expansion stroke is close to the atmospheric one, so there's no need to anticipate with a high crank angle the exhaust valve opening, since there's no more blowdown of the burning gases which was previously needed to avoid a high pumping work to the piston during the exhaust stroke.

- **Exhaust stroke:**
  The piston moves to the TDC to push the burned gas out of the cylinder; the exhaust valve remains open 10-30° c.a. after the TDC and the intake one opens 10-40° c.a. before the TDC: the period when both the valves are open is called overlap period, in which the inertia of the burned gas exiting from the cylinder is used to draw the fresh charge into the cylinder itself.

LIVC strategy

- **Intake stroke:**
  The piston moves from the TDC to the BDC to draw the air into the cylinder; the intake valve now remains open for all the duration of the intake stroke like in the Otto cycle.

- **Expulsion of the excess of charge:**
  The piston moves from BDC to TDC, but the intake valve is still open: the pressure remains constant while the volume decreases, and a portion of the air previously drawn into the cylinder is pushed back to the intake manifold; this process continues until the intake valve closes.

- **Effective compression stroke:**
  The piston is still moving from BDC to TDC, the intake valve is now closed as the exhaust one: the fresh charge is adiabatically compressed (no time for heat exchange) and when the piston approaches the TDC the combustion is started by the spark, and the pressure increases at constant volume.

- E**xpansion stroke:**
  The piston is pushed to the BDC by the pressure, and the charge expands in an adiabatic way; the expansion work done on the piston is exchanged with the crankshaft which transmits the motion, the torque and the power to the vehicle's wheel by means of the driveline.
- As for the case of EIVC strategy, the pressure at the end of the expansion stroke is close to the atmospheric one, so there's no need to anticipate with a high crank angle the exhaust valve opening, since there's no more blowdown of the burning gases which was previously needed to avoid a high pumping work to the piston during the exhaust stroke.

- **Exhaust stroke:**
  The piston moves to the TDC to push the burned gas out of the cylinder; the exhaust valve remains open 10-30° c.a. after the TDC and the intake one opens 10-40° c.a. before the TDC: the period when both the valves are open is called overlap period, in which the inertia of the burned gas exiting from the cylinder is used to draw the fresh charge into the cylinder itself.

2.3 main applications and advantages of the Miller cycle

The Miller cycle has a higher efficiency than the Otto cycle, so its main use is at partial load, where the fuel consumption must be reduced as much as possible, while is not used at full load, since a lower mass of fresh charge would be drawn in the cylinder.

One of the newest applications of the Miller cycle is on the hybrid vehicles' engines, because of its higher efficiency at partial load and because it allows to have high geometric compression ratios with a low risk of knocking, since the simple reduction of geometric compression ratio reduces the engine's efficiency, as previously seen in the explanation of the Otto cycle's working principle; one real case of this concept can be found in the Toyota Prius, the first hybrid car launched on the market in 1997 [3].

Another important example of application of the Miller cycle is the MAHLE engine, which is a 1.4 L direct injection gasoline engine which applies it by means of an asymmetric valve timing control which implements the LIVC strategy (the first intake valve is fixed, the second one is adjustable) using the CamInCam® technology, whose main characteristic is that the intake and the exhaust camshafts are coaxial and the first is inserted in the second: both the camshaft can be controlled independently one from the other, so the valve timing of the intake and exhaust valves can be modified separately; this solution requires a lower room in the cylinder head with respect to the two separate camshafts [2].

The Miller cycle has been also used by Nissan in a 1.2 L inline three cylinders supercharged gasoline engine, to reduce the pumping losses at partial load and to avoid the risk of knocking given by the compression ratio equal to 13; the risk of knock is also reduced by the increased turbulence obtained by means of a specific design of the intake port, which decreases the combustion duration and consequently the risk of knock (the more the combustion lasts, the more the air-fuel mixture is at conditions of temperature and pressure which favourite the auto ignition [5].

A second advantage given by the Miller cycle, in particular if implemented by means of a EIVC strategy, is the reduction of the NOx, whose production is directly proportional to the combustion temperature: the adiabatic expansion of the fresh charge after the intake valve closure implies a cooling of the fresh charge itself; since an excessive reduction of the air-fuel mixture temperature makes the combustion unstable, the fresh charge is heated in a heat exchanger in which it exchanges heat with the cooling water or with the exhaust gases.

The improvement of the efficiency is also given by the absence of the pumping losses generated by the throttling valve, as it has been explained in the introduction: the air-fuel mass which enters into the cylinder is no more reduced by the restriction of the effective area through which the air passes, but is reduced lowering the part of the intake stroke where the air is effectively drawn into the cylinder; this leads to a compression ratio lower than the expansion one, and consequently to a Miller cycle.

One critical point of the LIVC strategy of the Miller cycle is that the tumble and swirl vortexes are disturbed by the rejection of the air through the intake valve during the first part of the compression stroke: this problem can be solved by means of a swirl control valve as in the case of Nissan Pure Drive [5] or using an independent action for each intake valve if there are more than one intake valve per cylinder, in order to create the vortexes by the interaction of the air-fuel flows.

The Miller cycle is sometimes coupled with a variable compression ratio system, which further increases the expansion while keeping the geometric effective compression ratio (GECR) constant and consequently increases the efficiency [6]-[7].

2.4 comparison between Miller and Atkinson cycles

As previously seen, the other way to obtain an expansion ratio different from the compression one is to modify the engine from the mechanical point of view, making the real intake stroke effectively lower than the expansion one; this can be made by means of a two-bar linkage between the crankshaft and the conrod, which forms a kinematic chain that modifies the piston's effective stroke, since the two-bar linkage forms a sort of moving crankshaft on which the conrod is mounted and whose motion modifies its relative position with the cylinder head and consequently the piston's stroke.

This is the mechanism of the Atkinson cycle, which has a higher decoupling between the compression and expansion ratios with respect to the Miller cycle and so is capable to guarantee a higher expansion of the burned gases, up to the atmospheric pressure; as a consequence, the thermal efficiency of the Atkinson cycle is higher than the Miller cycle's one [3].

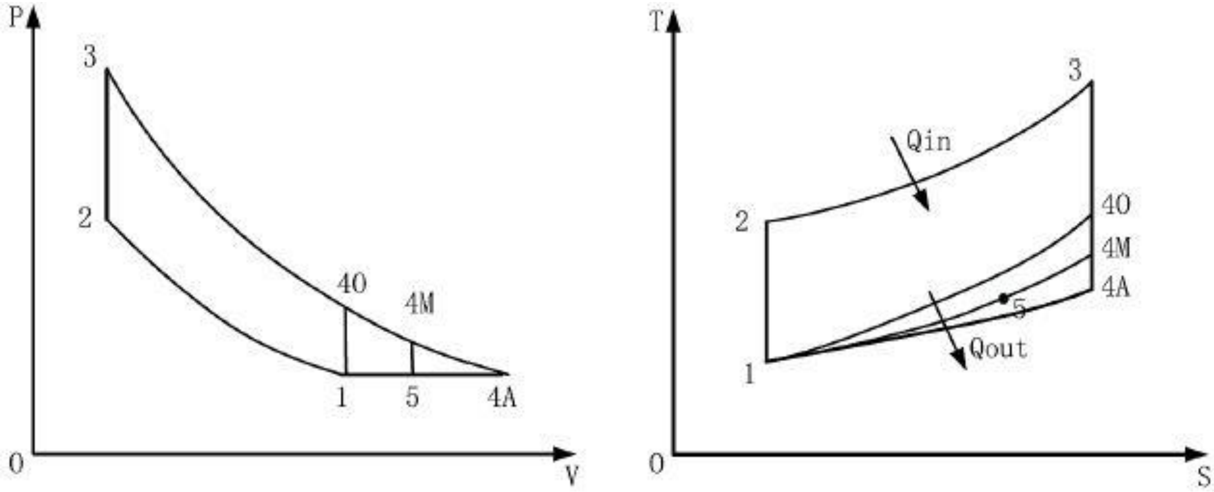

Figure 2.6 Atkinson's P-V and T-S diagram

As previously seen, the higher thermal efficiency of the Atkinson cycle is counterbalanced by the lower mechanical efficiency, because of the higher friction losses given by the longer strokes and the increased mechanical complexity.

Another important difference between the Miller cycle and the Atkinson one is that the first is often equipped with a boosting system, while the second is naturally aspirated [3].

It has been demonstrated [3] that, with the same compression ratio, the Atkinson cycle is more efficient than the Otto and the Miller ones, but its advantage decreases at the increasing of the compression ratio; there's an optimum value of $r_c = 20$ which maximizes the efficiencies od Atkinson and Miller cycles.

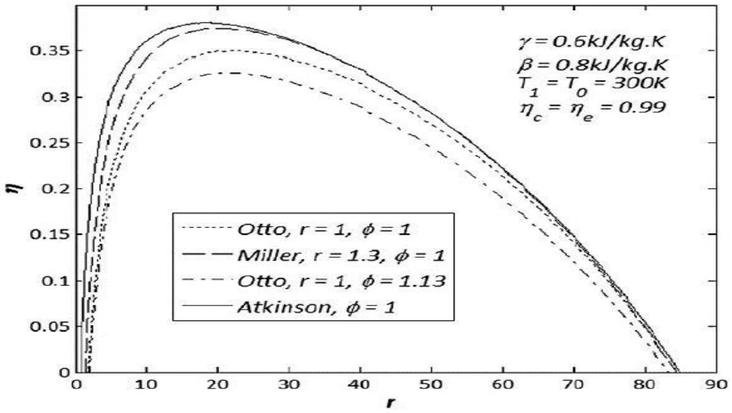

Figure 2.7comparison between Atkinson and Miller cycles

2.5 summary

In this chapter, the Miller cycle has been introduced and its main feature have been shown, starting from a brief description of the Otto cycle and its main sources of efficiency losses, mainly due to the incomplete exploiting of the expansion work of the burning gases.

The best way to improve the efficiency resulted to be the increase of the expansion ratio, but a decoupling of it by the compression ratio is needed, and it can be obtained with a variable valve acting which closes the intake valve early with respect to the original valve timing to draw a lower amount of air into the cylinder (EIVC strategy) or late to push the excess of air back to the intake manifold (LIVC strategy): this leads to an over expanded cycle called Miller cycle.

The formula of the Miller cycle's efficiency has been derived and its implementation has been explained.

In the end, the main advantages and fields of application of the Miller cycle have been shown.

The objective of the following chapters will be to show the improvement of efficiency for different load cases at a precise engine speed, on a 2000 cc four cylinders 16V naturally aspirated engine and on a four-cylinder turbocharged engine.

Chapter 3

Comparison between Gasdyn and Gt Power

3. 0 overview

The aim of this chapter is to show the software which has been used to carry out the whole analysis performed in this work and to demonstrate its reliability by means of the comparison of the results it provides after the analysis of a simple single-cylinder engine with the ones given by a similar software (whose reliability has been certified) after the simulation of the same engine.

The software used to perform the numerical simulations needed to complete the thesis is Gasdyn, a software developed by Politecnico di Milano and commercialized by Exothermia which allows to perform the mono-dimensional simulations of the non-stationary fluxes in the intake and exhaust systems of the internal combustion engines.

Gasdyn is divided into two main parts, the first one, called GasdynPre, is related to the data pre-processing and is characterized by a graphic interface in which there are some icons which represent the templates that must be filled to set the parameters needed by the software to solve the numerical model: the positioning of a certain icon on the worksheet corresponds to the creation of a precise object, whose parameters must be set (for example, the duct object requires the setting of the internal and the external diameter on the left and the right side, the imposition of the mesh size, the wall temperature and many other data); the second part, called GasdynPost, is related to the postprocessing of the results and consists in a simple interface which shows the plots of all the results.

GasdynPre allows to create and simulate whichever engine model by means of the assembly and the setting of all the needed objects: the engine can be naturally aspirated or turbocharged, two-stroke or four-stroke, S.I. or C.I., it can be equipped by a muffler, and many other configurations can be simulated.

The graphic interface of GasdynPre can be seen in the picture below:

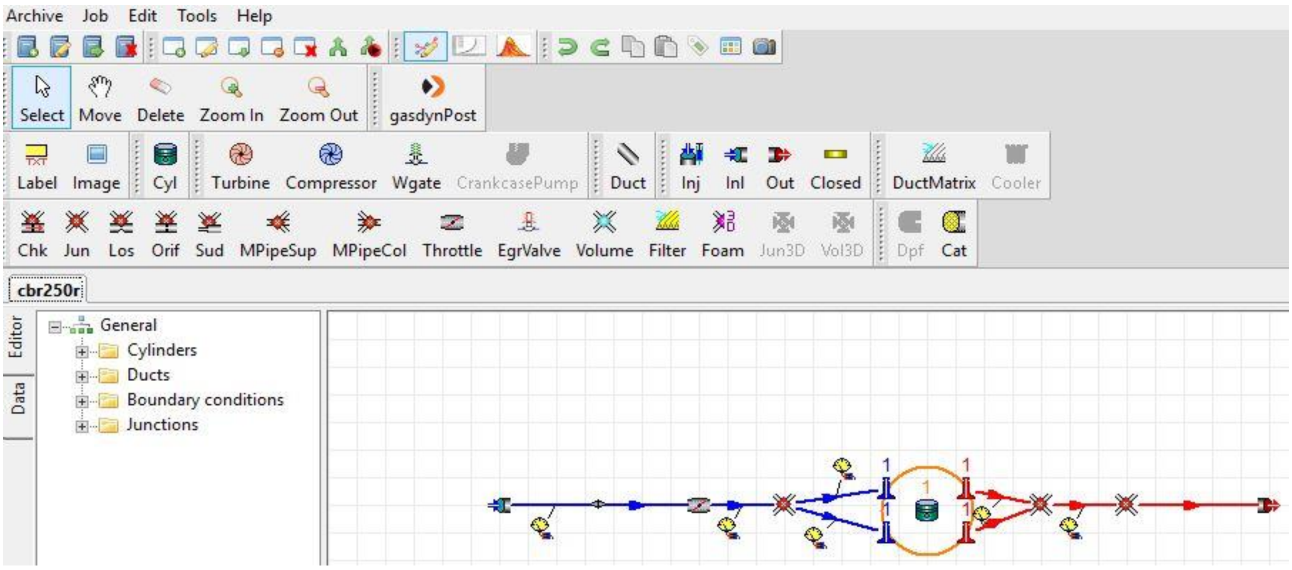

Figure 3.1 Gasdyn's interface

The clearness of the interface makes the model easy to be built, while the parameters must be carefully set in each object.

In order to verify the reliability of Gasdyn, a comparison of its outputs with the ones of the similar software Gt Power is made: the engine under analysis is the Honda Cbr 250 F used by the Polimi Motorcycle Factory team for the Motostudent competition.

Motostudent is an engineering competition attended by student teams from all the world, which design and realize their own motorbike which competes in static and dynamic tests before running in the final race; the competition is organized every two years on the Aragon raceway.

According to Motostudent's rules, only the manifolds' length, the spark advance and the injection law can be modified: the engine cannot be disassembled, the valve timing must be the same of the basis engine, the head and the combustion chamber can't be machined.

Since Aragon is a high-speed track, the length of intake and exhaust manifolds are optimized to have the maximum power at the highest engine speed.

This engine is a one-cylinder four stroke, four valves liquid cooled naturally aspirated engine:

Figure 3.2 Honda CBR 250

The main engine data are reported in the table below; the inlet and outlet ducts are already optimized

Bore	76 mm
Stroke	55 mm
Compression ratio	10.7
Max engine speed	12500 rpm
Inlet duct diameter	38 mm
Inlet duct length	230 mm
Inlet Head duct diameter	30 mm
Inlet Head duct length	60 mm
Outlet duct diameter	33 mm
Outlet duct length	1190 mm
Outlet Head duct diameter	27 mm
Outlet Head duct length	60 mm
Inlet duct temperature	300 K
Inlet Head duct temperature	350 K
Outlet duct temperature	900 K
Outlet Head duct temperature	450 K

All the data have been implemented in Gt Power by Polimi Motorcycle Factory team for the performance analysis; the model has been reconstructed in Gasdyn and the performance curves have been compared.

3.1 Comparison of the mean performances:

The comparison between the two software starts from the superimposition of the curves of the main mean quantities, as brake torque, brake power and volumetric efficiency: if the results given by Gasdyn and Gt Power are not consistent, the simulation's reliability cannot be guaranteed.

The instantaneous quantities calculated at the maximum torque regime will also be compared, to better understand if the fluid-dynamic is the same on both the software.

3.1.1 Brake torque:

The figures below show the brake torque curves and the indicated torque curves calculated by the two software: the discrepancy between them is small and both the curves have the same trend, so their results are concordant.

After the highest peak equal to 27 Nm, the brake torque decreases rapidly with the increase of the engine speed; the same trend can be seen in the indicated torque curve, whose peak is equal to 31.37 Nm.

The indicated torque is maximum when the volumetric efficiency reaches its highest value: from the indicated torque graph it can be understood that the engine speed at which the filling coefficient is maximum is 7500 rpm.

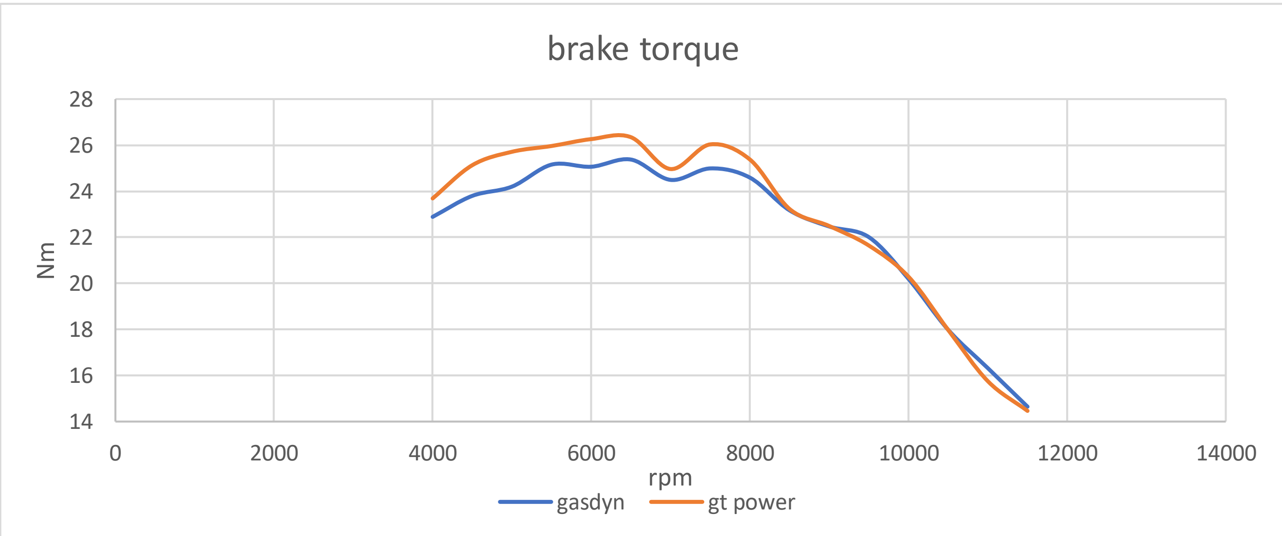

Figure 3.3 Comparison between the brake torques provided by Gasdyn and Gt power

It can be noticed that even if the shape of the curves is the same, they are not exactly coincident, and the torque estimated by Gt Power is always higher than the one given by Gasdyn up to 11000 rpm: this is due to a different management of the heat transfer and the distributed pressure losses between Gasdyn and Gt Power, since the heat transfer coefficient is calculated by The Coulburn's formula on Gt Power and by the Churchill's one on Gasdyn, where the Coluburn's formula is not implemented; in addiction, the increase of the heat transfer coefficient due to the pulsating motion of the fluid cannot be modified on Gt power, and the scaling factor for the pulsating motion has been set on Gasdyn by several trials and errors, looking to the brake power and to the temperature at the inlet of the head intake duct.

The two software calculate the pressure drops in the ducts according to the roughness of the material; even in this case the algorithm is different, since an equal surface roughness gives different results; this problem has been solved setting the duct friction multiplying factor of Gt power to 1.3 after many trials and errors.

3.1.2 Brake power:

The results provided by Gasdyn and Gt Power are concordant also for the brake power, even if the curve provided by Gt-power is slightly higher than the one calculated by Gasdyn up to 8500 rpm, as it has been seen for the brake torque.

The brake power is defined as following:

$$P_b = Bmep * V * \frac{n}{\varepsilon}$$

Where $Bmep$ is the brake mean effective pressure, n is the engine speed, V is the engine displacement and ε is equal to 2 because the cycle is a four-stroke one.

Since the brake power is given by the product between the brake torque and the engine speed, its peak is displaced towards the high regimes, where the engine speed's contribution becomes more important, while the highest positive slope of the curve is in correspondence of the maximum brake torque regime, where the term related to the brake torque is more relevant.

The maximum brake power is equal to 22 kW at almost 10000 rpm while both the curves have another high peak at 8000 rpm, where the brake torque curves have the second peak almost equal to the maximum one: the effects of torque and speed are both relevant, so the brake power is highly increased, remembering that the engine speed which maximizes the filling coefficient is 7500 rpm.

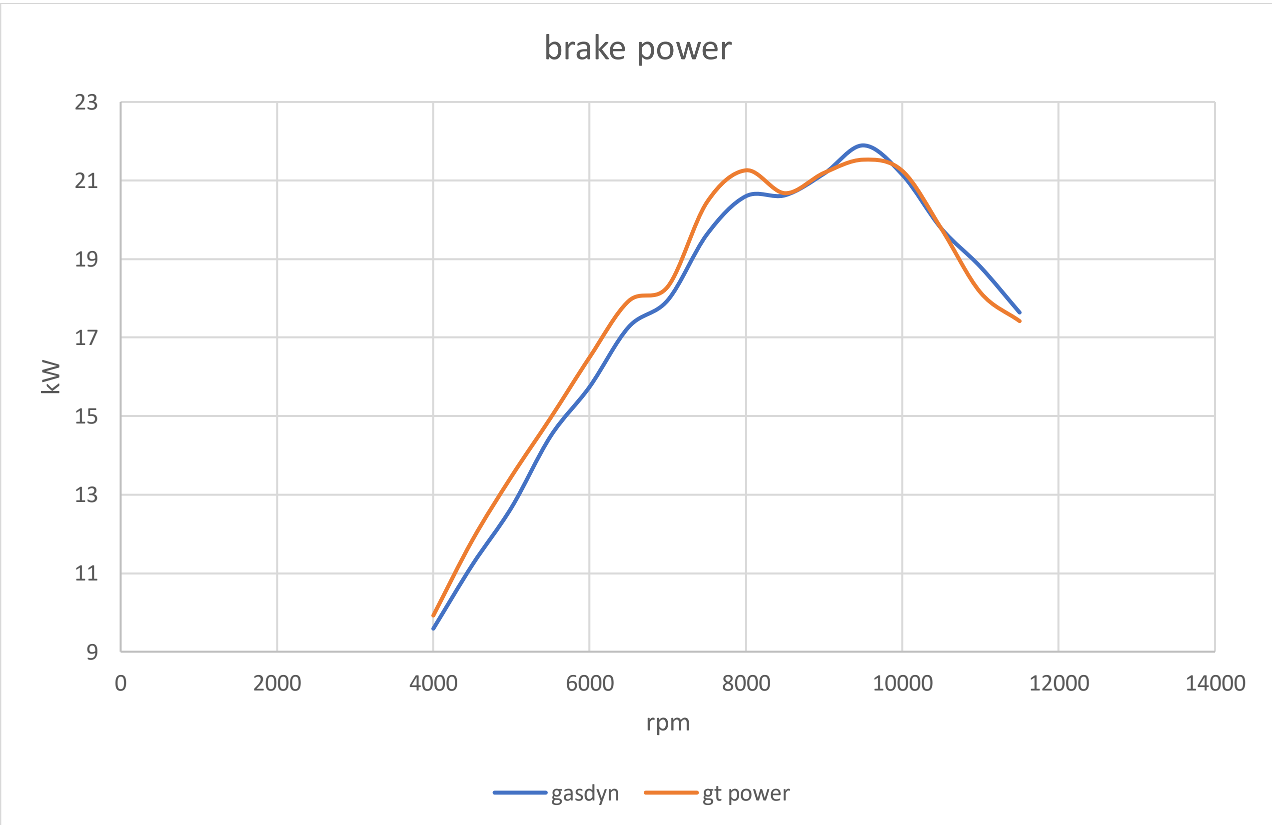

Figure 3.4 Comparison between the brake powers provided by Gasdyn and Gt power

3.1.3 Volumetric efficiency:

The volumetric efficiency is the ratio between the air mass effectively aspirated by the engine and the theoretical one, given by the product between the air density and the cylinder volume:

$$\eta = \frac{m_a}{V_c \rho_a}$$

The volumetric efficiency curves calculated by Gasdyn and Gt Power have the same shape, and the volumetric efficiency is higher than 100% in the engine speed range from 4200 to 10000 rpm: the air and wave effects allow to fill the cylinder with an air mass higher than the product between the air density and the cylinder's volume $(m_a > V\rho_a)$.

The highest volumetric efficiency is reached at 7500 rpm, as it has been understood in the graphs of indicated torque and brake power.

As said before, the setting of the Gt power friction multiplier has been performed by trials end errors looking to the brake torque and brake power diagrams: the results given by the two software in terms of torque and power became coincident when the friction multiplier has been set to 1.3; when that coefficient was set to 1.0, the volumetric efficiency calculated by Gt power was slightly higher than the one given by Gasdyn on all the speed range.

The non-perfect coincidence between the two curves proves again that the distributed pressure drops are treated in a different way on Gasdyn and Gt power, and the trends can be made closer to each other only varying the friction multiplier on Gt Power or imposing a friction coefficient on Gasdyn (after deactivating the Automatic Friction Coefficient option).

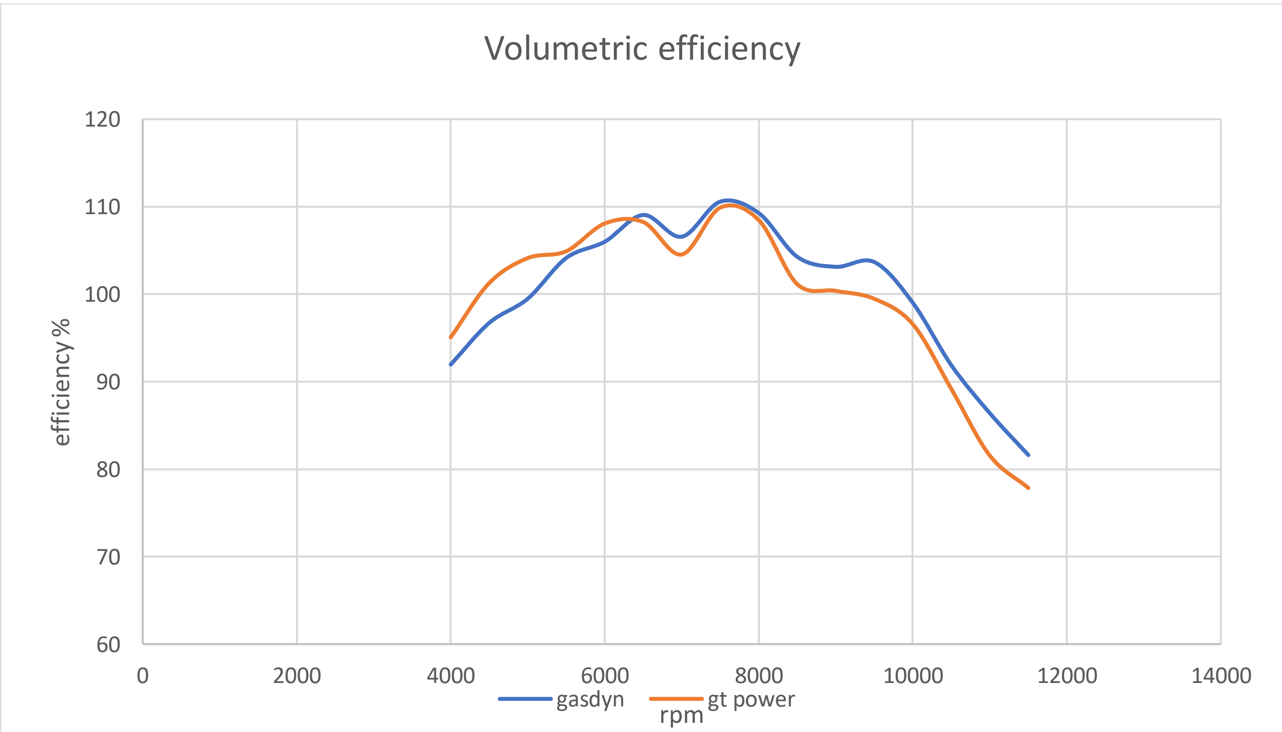

Figure 3.5 Comparison between the volumetric efficiencies provided by Gasdyn and Gt power

3.1.4 Brake specific fuel consumption BSFC:

The brake specific fuel consumption is defined as the ratio between the fuel mass flow rate and the brake power generated by the engine:

$$BSFC = BSFC = \frac{\dot{m}_c}{P_b}$$

Both the software show that the specific fuel consumption expressed by this engine remains almost constant and equal to 65 g/MJ from 4000 to 6500 rpm and after this range it increases linearly with the engine speed up to 95 g/MJ.

Even in this case the curves given by Gasdyn and Gt Power have the same shape, with an almost constant difference of 3 g/MJ because of the different simulation of the combustion process.

It can be also noticed that, differently from all the previous cases, the curve given by Gt Power is below the one calculated by Gasdyn: this confirms again that Gt Power has a model of combustion which provides higher efficiency with respect to Gasdyn's one.

Another reason of the remarkable difference between the curves of specific fuel consumption is that Gt Power is equipped by an algorithm which simulates the presence of a fuel injector, which is not present in Gasdyn, in which the air to fuel ratio is imposed in the panel which manages the general data of the engine: the difference in the solving algorithm increases the gap between the solution provided by the two software, considering that the engine under analysis is equipped by a port-fuel injection.

Figure 3.6 Comparison between the specific fuel consumptions provided by Gasdyn and Gt power

3.1.5 Brake mean effective pressure BMEP:

The curves of the brake mean effective pressure follow the same trend of the brake torque curves: this is not surprising, since the brake torque is related to the brake mean effective pressure by the relation

$$M_e = BMEP * \frac{V}{2\pi\varepsilon}$$

Where V is the engine displacement and $\varepsilon = 2$ because the engine is a four-stroke one.

It can be noticed again the peak at 7500 rpm, where the filling coefficient is maximum, equal to 13.27 bar.

It can be noticed that the curves become coincident from 8500 rpm to the end: this is due to the friction losses on Gt power's model, whose importance increases with the engine speed.

If the Gt power's friction multiplier was set to a value higher than 1.3, the curves would have their initial part equal to each other, but at high engine speed the curve given by Gt Power would be lower than Gasdyn's one, as for the brake torque and brake power curves.

Figure 3.7 Comparison between the brake mean effective pressures provided by Gasdyn and Gt power

3.2 comparison of the instantaneous performances:

After the comparison of the curves given by Gasdyn and Gt power about the mean physical quantities, the same procedure is implemented to compare the instantaneous ones.

Both the codes discretize the engine cycle with a resolution of one crank angle degree, even if Gasdyn starts from 0° and Gt Power starts from -180°, so, after the re-phasing of Gt power’s data arrays, the curves of air’s speed and pressure in the inlet and exhaust duct have been compared.

All the instantaneous quantities are referred to the maximum indicated torque regime (7500 rpm) and are kept at the inlet of the intake head duct.

3.2.1 Air temperature:

The air temperature at the inlet of the intake head duct s substantially the same in Gasdyn and Gt power: the temperature oscillations have the same amplitude and the same frequency, so the modelling of the heat transfer between the pipe walls and the air is the same: this means that the value of the multiplying factor of the heat transfer coefficient which takes into account the fluid’s pulsating motion is correctly set at the value of 5 (the Gasdyn User Manual suggest not to exceed that value)

The lowest peak is in correspondence of the intake valve opening.

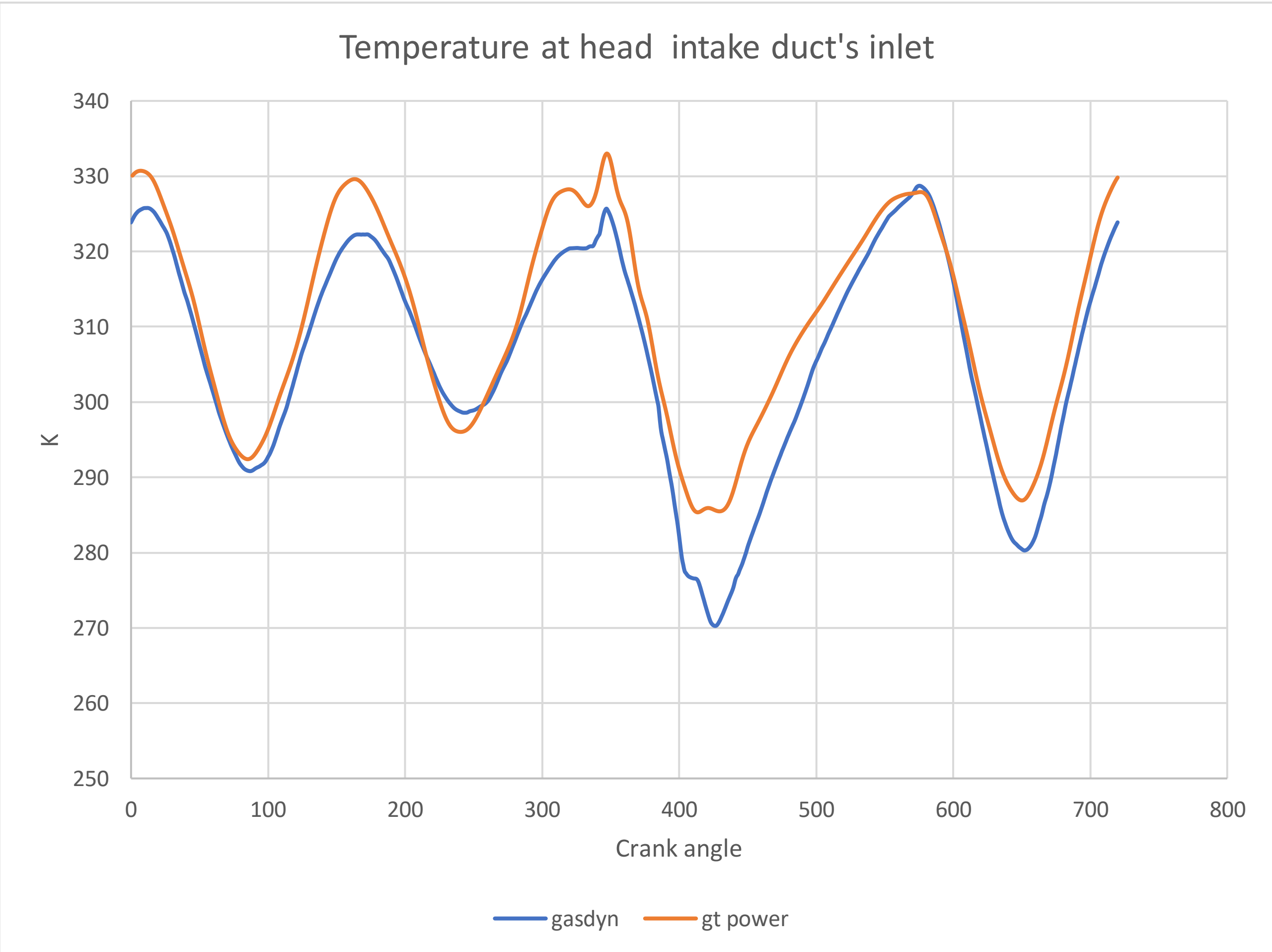

Figure 3.8 Comparison between the instantaneous temparatures provided by Gasdyn and Gt power

3.1.2 Air pressure:

As for the curves of instantaneous temperature, even the ones of instantaneous pressure are coincident: the pressure oscillations have the same amplitude and frequency.

The equality of the frequencies proves that the ducts have the same length on both the software.

The lowest peak is in correspondence of the intake valve open

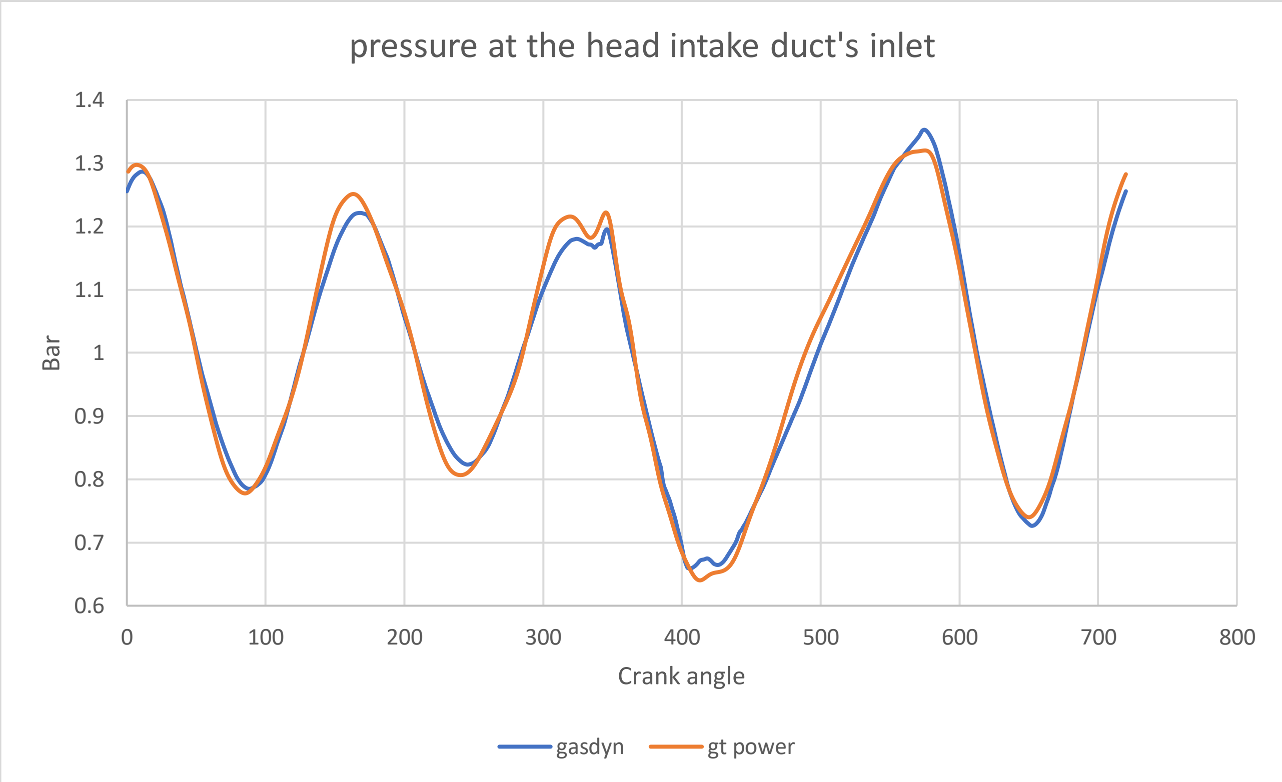

Figure 3.9 Comparison between the instantaneous pressures provided by Gasdyn and Gt power

3.3 Conclusions:

The reliability of Gasdyn has been studied comparing its main results with the ones given by Gt Power, which is a similar 1D modelling software.

The engine under analysis was the Honda CBR 250 F used by Polimi Motorcycle Factory for the Motostudent competition.

The comparison has been made firstly on the mean quantities, and the diagrams of brake torque, brake power, Brake Mean Effective Pressure and specific fuel consumption given by the two codes resulted to be very similar, even if not exactly coincident: this is due to the different algorithms of calculation of pressure drops and heat exchange under pulsating motion.

The strong similarity of the results proves however that Gasdyn is a reliable software.

The analysis has been continued on the instantaneous temperature and pressure: the plots of these quantities resulted to be coincident, demonstrating that the main parameters of the model were the same.

The results of this comparison demonstrate that Gasdyn is a reliable software, and it can be used to study the variable valve acting on a four-cylinder, turbo-charged engine and on a naturally aspirated one.

Chapter 4

Load control in a four-cylinder 2000 cc. 16 valves naturally aspirated engine obtained by means of the Miller cycle.

4.0 Overview:

The aim of this chapter is to make a comparison between the three engine control strategies which have been explained in the previous chapters, that is the classic throttling valve, the Early Inlet Valve Closure (EIVC) and the Late Inlet Valve Closure (LIVC), with a particular focus on the brake specific fuel consumption (BSFC), which is the main quantity to be lowered thanks to the decreasing of the pumping losses.

The engine under analysis is a four-cylinder 2000 c.c. 16 valves naturally aspirated engine, the Alfa Romeo Twin Spark, with a fixed valve action for the intake and the exhaust and an intake duct characterized by a variable geometry.

Figure 4.1 Alfa Romeo Twin Spark engine

The engine is studied under three different load levels: 66%, 27% and 11% of the maximum load, and the values of brake torque, brake power, volumetric efficiency, specific fuel consumption and indicated efficiency given by the three control strategies are compared, as for the pressure-volume diagrams expressed in the logarithmic scale; the case of full load is always used as a reference.

The process starts by the definition of the engine speed at which the load cases have to be analysed: the chosen engine speed is 4600 rpm, at which the engine reaches the maximum power; the successive step is the definition of the full load case at the speed under analysis, setting the engine with the original valve lift and the full throttle.

The following steps consist, for all the load cases, in the research of the optimal valve lift laws which implement the EIVC and LIVC strategies giving the same torque and power output given by the throttle-controlled engine: this research is done by trials and errors, modifying the original valve lift law and simulating the engine with the modified law until the coincidence of the output torque and power is reached.

The EIVC strategy is implemented in this way: the original law is truncated in correspondence of a precise point before the maximum peak, then the point with the equal value of the lift on the descendant part of the lift profile is found and the portion of the curve before the first point and after the second one are joined; lastly, the edge generated by the junction is rounded.

The more the edge is rounded, the lower are the output torque and power, this means that the optimization of the lift law is characterized by two degrees of freedom, the angle at which the law is truncated and the intensity of the rounding: when the "cutting and pasting" of the original lift law made in correspondence of a certain angle gives a torque and a power which are slightly higher than the ones provided by the load reduction obtained by the throttling valve, the edge is more and more rounded until the perfect coincidence of the load cases is reached (the degree of tolerance is 0.5 Nm and 0.5 kW); this method is quite fast and allows to preserve the slope of the original law, avoiding an excessive speed of the valve.

The LIVC strategy is implemented by a simple translation of the descendant part of the original valve lift law to a higher crank angle, without altering the shape of the curve; the angle at which the valve lift starts to decrease is the only optimization parameter of the LIVC strategy, and the given torque and power are extremely sensitive to the variation of the closure delay, since the effective valve area varies up to the 1% after a variation of one degree of the angle delay.

It must be considered that in Gasdyn the valve lift profile are defined on a relative crank angle: all the profiles that will be discussed in this chapter and in the next one start from a crank angle equal to zero, but in the complete model the intake valve opening doesn't start in correspondence of the top dead centre, since the effective angles of intake and exhaust valve opening are set in a specific item; the original angles of inlet and exhaust valve opening are respectively 341° and 111° c.a. and they haven't been modified for all the analysis.

These angles are referred to the camshaft: since Gasdyn takes into account the clearance of the valvetrain, the actual angles of opening and closure in correspondence of which the effective flow area starts to be different from or equal to zero will be different from the theoretical ones imposed in the valve lift profile: for what it concerns the original valve lift, the actual absolute opening angle of intake and exhaust valve are respectively 359° c.a. and 131° c.a., while the closing ones are respectively 597°c.a. and 369° c.a., given the crevice equal to 0.25 mm.

The perfect coincidence of torque and power with the case of engine control obtained by means of a throttling valve is consequently harder to be gained, so the load controls implemented by the throttling valve and the EIVC strategy are slightly adjusted to match the torque and the power given by the load reduction made by the LIVC strategy: this is the reason why the medium and low load configurations are respectively the 27% and the 11% of the full load instead of 25% and 10%.

The comparisons are expressed in percentages for all the cases, to better clarify the improvement given by the Miller cycle on the fuel consumption.

The last step consists in the graphical and quantitative comparison of all the results, with a distinction of open and closed cycle efficiency, from which the best solution is definitively derived.

4.1 The full load case

The definition of the full load case starts by the setting of the engine model, that is setting the throttling valve's opening angle and the intake valve lift law (the rest of the engine remains unchanged, in order to be sure that only the load control strategy influences the performance).

The full load obviously requires a throttling valve's angle equal to 90°, while the intake valve lift remains the original one, whose maximum peak is equal to 9.5 mm and is at 136° of crank angle:

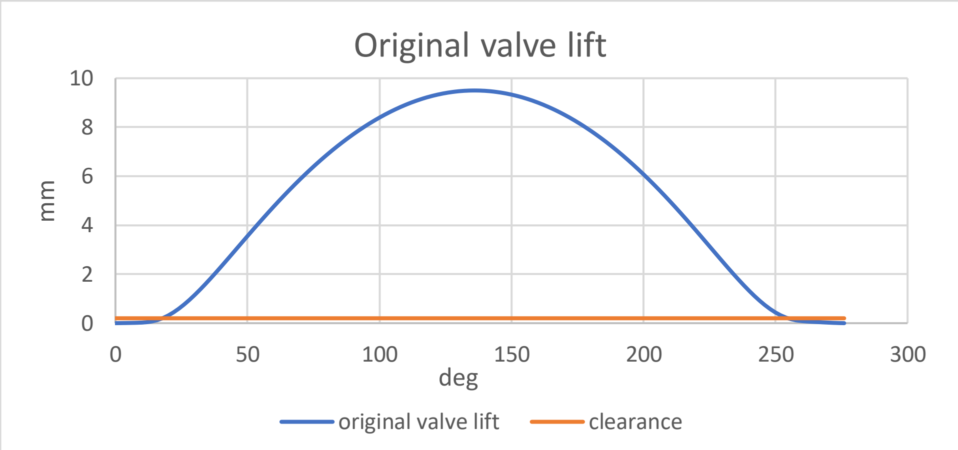

Figure 4.2 N.A. engine's original valve lift

The inlet valve opening (IVO) is at 341° c.a., while the exhaust valve opening (EVO) is at 111° c.a. (the zero of the crank angle scale is at the top dead centre): these quantities will not be modified during the following analysis, since the implementation of EIVC and LIVC strategies regards only the valve closure.

The engine will be always simulated with a stoichiometric air to fuel ratio, which is equal to 14.6, while the combustion process will be analysed by means of the Wiebe function, whose parameters are listed below:

- Efficiency parameter = 6.9
- Shape factor = 2.0
- $\Theta_{50} = 8.0$
- $\Delta\Theta_{10} = 25$

In order to decide the most suitable engine speed at which concentrate the comparisons between the control strategies, the brake torque, brake power and volumetric efficiency curves of the original engine have to be analysed:

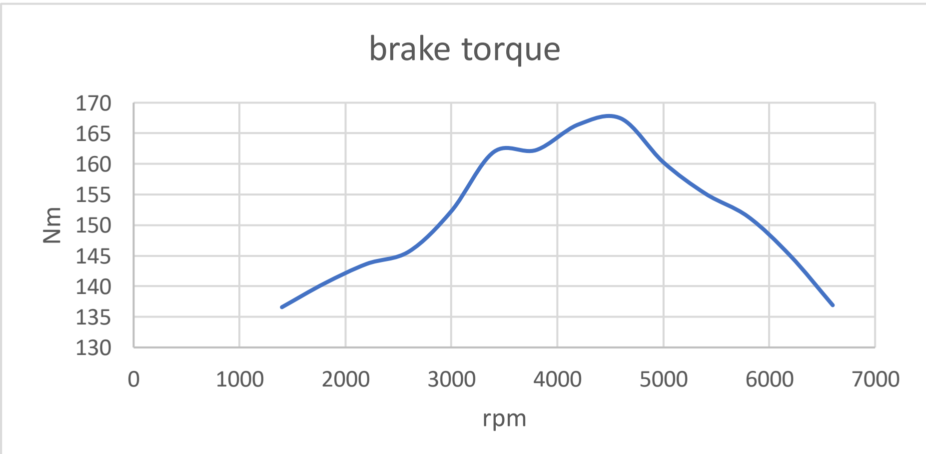

Figure 4.3 N.A. engine's brake torque

The brake torque curve shows that the engine is optimized for the medium regimes: the length of the intake and exhaust manifolds emphasizes the inertial and wave effects in the engine speed range from 3400 to 5000 rpm, in particular at 4600 rpm, which is consequently the maximum torque regime.

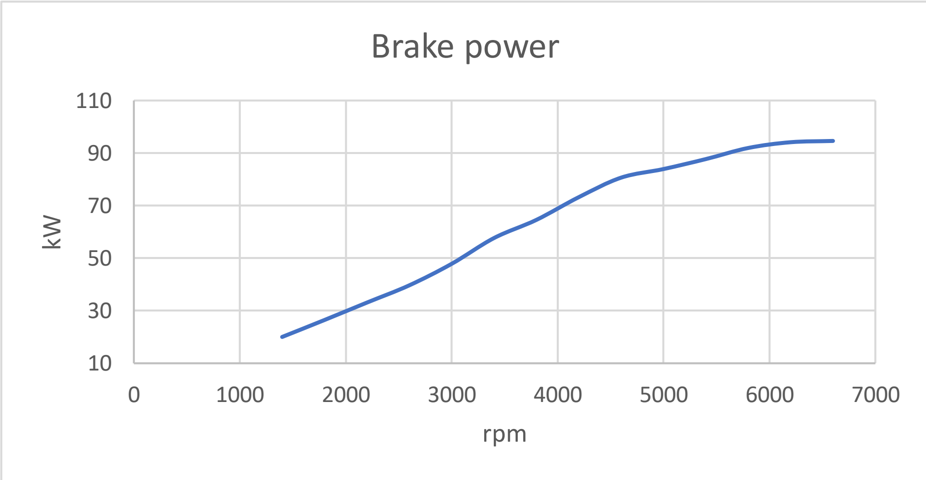

Figure 4.4 N.A. engine's brake power

The brake power curve rises almost linearly from 1400 to 4600 rpm, the peak due to the maximum brake torque at 4600 rpm is clearly visible.

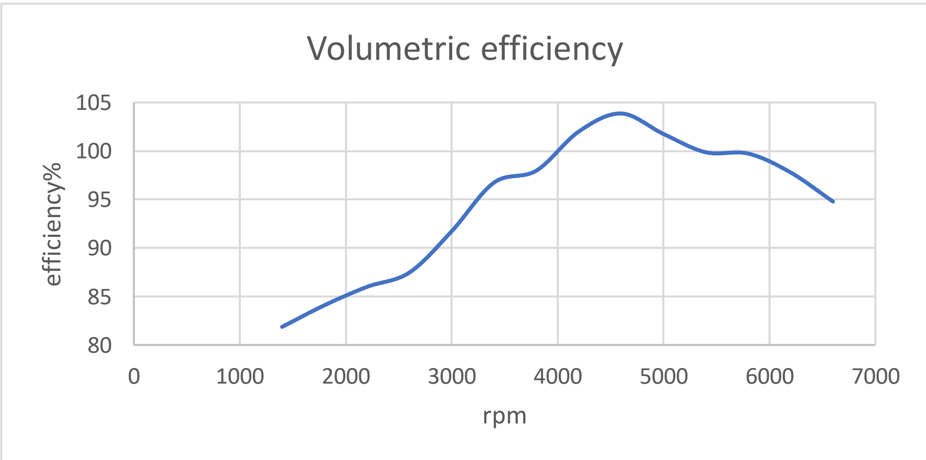

Figure 4.5 N.A. engine's volumetric efficiency

The volumetric efficiency curve has the same shape of the brake torque one, since the air density and the air to fuel ratio are constant along the engine speed range.

The most significative engine speed is the one at which the engine reaches its maximum torque (because the filling coefficient is maximum), that is 4600 rpm: all the following comparisons will be made at this regime.

The main performances of the engine at the maximum brake torque regime resulted to be the following:

Volumetric efficiency	103.831%
Brake torque	167.43 Nm
Brake power	80.653 kW
BSFC	74.424 g/MJ
Indicated efficiency	36.8%

These values will be used as a reference in all the following comparisons.

4.2 Load reduced to 66% of the full one

The first case under analysis is the high load one, that is 66% of the full load ($167.43 * 0.66 = 110.5\ Nm$).

The load reduction will be made by means of the throttling valve, the EIVC strategy and the LIVC one; for each strategy, the method of implementation will be explained, and the main results will be tabulated.

Since the load reduction is not severe, effect of the pumping losses in the throttled engine is expected to be low and consequently the improvement of efficiency given by the variable valve acting will not be high.

In the end, the comparison between the three strategies will be made, with a focus on the main reasons of the difference in the results.

4.2.1 Load reduction obtained by the throttling valve

The load equal to 66% of the maximum one is obtained when the throttling valve is inclined of an angle equal to 30° with respect to the vertical position; it can be noticed that a reduction of two-thirds of the throttling valve's opening angle causes the reduction of only one third of the load: even if the pumping losses are present also in case of low load reductions, their effect becomes strongly relevant only for high restrictions of the throttling valve, when the opening angle is lower than 25° and even a fraction of degree of the opening angle causes a remarkable difference in the engine's performance.

The engine performances obtained with a throttling valve's angle equal to 30° are the following:

Volumetric efficiency	76.394%
Brake torque	110.292 Nm
Brake power	53.129 kW
BSFC	83.127 g/MJ
Indicated efficiency	35.6%

The specific fuel consumption increases of 11.69% because of the pumping losses due to the throttling valve which also reduce the indicated efficiency almost by 3.2%.

The volumetric efficiency decreases by 26.4% with respect to the full load configuration, because of the reduction of the air mass entering into the cylinder.

4.2.2 Load reduction by means of EIVC strategy

As explained in the previous chapters, the mass entering in the cylinder can be reduced anticipating the inlet valve closure and so reducing the effective intake stroke; this leads to an effective compression ratio lower than the expansion ratio and so to a Miller cycle.

The valve lift law which gives the same torque and power than the throttling valve with an opening angle equal to 30° has been found by trials and errors: the peak is now at 99° crank angle and the maximum lift resulted to be equal to 7.482 mm after the rounding of the edge; since the maximum peak of the original valve lift is equal to 9.5 mm, the lift reduction is equal to 21.24%.

The crank angle at which the valve closes is now 202° instead of 276°, which means a reduction of almost 27% of the opening period of the valve; taking into account the crevice, the angle is equal to 180°c.a.: as for the original valve lift, the intake valve effectively closes the intake port 24° before the theoretical lift profile becomes equal to zero.

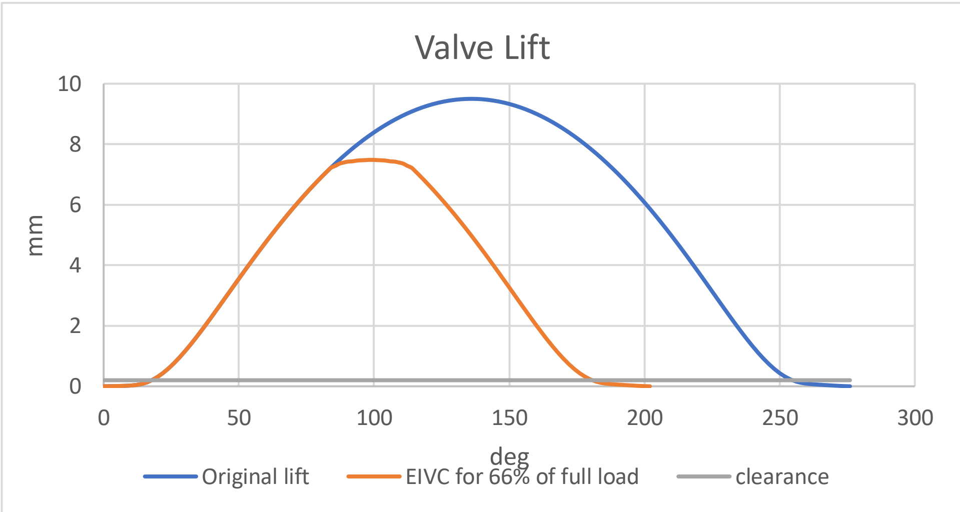

Figure 4.6 EIVC for 66%

The EIVC strategy previously explained gives the following results:

Volumetric efficiency	74.608%
Brake torque	110.739 Nm
Brake power	53.444 kW
BSFC	80.856
Indicated efficiency	36.5%

It can be noticed that BSFC is lower than the one resulting from the load control provided by the throttling valve, with a reduction equal to 2.7%, while the indicated efficiency increases of almost the same percentage (2.5%); this is not surprising, since the BSFC is inversely proportional to the indicated efficiency.

This is due to the reduction of the pumping losses given by the absence of the throttling valve and to the fact that Miller cycle's thermal efficiency is higher than the Otto's one.

The EIVC strategy generates a lower volumetric efficiency with respect to the throttling valve by 2.34%: the lower pumping losses with respect to the throttling valve allow to get the same torque and power burning a lower mass of air-fuel mixture, so a lower quantity of air is drawn into the cylinder and consequently the volumetric efficiency decreases.

4.2.3 Load reduction by means of LIVC strategy

The second way to reduce the mass of air entering into the cylinder at partial load without using the throttling valve is to push the excess of air back to the intake manifold during the first part of the compression stroke, when the piston rises from BDC to TDC; this can be obtained with a delayed closure of the intake valve, that is with the LIVC strategy.

The more the intake valve closure is delayed, the more is the air pushed back to the intake manifold and consequently the higher is the load reduction.

As previously seen in the overview, the crank angle of closure delay is the only parameter which can be varied to implement the LIVC strategy; after some trials and errors, the optimal angle of closure delay resulted to be equal to 190° c.a.

The maximum lift and the slopes of the ascending and descending parts of the curve are not varied, while the angle of complete valve closing is now equal to 308° c.a., which means an increase of the angular period in which the intake valve is open equal to 11.6%.

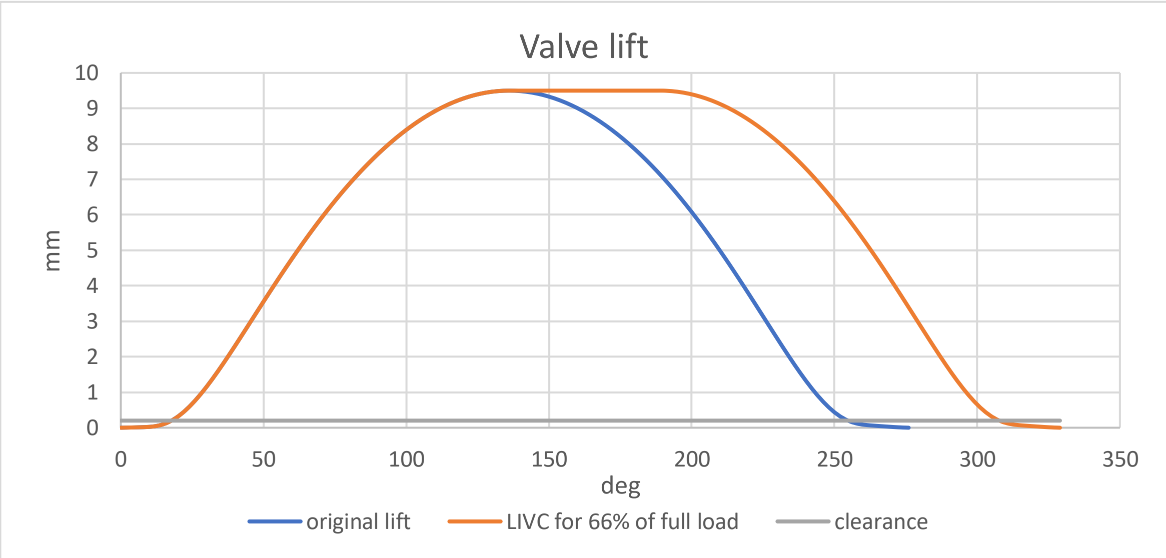

Figure 4.7 LIVC for 66%

The LIVC strategy explained above leads to the following engine performances:

Volumetric efficiency	75.702%
Brake torque	110.485 Nm
Brake power	53.222 kW
BSFC	82.229 g/MJ
Indicated efficiency	35.9%

It can be immediately seen that the improvement in the specific fuel consumption with respect to the case of the throttling valve is much lower than the one given by the EIVC strategy: the BSFC is reduced by 1% and the increase of the indicated efficiency is equal to 0.83%.

This is due to the pumping losses which are generated by the expulsion of the fresh charge to the intake manifold, which subtract useful work to the crankshaft and so decrease the efficiency.

The volumetric efficiency higher than the one provided by the EIVC strategy by 1.4%, the reason of this phaenomenon will be explained later.

4.2.4 Comparison between the control strategies

The control strategies previously shown are now compared: the graphs of volumetric efficiency, indicated efficiency and specific fuel consumption will be analysed in order to find the best control strategy for this speed and load.

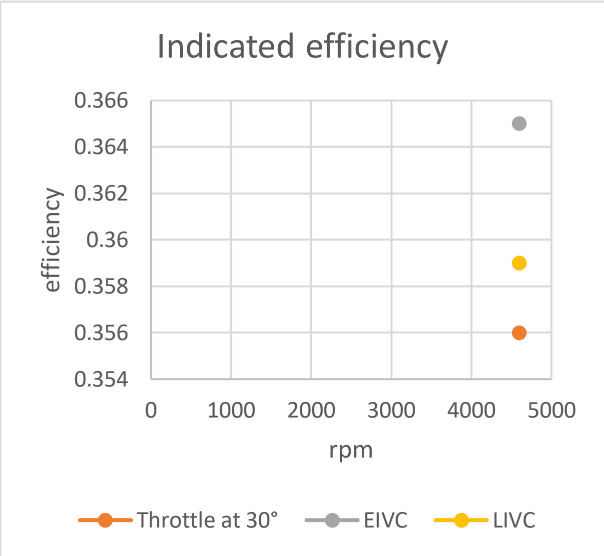

Figure 4.8 indicated efficiencies at 66%

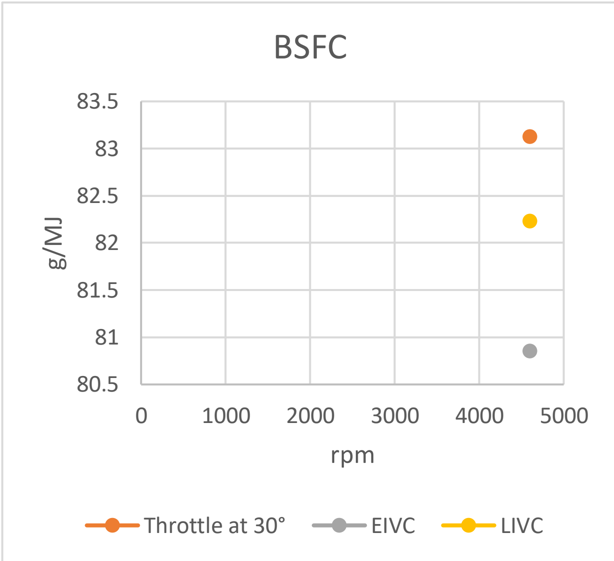

Figure 4.9 BSFC at 66%

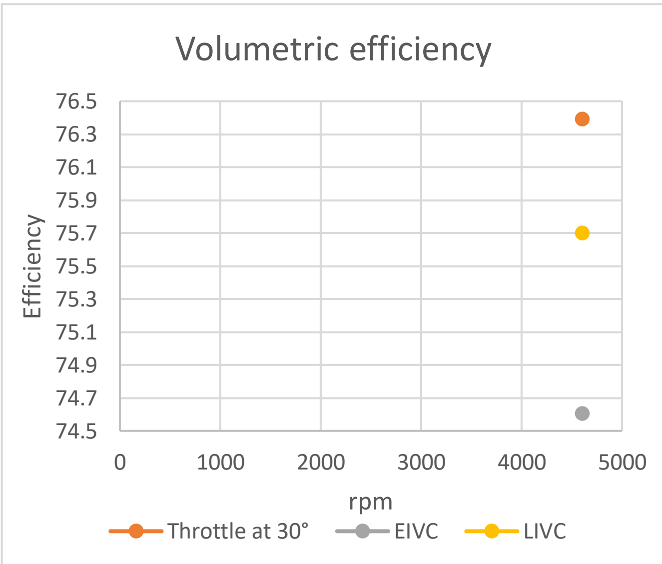

Figure 4.10 volumetric efficiency at 66%

The graphs of BSFC and indicated efficiency emphasise that the EIVC strategy gives the best results in fuel saving and efficiency improving for a low reduction of the load: this advantage is mainly due to the lower pumping losses with respect to the other control strategies, since the air doesn't pass through a restricted section and is not pushed back into the intake manifold.

The LIVC strategy is however better than the throttling valve since the Miller cycle has a higher thermal efficiency than the Otto one, overcoming the pumping losses; as it will be seen, a higher load reduction implies a higher delay for the intake valve close which leads to higher pumping losses, which will be no more counterbalanced by the higher thermal efficiency.

The pressure-volume diagram expressed in logarithmic scale gives important information about the gas exchange process, since the intensity of the pumping losses and the maximum pressure reached after the combustion of the air-fuel mixture are graphically shown.

Recalling what have been explained in the introduction, the area of the P-V diagram represents the work done by the system in the whole cycle, which is positive is the curve is swept in the clockwise direction and is negative if the curve is swept in the anti-clockwise one.

It can be easily seen that the positive part of the curves given by the three control strategies are translated downward with respect to the one provided by the full load case, since a lower amount of air is drawn into the cylinder and so a lower pressure is reached in the combustion chamber at the end of the compression stroke.

The most interesting part of the diagram is the anti-clockwise one, which shows the pumping losses: the throttling valve causes the highest pumping losses, because of the pressure drop due to the passage of the air in a restricted section; the EIVC and LIVC control strategies generate a remarkable improvement in pumping losses reduction, the lower part of their P-V diagram is much closer to the one of the full load case.

The curve path from point A to point B of the anti-clockwise portion of the curve given by the EIVC strategy (from $Log(V) = 1.75$ to $Log(V) = 2.53$) is more negative than the one resulted from the LIVC strategy, this means that the pumping losses generated in the first part of the intake stroke (the effective one) are higher: the valve lift implied in the EIVC strategy is lower than the one used in the LIVC strategy, so the air passes through a more restricted section which causes higher pressure drops.

This effect is however counterbalanced by the pumping losses due to the expulsion of the air back to the intake manifold in the case of LIVC strategy: their presence is pointed out in the right part of the diagram, from point B to point D, where there is a larger area with respect to the case of EIVC control strategy.

It must be also noticed that the clockwise area of the diagram resulting from the LIVC strategy is lower than the ones given by the other strategies, and the difference consists in the empty area between points C,D and E, which is due to the rejection of the air excess to the intake manifold: thanks to the almost total elimination of the pressure losses given by the absence of the throttling valve and the use of the maximum valve lift for the air passage, the point D, which represents the start of the compression stroke, is at the same pressure of the respective point of the full load case and consequently is not on the same line which contains the points C and E; in order to reduce the load, a determinate quantity of air is pushed back to the intake manifold, so the volume of the combustion chamber is lowered at almost constant pressure, and this event is represented by the line D-E, which forms together with the segments C-D and C-E an area which is subtracted to the clockwise part of the diagram, which otherwise would be coincident with the one resulting from the load reduction obtained by means of the throttling valve.

All the phaenomena explained above become more and more evident in the P-V diagram at the decreasing of the load.

The decreasing of the positive area lowers the useful work and consequently the indicated efficiency, and this effect increases at the decreasing of the load, since a higher and higher part of the compression stroke is devoted to the expulsion of the air excess; over a certain threshold, the decrease of the clockwise part of the P-V diagram becomes dominant with respect to the decrease of the anti-clockwise part with respect to the throttling valve control, so the LIVC strategy becomes less efficient than the throttling valve

The quantitative comparison of the pumping losses will be done in the last paragraph.

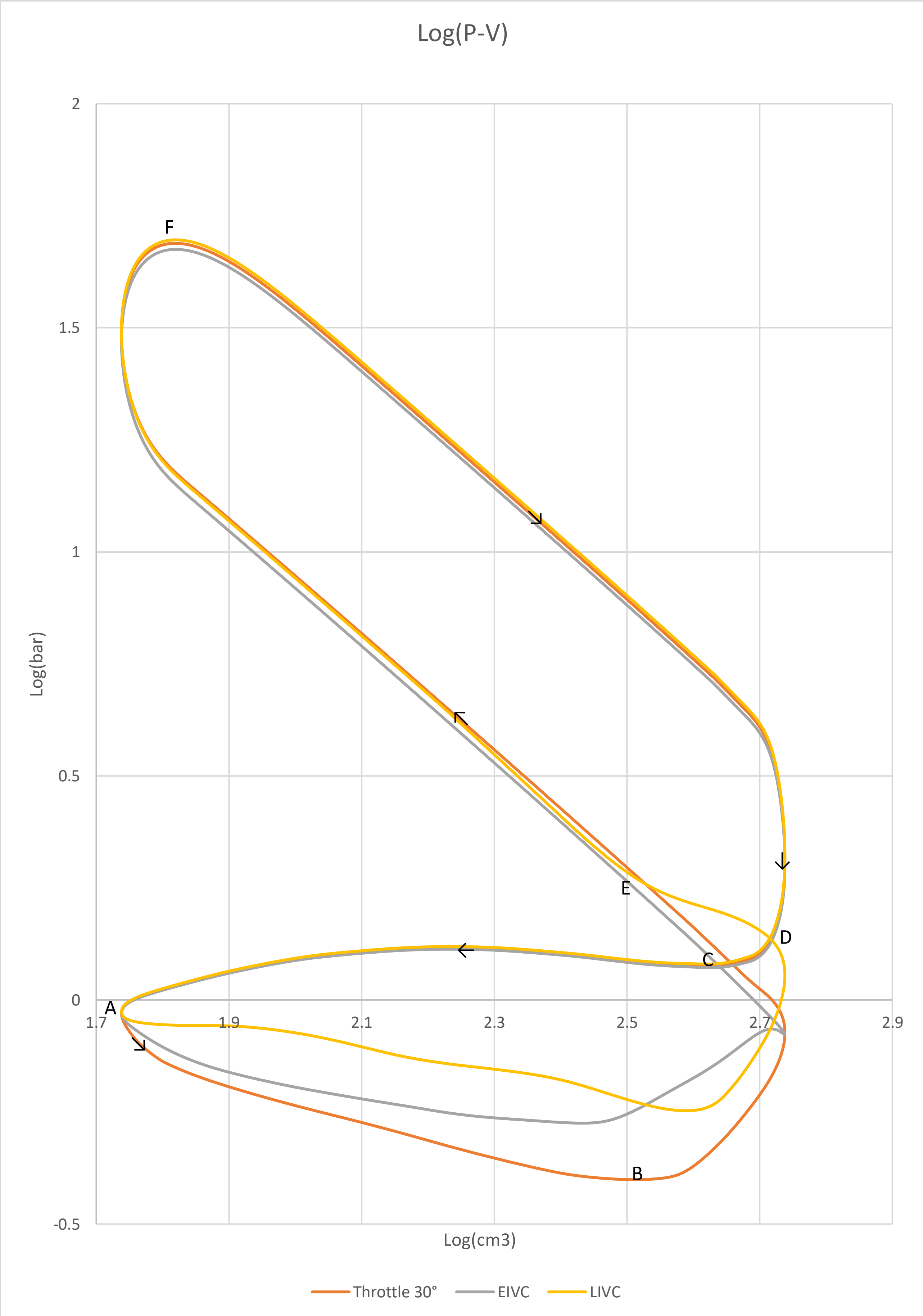

Figure 4.11 comparison between the P-V diagrams at 66% of full load

4.3 Load reduced to 27% of the full one

The load reduction under analysis is now the medium one, that is the 27% of the full load.

Even in this case the three load control strategies will be analysed separately, the mode of realization of each strategy will be explained and the main engine's performance will be shown.

Since the load reduction is heavier, the variation of the original valve lift will be more severe both in the case of EIVC strategy and in the one of LIVC strategy, and the effects on the P-V diagram induced by the modifications will be amplified.

In the end, the graphical comparison between the strategies will be made, with a discussion about the P-V diagram.

4.3.1 Load reduction by means of the throttling valve

The engine load is reduced to 27% of the full load if the throttling valve is inclined of an angle equal to 21° with respect to the vertical position, which is in the range of low opening angle, where a single degree of difference in the angular position of the throttling valve causes non-negligible variations of performance: a 30% of reduction of the opening angle with respect to the previous case implies a 59% of load reduction.

This is due to the higher pumping losses caused by the throttling valve, which forces the air to pass in a more restricted area causing a concentrated pressure loss.

The used valve lift obviously remains the original one, no other engine parameters are modified.

The load reduction to 27% of the full load obtained with the throttling valve gives the following performances:

Volumetric efficiency	44.74%
Brake torque	44.748 Nm
Brake power	21.555 kW
BSFC	119.994 g/MJ
Indicated efficiency	32.3%

The specific fuel consumption increases by 44.35% with respect to the case of 66% of the full load and by 61.22% with respect to the full load one.

The indicated efficiency has a 9.2 % of reduction with respect to the previous case and a 11.5% of reduction with respect to the full load one.

The volumetric efficiency decreases by 41.4% with respect to the previous case and by 57% with respect to the full load one.

The high decrease of efficiency with respect to the case of load reduction to 66% of the maximum one shows how severe the pumping losses are when the engine undergoes to a medium or high load reduction made by means of a throttling valve.

4.3.2 Load reduction by means of EIVC strategy

The implementation of the EIVC strategy to obtain a medium reduction of the engine load simply consists in the earlier cutting of the intake valve lift, so that a lower amount of air is drawn into the cylinder.

The intake valve lift which allows to reduce the engine load by 27% of the maximum has been derived by trials and errors: the new angle of effective valve closure resulted to be equal to 154° crank angle (132° considering the crevice), which means a reduction of 44.2% of the angular period in which the valve is open with respect to the original valve lift, while the maximum lift is equal to 6.22 mm, so it has been reduced by 34.5% with respect to the original profile; the peak is now placed at 75° camshaft angle and is sharper than the one of the valve lift law which allows the engine to run at 66% of its maximum load, since the cutting of the original curve after 75° c.a. with the same shape for the ascending and the descending part guarantees the same torque and power than the original engine with the throttling valve positioned at 21° from the vertical position.

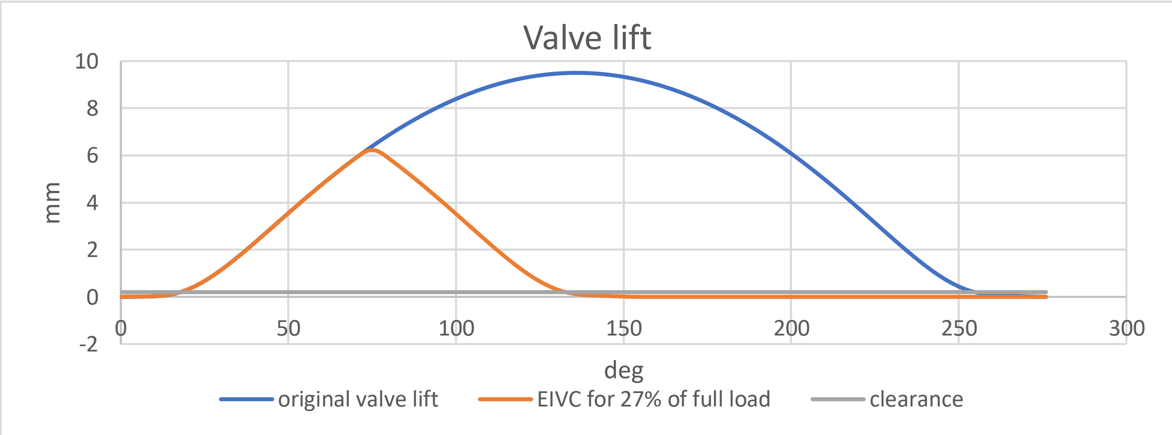

Figure 4.12 EIVC for 27%

It must be clarified that the systems which control the valve lift and the valve timing allow to obtain whichever shape of the valve lift curve, as a multiple lift (used in Fiat Multi Air at very low loads) or a vertical descending part; in this work, the modified valve lifts conserve as much as possible the shape of the original curve, in order to be implemented by means of simpler control system, like a mechanic system which cannot vary the valve lift with a sufficient velocity to guarantee a multiple lift during the intake stroke.

The EIVC strategy previously explained gives the following engine performances:

Volumetric efficiency	41.369%
Brake torque	44.767 Nm
Brake power	21.565 kW
BSFC	110.906 g/MJ
Indicated efficiency	35%

The EIVC strategy allows to reduce the specific fuel consumption by 7.5% and to increase the indicated efficiency by 8.34% with respect to the same load reduction obtained by means of the throttle plate, while the volumetric efficiency decreases by 7.53%.

The percentage of improvement between throttling valve and EIVC strategies are higher than the case of low load reduction, since the pumping losses avoided thanks to the second one increase with the load reduction. As for the low load reduction configuration, the EIVC strategy generates a lower volumetric efficiency with respect to the throttling valve, because of the lower amount of air needed to generate the same performance thanks to the lower pumping losses.

4.3.3 Load reduction by means of LIVC strategy

The LIVC strategy which makes the engine work at 27% of the maximum load simply consists in an increase of the intake valve closure delay, which allows the piston to push back to the intake manifold a higher amount of air in the first part of the compression stroke.

The maximum lift remains equal to 9.5 mm and is kept constant for 83° crank angle, so the intake valve starts to close at 219° and the angle of its effective closure is equal to 359° (the valve actually closes at 338° c.a. because of the crevices), so the angular period in which the valve lift constantly remains at its maximum value increases by 53.7% and the one during which the intake valve is open increases by 16.6% with respect to the LIVC strategy needed to run the engine at 66% of the maximum load.

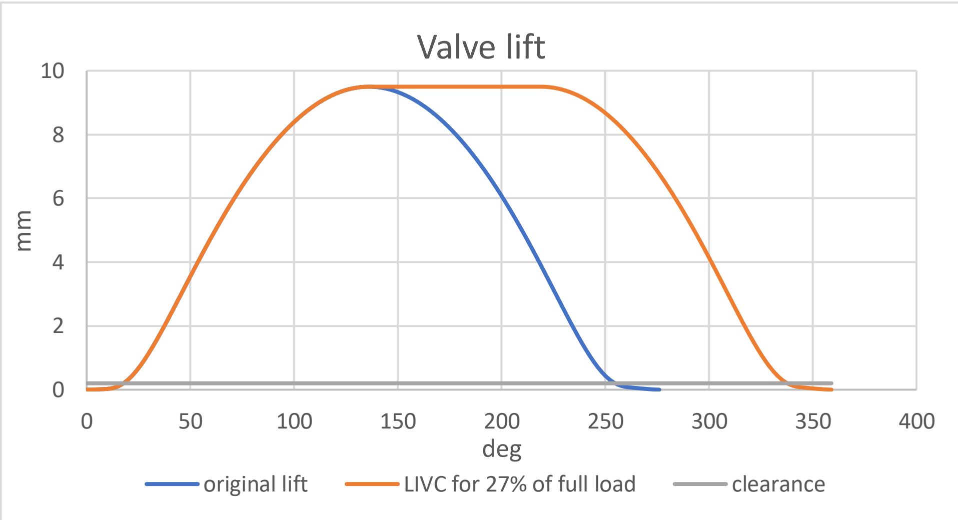

Figure 4.13 LIVC for 27%

The LIVC strategy previously described gives the engine the following performances:

Volumetric efficiency	44.77%
Brake torque	45.03 Nm
Brake power	21.693 kW
BSFC	119.311 g/MJ
Indicated efficiency	32.4%

The efficiency improvement given by the LIVC strategy at this level of load reduction is almost negligible: the specific fuel consumption decreases only by 0.57% and the indicated efficiency increases only by 0.3% with respect to the same load reduction obtained by the throttling valve.

This is due to the pumping work that the piston must spend to push the air out from the cylinder, which increases at the increasing of load reduction, since the more the load is reduced by means of the LIVC strategy, the more is the excess of air that must be pushed back to the intake manifold by the piston.

It must be kept in mind that when the intake stroke ends, the air inside the cylinder is at ambient pressure like the one inside the intake manifold, so the process of expulsion of the excess of air is not helped by a pressure difference as for the blowdown of the exhaust gases at the end of the expansion stroke.

The volumetric efficiency is substantially equal to the one given by the throttling valve with an opening angle of 21°, since the reduction of the pumping losses is counterbalanced by the decreasing of useful work due to the expulsion of the air excess.

4.3.4 Comparison between the control strategies

As for the low load reduction case, the control strategies will be compared looking to the graphs of volumetric efficiency, indicated efficiency and specific fuel consumption and also to the P-V diagram, in order to understand which is the most addicted for this level of load reduction.

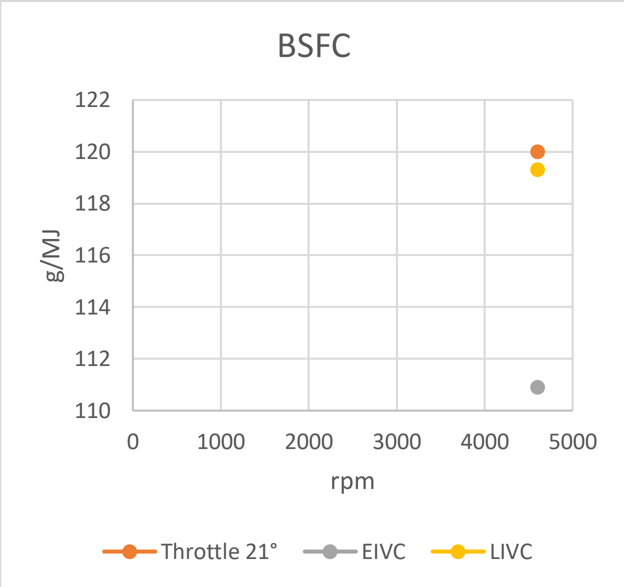

Figure 4.14 BSFC at 27%

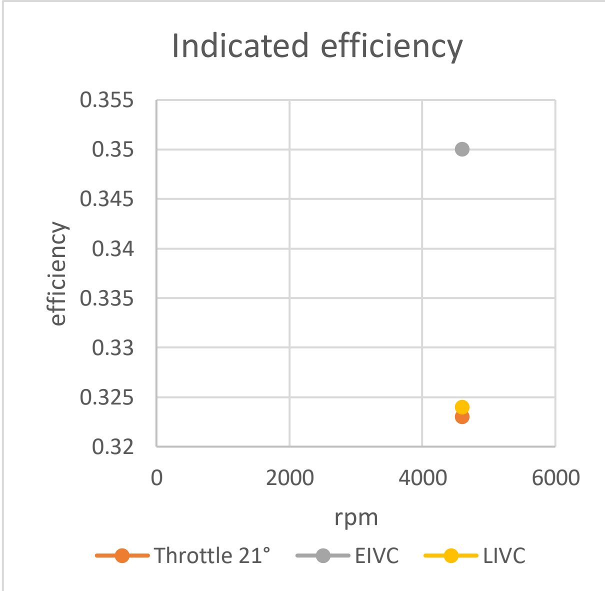

Figure 4.15 Indicated efficiency at 27%

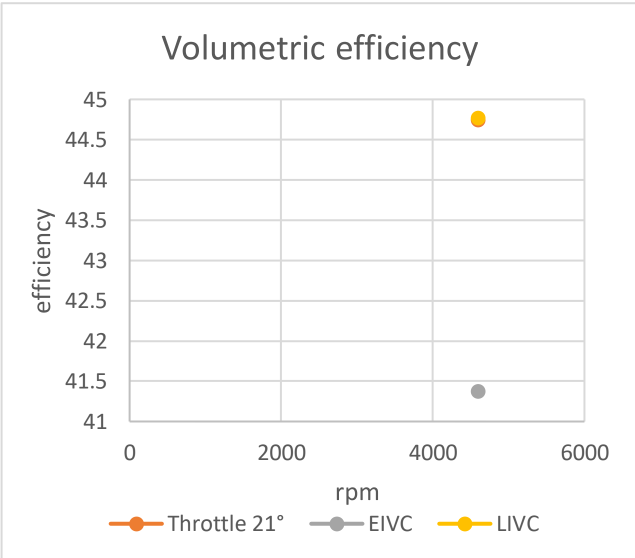

Figure 4.16 Volumetric efficiency at 27%

The difference of volumetric efficiency between the EIVC strategy and the other ones is now clearly visible, given its lower air request to generate the same torque and power.

The graphs of BSFC and indicated efficiency show that the EIVC strategy is better than the LIVC one, and its advantge is higher with respect to the case of low load reduction, because of the higher pumping losses and the reduction of the positive work generated by the LIVC strategy, which are proportional to the load reduction.

The LIVC strategy is consequently useless in case of medium or high level of load reduction, in case of high load reduction it generates a higher fuel consumption than the throttling valve.

As for the case of low load reduction, the P-V diagram expressed in logarithmic scale shows the intensity of the pumping losses which subtract work to the crankshaft in the single cycle: the pumping losses caused by the throttling valve are strongly increased with respect to the previous case, the maximum negative peak passes from 0.38 to 0.62 bar, which means a 63.2% of increase.

The pressure-volume curves of the three strategies of load control are obviously translated downward with respect to the one given by the full load case, since a lower amount of air is drawn into the cylinder.

Looking at the positive peaks of pressure reached after the combustion (point F) it can be noticed that the one resulting from the EIVC strategy is slightly lower than the ones given by the other control strategies, which are coincident, because of the lower volumetric efficiency which characterizes the EIVC strategy.

The P-V diagram obtained with the LIVC strategy has an almost horizontal line in its clockwise part, from point D to point E, which represents a decrease of volume at constant pressure, that is the rejection of the air excess to the intake manifold, which causes a loss of useful work with respect to the throttling valve control and the EIVC strategy equal respectively to the areas defined by the points D,C and E and D,G,H.

Considering the lowest part of the anti-clockwise area of the P-V diagram, the EIVC strategy causes higher pumping losses than the LIVC one during the initial part of the intake stroke, from point A to point B, this is due to the lower lift of the intake valve which forces the air to pass in a restricted section and consequently to undergo to a pressure drop; the curve defined by point A and point Bis shorter with respect to the case of low load reduction, since the intake valve is closed earlier.

The EIVC strategy resulted to have the lowest pumping losses, as it has been pointed out by the data about specific fuel consumption and indicated efficiency.

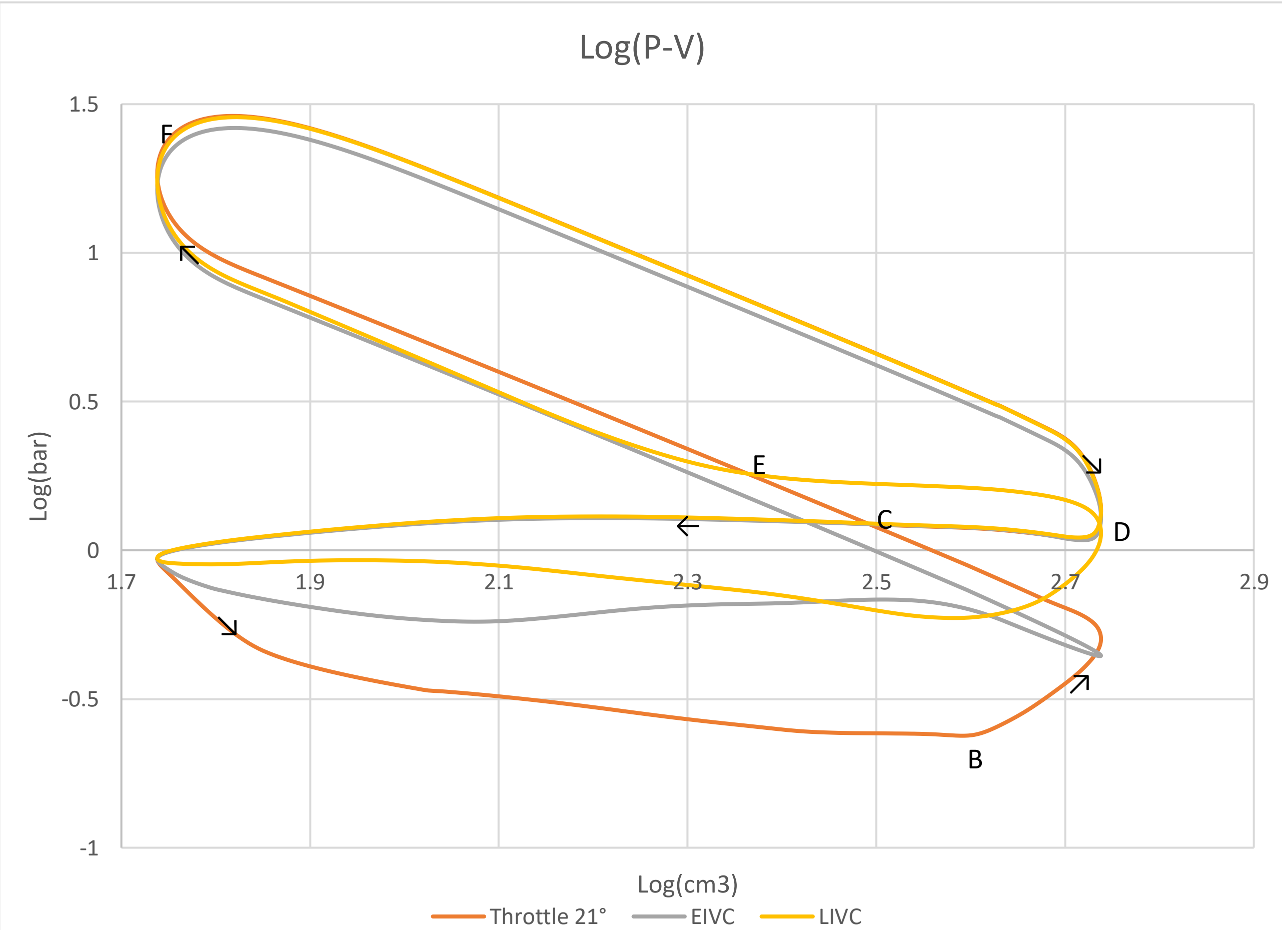

Figure 4.17 comparison between the P-V diagrams at 27% of full load

4.4 Load reduced to the 11% of the full one

The last configuration under analysis is characterized by an engine load equal to 11% of the maximum load, which is a high level of load reduction.

The amount of air entering in the combustion chamber is strongly decreased, and its reduction generates a huge pumping work if obtained by means of a throttling valve: all the effects of the pumping losses which have been seen in the previous load cases will be amplified, this is the load configuration in which the EIVC strategy shows its maximum utility, as it will be seen later.

The procedure of analysis will be the same as before: the three control strategies will be explained separately and then they will be compared.

4.4.1 Load reduction by means of the throttling valve

The load reduction which makes the engine work at 11% of its maximum performance requires an opening angle of the throttle plate equal to 17.4° with respect to the vertical position, which means that the throttling valve is almost closed.

The angle has been derived after some trials and errors in which it has been discovered that a difference of one tenth of degree in the angular position of the plate causes a remarkable change in the torque and power output, increasing or decreasing them even by 5%:

- $\alpha = 17.4° \rightarrow T = 18.478\ \mathrm{Nm}$
- $\alpha = 17.3° \rightarrow T = 17.343\ \mathrm{Nm}$
- $\alpha = 17.5° \rightarrow T = 19.125\ \mathrm{Nm}$

Because of this phaenomenon, a reduction of 3.6° (17.14%) in the angular position of the plate causes a decrease of engine performance by 58%.

The load reduction obtained by the throttling valve makes the engine capable of these performances:

Volumetric efficiency	32.076%
Brake torque	18.478 Nm
Brake power	8.901 kW
BSFC	208.331 g/MJ
Indicated efficiency	29.2%

The specific fuel consumption increases by 73.6% with respect to the medium load reduction and by 180% with respect to the full load configuration, while the indicated efficiency decreases respectively by 9.6% and by 20%: this confirms that the throttling valve causes an unacceptable loss of efficiency when its opening angle is low, and a variable valve acting system is the only way to reduce the fuel consumption at partial load.

The volumetric efficiency decreases by 28.3% with respect to the previous case and by 69% with respect to the full load one.

4.4.2 Load reduction by means of EIVC strategy

The reduction by 89% of the full load reached by the EIVC control strategy requires a very short intake valve lift and a strongly restricted opening period.

Even this EIVC strategy has been derived by trials and errors, the maximum lift and the angle of effective closure of the intake valve resulted to be respectively equal to 5.13 mm and 134° c.a. (113° if the clearance is considered), while the peak is reached after 65°: the maximum lift has been decreased by 46% and the angular period of effective valve opening has been reduced by 51.45% with respect to the original valve lift profile.

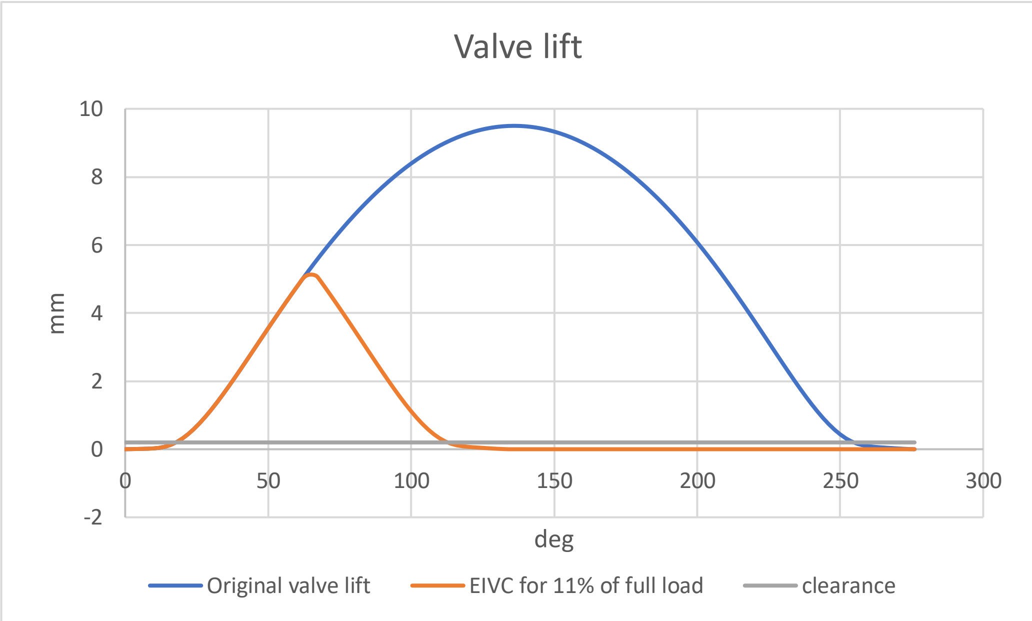

Figure 4.18 EIVC for 11%

The valve lift profile shown before gives the engine the following performances:

Volumetric efficiency	28.646%
Brake torque	18.142 Nm
Brake power	8.739 kW
BSFC	189.507 g/MJ
Indicated efficiency	32.5%

The specific fuel consumption decreases by 9% with respect to the same load reduction given by the throttling valve, while the indicated efficiency increases by 11.3%: this confirms again that the improvement of the engine's efficiency given by the EIVC strategy increases with the severity of the load reduction, since it allows to avoid higher and higher pumping losses.

The volumetric efficiency decreases by 10.7% with respect to the one resulting from the throttle plate with an opening angle of 17.4°, since the air mass which must be drawn into the cylinder to obtain the same torque and power provided by the throttling valve is strongly reduced.

4.4.3 Load reduction by means of LIVC strategy

The load reduction by 89% can be obtained also by means of the LIVC strategy, but the intake valve closure delay must be strongly increased, since almost all the air drawn into the cylinder during the intake stroke must be rejected back to the intake manifold during the compression stroke.

This causes a high increase of the pumping losses, which make this control strategy useless if the engine load is reduced over a certain threshold.

The valve lift profile which reduces the engine load by 89% implementing a LIVC strategy has been found by trials and errors, the angle of complete valve closure resulted to be equal to 370° crank angle (349° considering the clearance), which implies an increase of the valve's opening period by 34%, while the maximum lift remains equal to the original one and is kept constant for 94°, making the valve closure starts at 231° c.a. and so increasing the angular period in which the valve is at its maximum lift by 13.25% with respect to the case of medium load reduction.

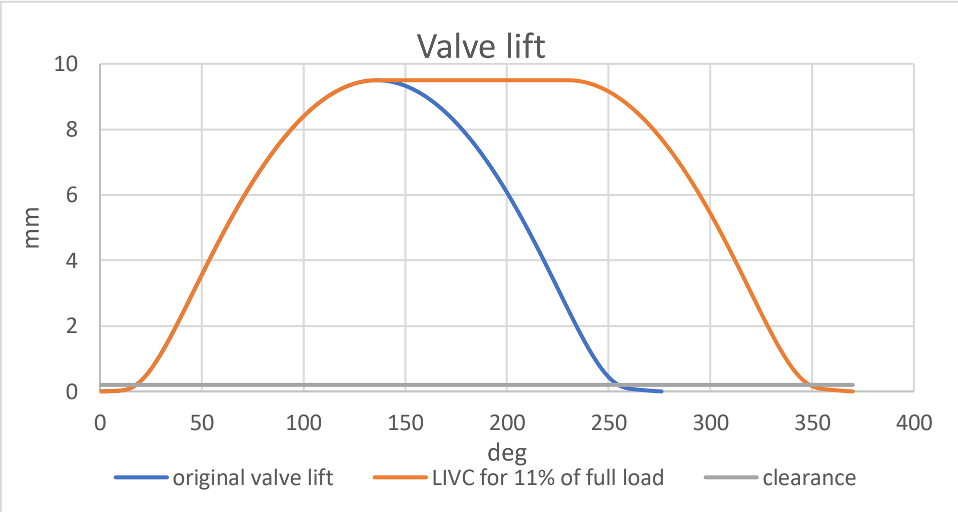

Figure 4.19 LIVC for 11%

The LIVC strategy explained before ensures the following performances to the engine:

Volumetric efficiency	32.827%
Brake torque	18.271 Nm
Brake power	8.801 kW
BSFC	215.647 g/MJ
Indicated efficiency	28.4%

These results prove definitively that the LIVC strategy worsens the specific fuel consumption if applied at low engine load: the BSFC is increased by 3.5% and the indicated efficiency is decreased by 2.74%, this means that the pumping losses and the reduction of the positive work generated by the expulsion of the air excess to the intake duct are higher than the ones due to the throttling valve which are avoided thanks to the LIVC strategy itself.

The volumetric efficiency is slightly higher than the one provided by the throttling valve, with an increase of 2.34%; the better filling of the cylinder is allowed by the lower resistance that the air flux finds in the intake manifold thanks to the absence of the throttling valve, and a higher mass of air is needed to obtain the same torque and power given by the other strategy because of the reduction of the indicated efficiency explained before.

4.4.4 Comparison between the control strategies

The graphs of volumetric efficiency, brake specific fuel consumption and indicated efficiency will be analysed to compare the different control strategies from a graphical point of view and confirm definitively that the EIVC strategy is the best one for what concerns the load reduction in a naturally aspirated engine.

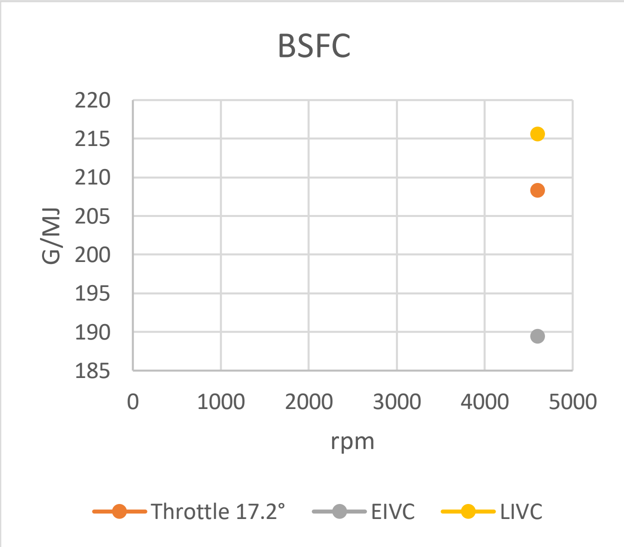

Figure 4.20 BSFC at 11%

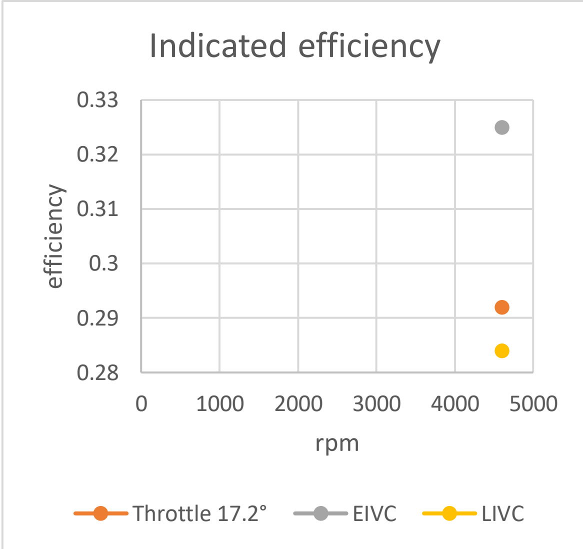

Figure 4.21 Indicated efficiency at 11%

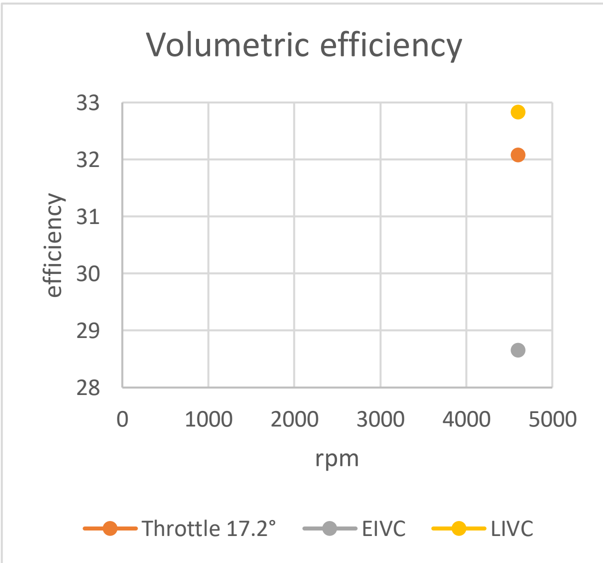

Figure 4.22 Volumetric efficiency at 11%

The three figures show that the advantage of the EIVC strategy is increased; the figure about BSFC clearly shows that the LIVC strategy is characterized by a higher specific fuel consumption than the throttling valve.

The graph of indicated efficiency shows that the EIVC strategy allows to maintain a high indicated efficiency even in case of strong load reductions: after a load reduction by 89%, the indicated efficiency decreased by 11.7%.; the LIVC strategy has the lowest indicated efficiency, remembering that this quantity is inversely proportional to BSFC.

The volumetric efficiency given by the EIVC strategy is sligthly lower than the ones resulted from the other.

The influence of the pumping losses on the engine efficiency are clarified by the logarythmic pressure-volume diagram, as it has been done in the previous load configurations.

It can be noticed that the anti-clockwise area of the curve resulting from the throttling valve control is highly expanded with respect to the medium load case: the little opening angle of the plate causes a hard pressure drop which increases the pumping work that the piston must spend to draw the fresh air into the cylinder, lowering the engine's efficiency.

The clockwise part of the diagrams generated by the three control strategies are further reduced because of the decreasing of the air mass entrapped in the combustion chamber.

Considering the lower part of the diagram,which refers to the intake stroke, the curve defined by the points A and B which shows that the EIVC strategy has higher pumping losses with respect to the full load case is further reduced, since the intake valve closes earlier than the previous case.

The horizontal segment D-E in the clockwise part of the diagram resulting from the LIVC strategy is longer than before, since the piston travels a higher distance before the intake valve closing, so there is a high reduction of volume at constant pressure; as for the other load configurations previously analysed, that segment forms with the lines H-D and D-G an area which is subtracted to the positive part of the diagram causing a reduction of positive work which reduces the indicated efficiency with respect to the EIVC strategy.

The lowest anti-clockwise area resulted to be the one given by the EIVC strategy, which consequently guarantees the highest efficiency.

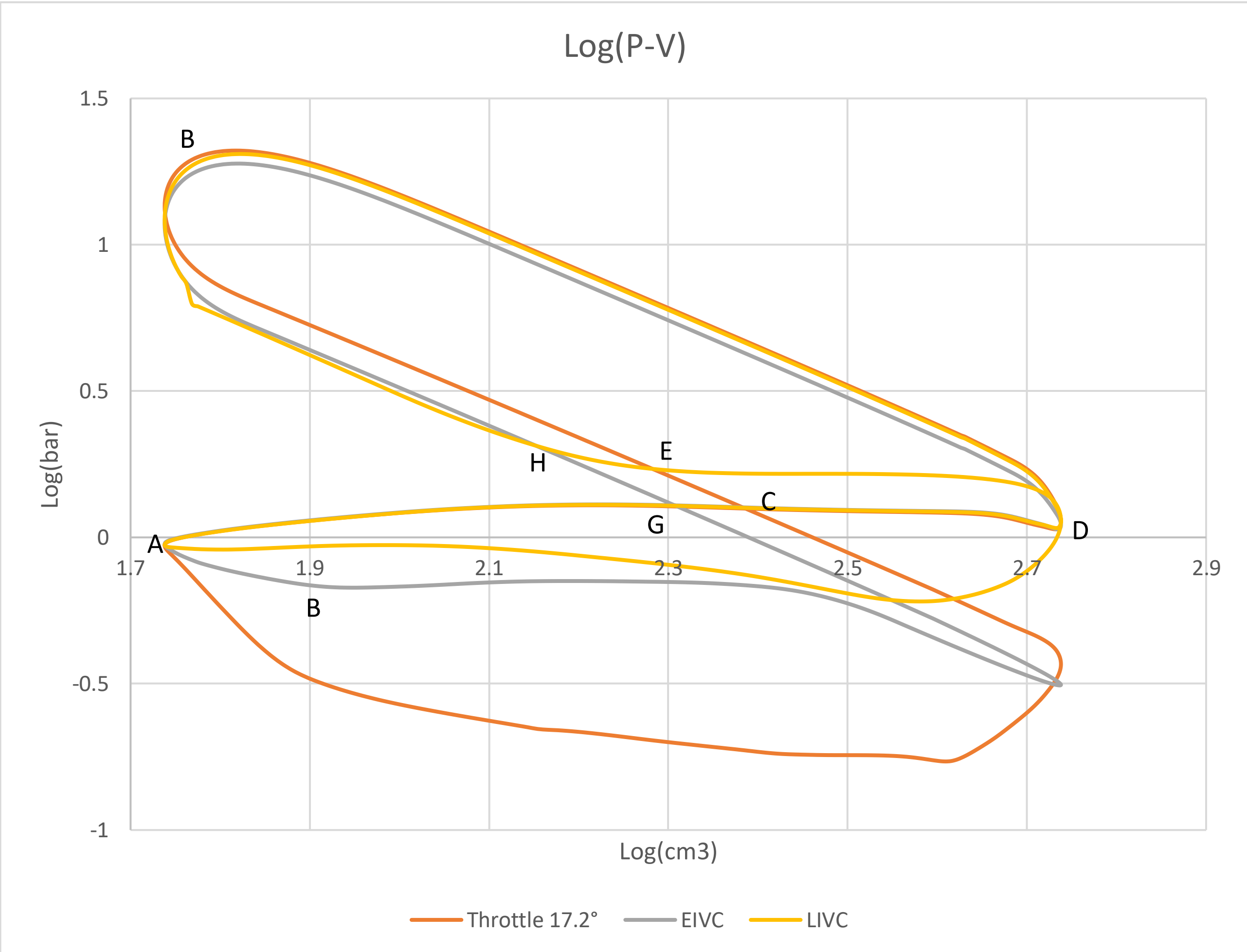

Figure 4.23 comparison between the P-V diagrams at 11% of full load

4.5 Global comparison of all the control strategies for all the load configurations

The study of the benefits in specific fuel consumption and engine's efficiency consisted so far in the analysis of three levels of load reduction, by 33%, 73% and 89%; all these load configurations have been reached by means of the throttling valve, the early inlet valve closure and the late one.

Each control strategy has been separately explained and the main performance it ensures have been shown.

At the end of the study of each single load configuration, the results given by each control strategy at that engine load have been graphically compared, to have a first indication about the best control strategy; all these comparisons indicated that the EIVC strategy is the best one, in particular at low loads.

All the analysis previously performed will be now synthetized, in order to completely understand and clarify the advantages of the EIVC control strategy with respect to the other ones.

Fisrtly, all the testing points previously analysed will be tabulated, to show the increase or decrease of valve lift and opening period, then the column diagrams of volumetric efficiency, indicated efficiency and brake specific fuel consumption obtained in the three load configuration will be compared.

All the testing points previously analysed are listed in the table below, in which the load is expressed by means of the brake mean effective pressure: the engine at full load provides a BMEP equal to 10.683 bar, so
$10.683 * 0.66 = 7\ bar$ $\quad 10.83 * 0.27 = 2.86\ bar$ $\quad 10.683 * 0.11 = 1.16\ bar$

	Throttled 7 bar	EIVC 7 bar	LIVC 7 bar	Throttled 2.86 bar	EIVC 2.86 bar	LIVC 2.86 bar	Throttled 1.16 bar	EIVC 1.16 bar	LIVC 1.16 bar
Throttle opening	33.3%	100%	100%	23.33%	100%	100%	19.33%	100%	100%
IVO	359°	359°	359°	359°	359°	359°	359°	359°	359°
IVC	597°	522°	649°	597°	474°	679°	597°	454°	690°
Lift [mm]	9.5	7.482	9.5	9.5	6.22	9.5	9.5	5.13	9.5
Change %	0	-21.24	0	0	-34.52	0	0	-46	0
Duration	238°	163°	290°	238°	115°	320°	238°	95°	331°
Change%	0	-46	21.85	0	-75.46	34.45	0	-87.73	39

The table above shows how the reduction of the valve lift needed to implement the EIVC strategy becomes more and more severe at the decreasing of the load because of the decrease of the air mass which must be drawn into the cylinder to obtain the desired engine performance.

The progressive decrease of the opening period in case of load controlling by means of EIVC strategy and the progressive increase of it in case of load reduction by means of LIVC strategy is also clearly visible in the table: if a lower quantity of air has to be drawn into the cylinder, the intake valve must be closed earlier, if a higher air excess must be bushed back to the intake manifold, the intake valve must be closed later.

In the first raw of the table it can be noticed that a sligth reduction of the throttle plate's opening angle causes a remarkable reduction of the engine load, as it has been largely seen before.

The analysis of the column diagrams of volumetric efficiency, indicated efficiency and BSFC allows to have an immediate view of the order of magnitude of the improvements on these quantities given by the load control strategies and to make a synthesis of the phaenomena which lead to that improvements.

The column diagram of the volumetric efficiencies shows that, independently from the the used control strategy, the volumetric efficiency decreases at the increasing of the load: this is an expected phaenomenon, since the volumetric efficiency is defined as the ratio between the air mass effectively drawn into the cylinder and the theoretical one, calculated as the product between the engine displacementand the air density:

$$\eta_{vol} = \frac{m_a}{V\rho_a}$$

Since the air mass m_a must be decreased to reduce the load in a S.I engine (while in a Compression Ignition engine the load is reduced by the decreasing of the injected fuel quantity) the volumetric efficiency is lowered when the load is reduced.

Considering now the single control strategies, it can be seen that the EIVC one has the lowest volumetric efficiency whatever the load reduction: since this strategy generates lower pumping losses with respect to the others, the same torque and power output can be reached burning a lower mass of air-fuel mixture, so a lower amount of air is drawn into the cylinder and consequently the volumetric efficiency decreases.

The decrease of volumetric efficiency with respect to the throttling valve control is by 2.74% when the BMEP is equal to 7 bar, by 7.53% when it is equal to 2.86 bar and by 10.7% when it is equal to 1.16 bar.

The volumetric efficiency given by the LIVC strategy is slightly lower than the one provided by the throttling valve at high load, the two methods of load control ensure the same volumetric efficiency when the BMEP is equal to 2.86 bar and the first one gives a bit higher efficiency than the second one at very low loads.

It has been already shown that the indicated efficiency resulting from the LIVC strategy becomes lower than the one provided by the throttling valve at very low loads, so, to counterbalance this phaenomenon and get the same torque and power output, a higher amount of air-fuel mixture must be introduced into the cylinder, so the same load reduction is obatined with a higher volumetric efficiency.

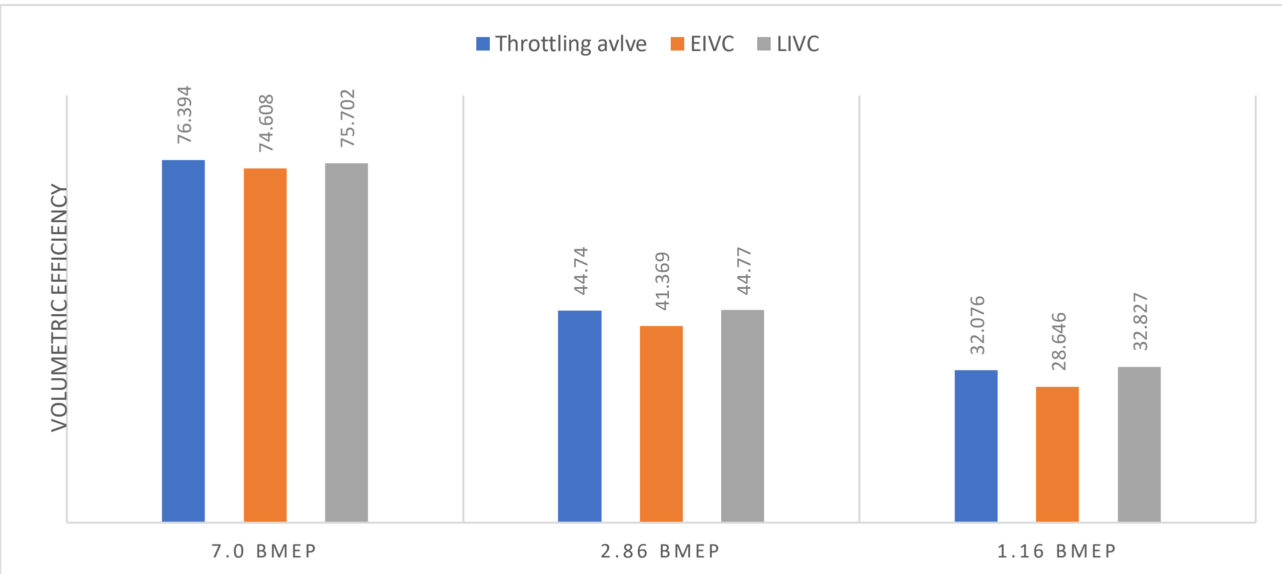

Figure 4.24 column diagram of volumetric efficiency

The column diagram of the indicated efficiency shows that, independently from the load reduction method, this quantity doesn't decrease with the reduction of the engine load as much as the volumetric efficiency: considering the throttling valve control strategy and recalling that the indicated efficiency of the engine at full load is equal to 0.368, it can be seen that the maximum decrease of indicated efficiency with respect to the full load configuration at a very low load is by the 20.65%, while the reduction of the volumetric efficiency considering the same case is by 69%.

Considering now the single load reduction strategies, it can be seen that the EIVC one provides the best indicated efficiency in each load condition, and its advantage becomes more and more important at the decreasing of the engine load: when the BMEP is equal to 7 bar the indicated efficiency provided by the EIVC strategy is better than the one given by the LIVC strategy by 1.67%, when it is equal to 2.86 bar the percentage of improvement rises up to 8% and when it is equal to 1.16 bar the EIVC control strategy gives an indicated efficiency better than the one provided by the throttling valve by 10.15%.

The reason of the superiority of the EIVC strategy on the other control methods consists in the higher indicated work that this strategy can guarantee: in each analysed configuration and in particular at low loads the pumping losses generated by the early intake valve closing were lower than the ones caused by the throttling valve, and the positive part of the P-V diagram was higher than the one resulted from the LIVC strategy, since there was no reduction of positive work caused by the expulsion of the air excess back to the intake manifold (it must be kept in mind that this process generates pumping losses which increase the anti-clockwise part of the P-V diagram and consequently causes a further decrease of the net useful work available at the engine crankshaft).

The higher indicated work leads to a higher indicated efficiency, since that quantity is given by the ratio between the indicated power and the fuel power, and the first is given by the product between the indicated work and the engine speed:

$$\eta_i = \frac{P_i}{\dot{m}_c H_i}$$

$$P_i = \frac{L_i \eta}{\varepsilon}$$

This graph, together with the one of BSFFC, proves definitively that the EIVC strategy is the best option to reduce the engine load in a naturally aspirated engine

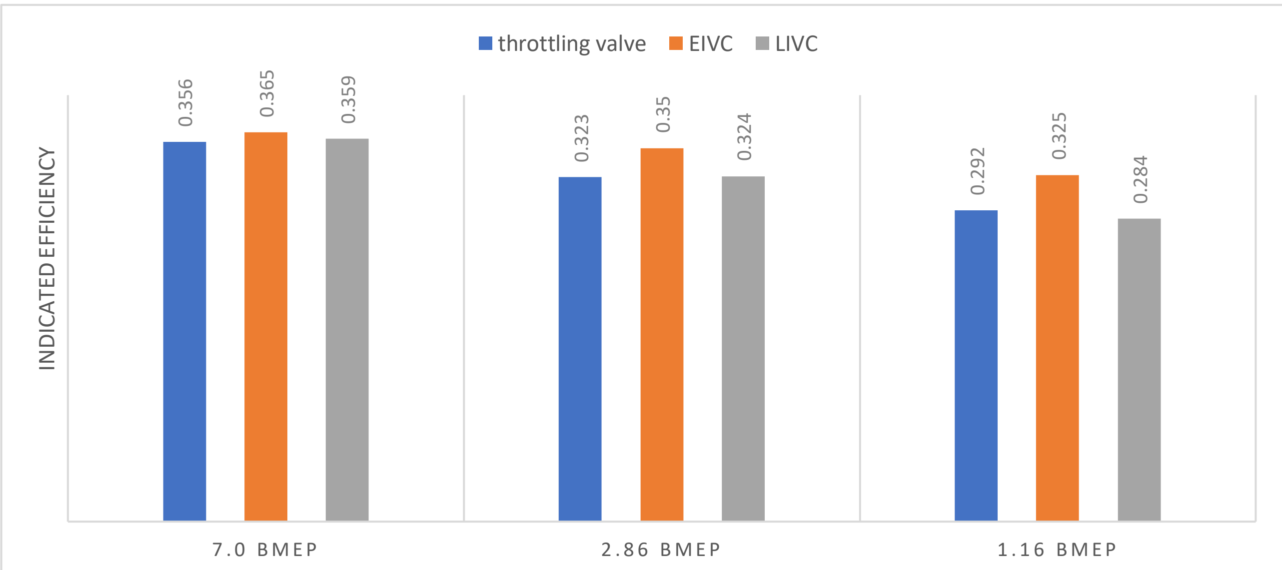

Figure 4.25 column diagram of Indicated efficiency

The column diagram about the brake specific fuel consumption actually provides the same information given by the one about the indicated efficiency: the EIVC strategy is the best one whatever is the load reduction, the more the BMEP drops down, the more the utility of this control strategy rises up.
The EIVC strategy improves the specific fuel consumption with respect to the load reduction based on the throttling valve by 2.73% in the high load configuration and by 7.57% in the medium load one, while at very low load it improves the BSFC by 12.12% with respect to the LIVC control strategy.

It can be noticed that the percentages of improvement of the specific fuel consumption are close to the ones of improvement of indicated efficiency, in particular in case of mid load reduction: as explained before, these two quantities are inversely proportional to each other but they are linked by a multiplicative factor given by the product between the lower heating value of the fuel and the engine's organic efficiency:

$$\eta_i = \frac{P_i}{\dot{m}_c H_i}$$

$$BSFC = \frac{\dot{m}_c}{\eta_o P_i} = \frac{1}{\eta_i \eta_o H_i}$$

Differently from the fuel's lower heating value, the organic efficiency varies with the speed and the load of the engine, in particular a reduction of the organic efficiency is caused by an increase in the engine speed (because of the higher inertial forces) or a decrease in the engine load, which makes the energy spent to drive the engine auxiliaries become proportionally more important.
This is the reason why the percentages of improvement of indicated efficiency are not coincident with the ones of enhancement of brake specific fuel consumption and a reduction of the BMEP by 59.44% (from 2.86 to 1.16 bar) causes an increase of specific fuel consumption by 73.62% considering the load control implemented by the throttling valve.

Considering that the specific fuel consumption of the engine at full load is equal to 74.424 g/MJ, in case of use of EIVC strategy to obtain a reduction by 89% of the load the BSFC increases by 154.63%, while it would be increased by 180% in case of throttled engine: this data underline the most critical point of the spark ignition engine, which needs a reduction of the air mass and the injected fuel (keeping constant the air-fuel ratio) to reduce the load.

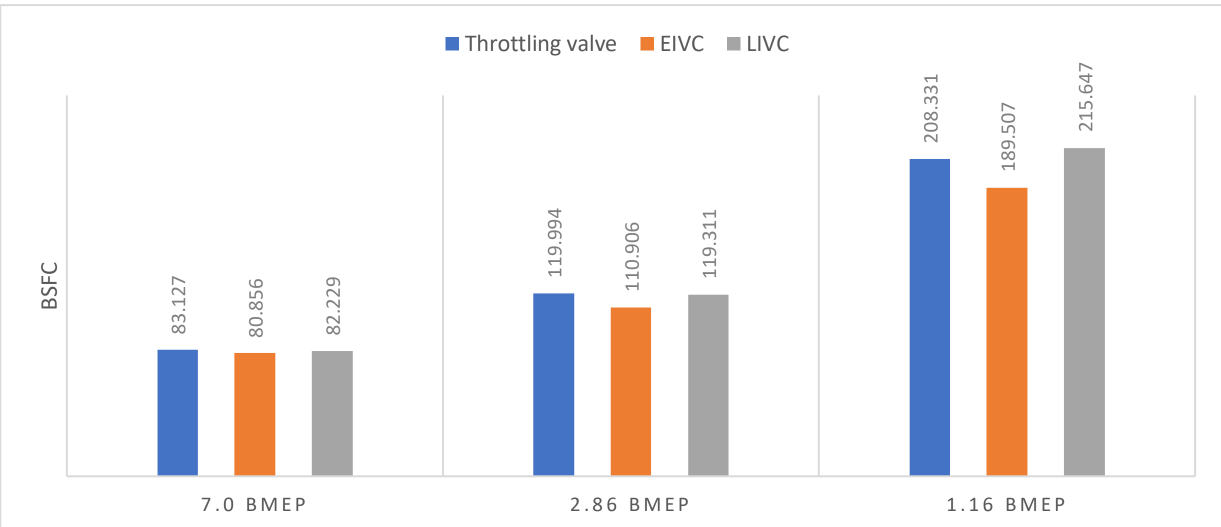

Figure 4.26 column diagram of BSFC

4.6 Open cycle efficiency analysis

The quantitative analysis of the efficiency of the gas exchange process, which is named open cycle efficiency, completes the comparison between the three ways of load reduction.
It can be done knowing the exact value of the integral of both the positive than the negative part of the P-V diagram, since is defined as the ratio between the gross indicated mean effective pressure (GIMEP), which evaluates the mean effective pressure in the cylinder during the compression end the expansion strokes, and the net one (NIMEP), which evaluates it on the whole cycle and is always called indicated mean effective pressure (IMEP) [2]:

$$\eta_{\text{open}} = 1 + \frac{\text{PMEP}}{\text{GIMEP}} = \frac{\text{NIMEP}}{\text{GIMEP}}$$

$$\text{NIMEP} = \text{GIMEP} + \text{PMEP}$$

Where PMEP is the pumping mean effective pressure, which measures the mean effective pressure during the intake and exhaust strokes.

Since Gasdyn calculates only the NIMEP, it has been necessary to write a Matlab code which calculates separately the integrals of the positive and negative parts of the P-V diagram.

The data about the open cycle efficiency of each load configuration are listed in the table below, whose last column contains the data about the reduction of pumping mean effective pressure guaranteed by EIVC and LIVC strategy with respect to the throttled engine in each single load configuration.

		NIMEP (bar)	GIMEP (bar)	PMEP (bar)	$\eta_{\text{open}}\%$	PMEP REDUCTION %
10.683 bar	**Full load**	12.59	13.1445	-0.5543	95.78	-
7 bar	**Throttled**	8.9460	9.6178	-0.6719	93	-
	EIVC	8.9804	9.4562	-0.4758	94.97	29.19
	LIVC	8.9647	9.4689	-0.5042	94.67	24.96
2.86 bar	**Throttled**	4.7708	5.4959	-0.7251	86.81	-
	EIVC	4.8024	5.1130	-0.3106	93.93	57.16
	LIVC	4.7823	5.2407	-0.4583	91.25	36.79
1.16 bar	**Throttled**	3.0965	3.7752	-0.6787	82	-
	EIVC	3.0659	3.2998	-0.2338	92.91	65.56
	LIVC	3.0766	3.5279	-0.4512	87.21	33.52

The progressive decrease of the open cycle efficiency with the reduction of the load is evident and is also clear that the EIVC strategy guarantees the highest efficiency whatever the load reduction, in particular when the brake mean effective pressure is very low: the maximum decrease of open cycle efficiency obtained in case of use of EIVC strategy is by 3%, while the throttled engine would lead to an efficiency drop by 14.38%.

The last column of the table confirms in a quantitative way that the pressure drops caused by the throttle plate become extremely severe at low loads, with a decrease of open cycle efficiency by 11.25% with respect of wide open throttle (WOT) configuration and that the early intake valve closing strategy gives the lowest pumping losses: when the BMEP is equal to 1.16 bar, it provides a reduction of the PMEP by 65.56%, which means that the improvement it gives is almost double than the one offered by the LIVC strategy.

4.7 Close cycle efficiency analysis

The data about the gross indicated mean effective pressure, which indicates the mean pressure in the cylinder during the compression and the expansion phases, allow to perform the closed cycle analysis and calculate the closed cycle efficiency, which measures the efficiency of the compression, combustion and expansion processes.
As for the case of open cycle efficiency, the mathematical definition is kept by [2], and states that the closed cycle efficiency is the ratio between the gross indicated power of the engine and the provided fuel power:

$$\eta_{closed} = \frac{GIMEP}{\dot{m}_c H_i} * \frac{V_d n}{\varepsilon} = \frac{grossIP}{fuel\,power}$$

The lower heating value of the fuel is equal to 43000 kJ/kg, the regime under analysis is 4600 rpm and the fuel mass flow rate can be derived from the brake specific fuel consumption and the brake power calculated by Gasdyn:

$$\dot{m}_c = BSFC * P_b$$

The table below collects the data about the gross indicated power, the fuel power and the closed cycle efficiency:

		Gross IP (kW)	**Fuel power (kW)**	η_{closed}%
10.683 bar	**Full load**	99.24	258	38.4
7 bar	**Throttled**	72.62	189.9	38.23
	EIVC	71.39	185.46	38.49
	LIVC	71.49	188.18	37.99
2.86 bar	**Throttled**	41.50	111.22	37.31
	EIVC	38.60	102.84	37.53
	LIVC	39.57	111.29	35.55
1.16 bar	**Throttled**	28.50	79.73	35.74
	EIVC	24.91	71.21	34.98
	LIVC	26.64	81.61	32.6

It can be noticed that the closed cycle efficiency is characterized by a low decrease at the reducing of the load: the gap between the best value given by the full load case and the worst one resulting from the LIVC strategy when the BMEP is equal to 1.16 bar is by 15.1%.
The fuel power provided by the EIVC strategy is the lowest in each load configuration, since the fuel mass flow into the cylinder is decreased by the early closure of the intake valve; when the load reduction is medium or high, the late intake valve closing guarantees the highest fuel flow rate, since a higher quantity of air is drawn into the cylinder thanks to the absence of pressure losses and so a higher fuel mass is necessary to keep the air to fuel ratio constant.

Differently from the case of the indicated efficiency, the EIVC strategy doesn't cause an improvement with respect to the throttled engine, while it causes a decrease of efficiency at low load: it has been demonstrated in [8] that this method of load control causes a reduction of the air velocity during the ignition, flame initiation and propagation phases, leading to a slower combustion and so to an instability of the flame propagation.

4.8 Summary

The objective of this chapter is to show the benefits and the drawbacks of the EIVC and LIVC strategies on the engine specific fuel consumption after their application on a 2000 c.c. 16 valves 4 cylinders naturally aspirated engine, and to derive the best one in the conditions of low, medium and high load reduction.

In the overview, the design method of the valve lift profiles needed for the implementation of the early and late intake valve closing has been pointed out, focusing on the fact that the first one can be designed varying the opening period and the valve lift, while the second one can be designed acting only on the valve closure delay, and that the optimal profile for each load configuration is reached by trials and errors.

In the first paragraph the full load engine configuration has been described, showing the original valve lift, the absolute crank angle of intake valve opening, the combustion parameters and the torque and power curves, which were necessary to find the maximum brake torque regime, in correspondence of which all the analysis shown in the chapter has been carried out: the optimal vale profiles for the implementation of EIVC and LIVC strategies change with the engine speed at constant load, the study of the three load configurations on all the engine speed range would have been too long.

The successive paragraphs analysed respectively the configurations of the engine simulated at 66%, at 27% and at 11% of full load: in each of them the load control methods optimized for the specific load have been explained, showing the opening angle of the throttle plate or the modified valve lift profile for the EIVC and LIVC strategies and pointing out the main performance provided by each single strategy.
At the end of each paragraph, a graphical comparison between the methods of load reduction has been done looking at the diagrams of volumetric efficiency, indicated efficiency and brake specific fuel consumption, then a careful analysis of the pressure-volume diagram expressed in logarithmic scale has been performed to find the motivation of the differences in the results given by each load reduction method.

In all the three load configurations it has been demonstrated that the EIVC strategy is the one that ensures the lowest specific fuel consumption and the highest indicated efficiency, because of its higher indicated work due to the lower pumping losses with respect to the other strategies: the early intake valve closing allows to draw into the cylinder the exact quantity of air needed to reduce the load and consequently avoids the pressure drop due to the passage of the air through a restricted section typical of the throttled engine, and also avoids the reduction of positive work due to the rejection of the air excess back to the intake duct (which also causes additional pumping losses) performed by the LIVC strategy.

The paragraph 3.5 consisted in the graphical comparison of all the results previously obtained, performed plotting on column diagrams the volumetric efficiencies, the indicated efficiencies and the specific fuel consumptions given by the three control modes in all the load configurations.
The information given by these plots confirms what has been stated in the previous part: the EIVC strategy ensures the lowest specific fuel consumption and the highest indicated efficiency whatever the load, and its advantage on the other strategies of load reduction increases at low engine loads.

In the last two paragraphs, the open and closed cycle efficiency analysis have been performed: the P-V diagram has been integrated separating the positive part by the negative one, to derive the gross indicated mean effective pressure and the net one, and then the gross indicated power.
The analysis demonstrated in a quantitative way that the EIVC strategy is much more effective in the reduction of the pressure losses with respect to the LIVC one, while is useless to improve the efficiency of the compression, combustion and expansion phases.

In conclusion, the best way to reduce the efficiency loss at partial load of a naturally aspirated engine is to reduce the load by means of an early close of the intake valve.
The next chapter will be focused on the same kind of analysis applied on a turbocharged engine.

CHAPTER 5

Load control in a four-cylinder turbocharged engine obtained by the Miller cycle.

5.0 overview

In this chapter, all the previous analysis is repeated on a four-cylinders turbocharged engine, whose model will be simulated at full load and at 70%, 48.7% and 37.8% of it: all the load reductions will be performed by means of the throttling valve, the early inlet valve close and the late one, then the results provided by each method in terms of efficiency and specific fuel consumption will be compared, to define the best one and to study the effect of the Miller cycle in the efficiency improving.

The engine under analysis is not a real engine nowadays on the market, but is an experimental model based on a turbocharged engine produced by FIAT on which a Multi-Air system is implemented; the turbocharging is obviously pulsating type, since it's the most addicted to the road traction, thanks to its quick response to the variations of speed and load due to the small distance between the exhaust valve of each cylinder and the turbine inlet, which allows to fully exploit the kinetic and pressure energy of the exhaust gases instead of wasting it in the large volume typical of the constant pressure turbocharging system.

Differently from the naturally aspirated engine, the turbocharged one is characterized by another parameter of load control, that is the boosting pressure: the higher is the boosting pressure at a specific speed, the higher is the mass of air entering into the cylinder and consequently the higher is the load, this means that the torque and power can be reduced lowering the pressure provided by the compressor instead of throttling the engine, avoiding so the pressure drops related to the throttle plate.

The boosting pressure provided by the compressor can be reduced by means of a wastegate valve, that makes a part of the exhaust gases bypass the turbine, which consequently can generate a lower power to move the compressor, forcing it to produce a lower boosting pressure.

This method cannot be used to apply a high load reduction, since if the boosting pressure is reduced to the atmospheric one, the engine returns to be a naturally aspirated one, so, in order to study its effect on three load configurations, the engine will be always analysed above the 36% of the full load, because it has been derived that the atmospheric pressure is reached by the compressor at that load

As for the optimal values of effective valve closure angle and maximum valve lift (for EIVC strategy), the boosting pressure which allows to simulate the engine at a precise load level is determined by trials and errors; the volumetric efficiency, the indicated efficiency and the specific fuel consumption produced by the action of the wastegate valve will be compared with the ones obtained by the previous strategy of load control, to understand if this option can be better than the EIVC.

The analysis is structured in the same way of the previous chapter: firstly the engine speed at which the three levels of load will be studied is chosen looking to the torque and power curves of the engine, then the full load at that regime is studied; even in this case chosen engine speed is the one at which there is the power peak, that is 4500 rpm.

The successive steps consist in the definition of the optimal values of the angle of effective closure, the maximum lift and the boosting pressure according to the load control strategy under analysis, the simulation of each single configuration and the comparison of the data about volumetric efficiency, indicated efficiency and brake specific fuel consumption, for each load level; the last step consists in the graphical and quantitative comparison of all the results, with a distinction of open and closed cycle efficiency, from which the best solution is definitively derived.

5.1 Full load configuration

The full load configuration can be studied simply keeping the valve lift and the boost pressure at their original value, and the opening angle of the throttling valve at 90°.

The only modification has been the setting of the combustion model, passing from the Advanced model to the Wiebe one, which guarantees more reliable results at low load; differently from the model of the naturally aspirated engine, the air to fuel ratio and the parameters of the Wiebe function don't remain constant at the increasing of the engine speed, as it will be shown in the appendix.

The engine at full load provides the following curves of volumetric efficiency:

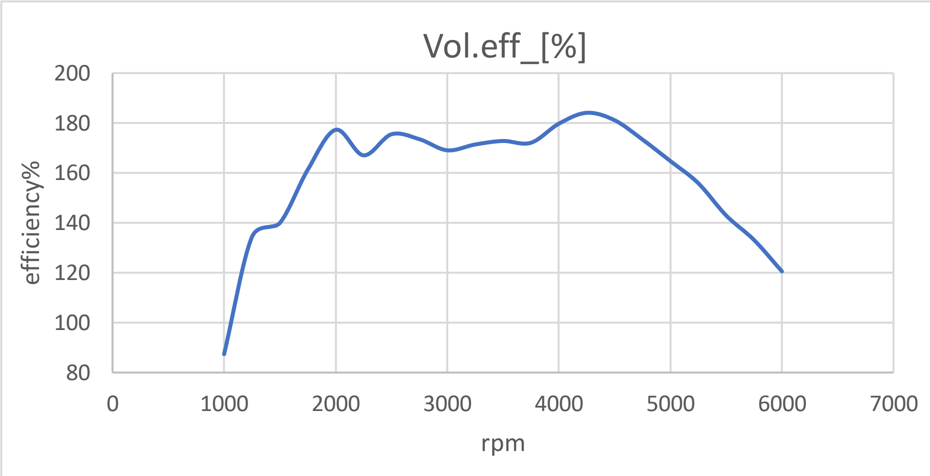

Figure 5.1 T.C. engine's volumetric efficiency

The curve of volumetric efficiency is characterized by several peaks and by an almost flat portion where the volumetric efficiency is at the highest range: this behaviour is typical of the turbocharged engine, since the compression of the air entering in the cylinder lowers the importance of the inertial and wave effects in the filling of the cylinder itself, even if their presence can be noticed in correspondence of the two peaks at 2000 and 4500 rpm, whose values are respectively 168.4% and 172.975%.
The volumetric efficiency of this engine remains always higher than 100% from 1250 to 6000 rpm, since the compression of the intake air allows the piston to draw in the cylinder an effective mass of air which is higher than the one which would completely fill it at ambient pressure; this is another typical feature of the turbocharged engines.

The curve of brake torque follows the same trend of the volumetric efficiency's one, but differently from the case of the naturally aspirated engine, the shape of the two diagrams are not exactly coincident, since the air to fuel ratio and the air density don't remain constant on the whole engine speed range, remembering that the brake torque is defined as:

$$M_e = \eta_g \lambda_v V \rho_a * \frac{H_i}{\alpha} * \frac{1}{2\pi\varepsilon}$$

The values of air to fuel ratio and boosting pressure (which influences the air density) are set to keep the resultant brake torque as constant as possible in the medium speed range, from 2000 to 4500 rpm, at the value of 200 Nm; this solution guarantees a very good drivability of the car and allows the driver to reduce the use of the gearbox, since he's no more forced to keep the engine speed in a short range to have the maximum torque available.

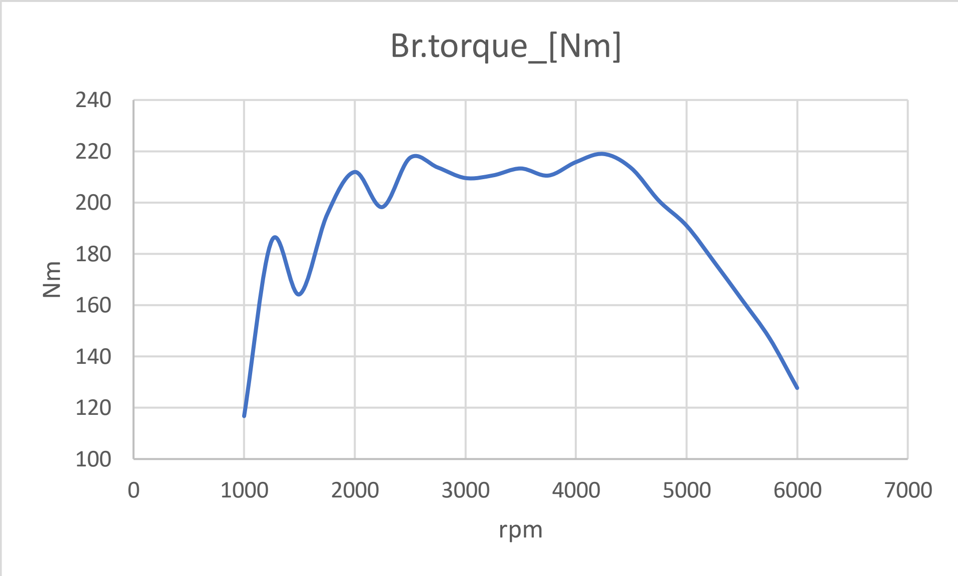

Figure 5.2 T.C. engine's brake torque

The brake power has its highest peak at 4500 rpm, the same regime where the volumetric efficiency is maximum, while in the naturally aspirated engines the speed of maximum volumetric efficiency coincides with the he one of maximum torque: this confirms that the length of the intake and exhaust manifolds is optimized to increase the peak of power at 4500 rpm, while the cylinder filling at low engine speed is recovered by means of the turbocharging, and demonstrates that the turbocharging can be used to extend the field of drivability of the vehicle.

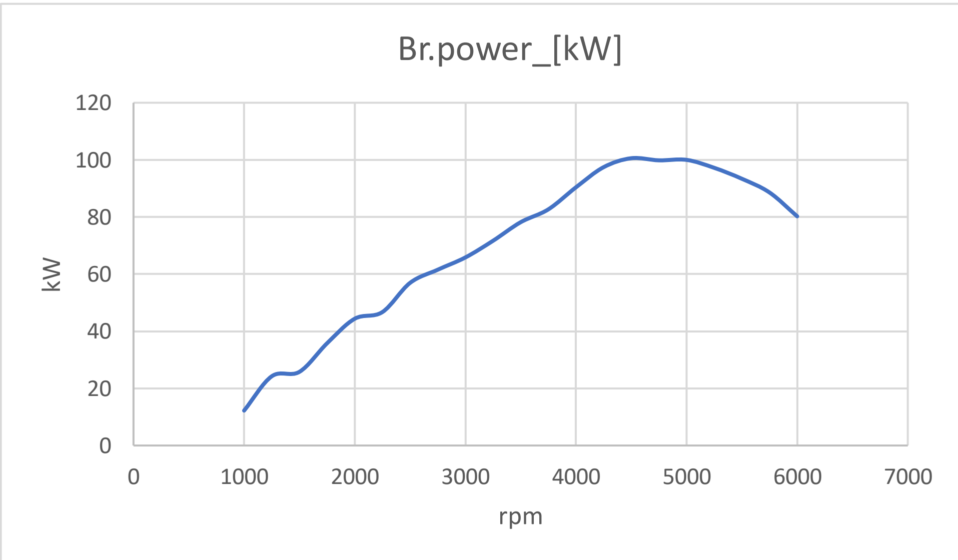

Figure 5.3 T.C. engine's brake power

The almost linear growth of the engine power guarantees a predictable behaviour of the vehicle during the acceleration phase, making it more comfortable and safer to drive.

Since the aim of the thesis is to study the effectiveness of the non-conventional methods of load control with respect to the throttling valve, the most interesting engine speed at which perform all the simulations is the one where there is the highest peak of volumetric efficiency, where the mass of air drawn into the cylinder is maximum and so the effect of the pumping losses are more evident; this engine reaches the highest volumetric efficiency at 4500 rpm, so the following analysis will be performed at that regime.

Looking to the original intake valve lift profile at 4500 rpm, it can be noticed that the engine under analysis is equipped by a system of variable valve action, since the shape of the curve is different from the one which would be impressed by a fixed cam: the maximum lift is kept constant from 140° c.a. to 180° c.a. and there is a sort of secondary peak from 225° to 250° instead of a regular decrease of the lift; even this turbocharged engine is characterized by a clearance equal to 0.2 mm between the cam and the valve, which influences the actual angles at which the intake valve touches its seat.

The variable valve train implemented on this engine is used to vary the inlet valve's closure delay with the engine speed: the closure delay is increased at the increasing of the engine speed, to obtain a better exploitation of the ram effect, since the air mass is characterized by a high inertia which draws it into the cylinder for a longer angular period after the bottom dead centre.

The intake valve has an opening angular period of 307° c.a. and the absolute crank angle of intake valve opening is 264° c.a., while the angle of effective exhaust opening is 97° c.a.; considering the clearance, the intake valve actually opens at 273° and closes at 569°, with an opening period equal to 296° c.a.

As for the analysis related to the naturally aspirated engine, the valve lift profile will be modified in order to implement the EIVC or the LIVC strategy for the load reduction, but the effective angles of intake and exhaust valve opening won't be modified in any phase of the analysis.

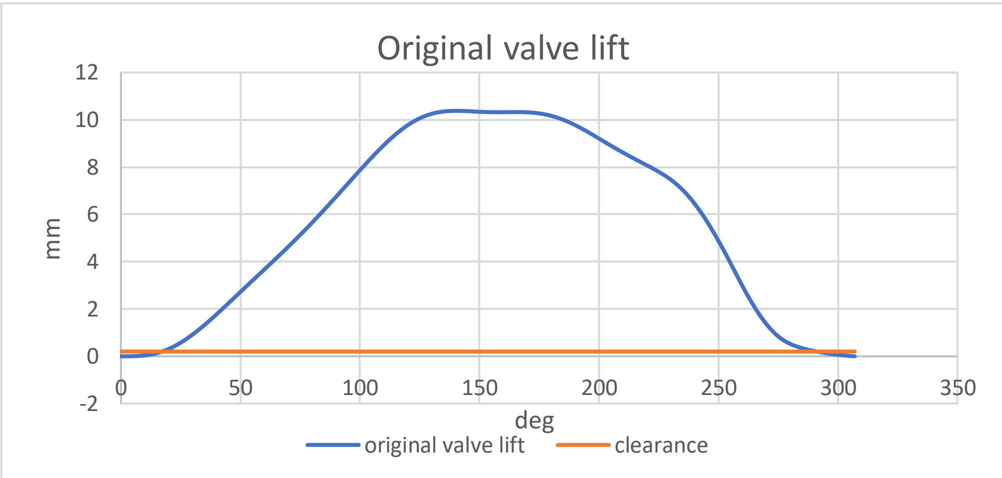

Figure 5.4 T.C T.C. engine's original valve lift

The main performance of the engine at 4500 rpm are listed below:

Volumetric efficiency	184.316 %
Brake torque	216.223 Nm
Brake power	101.893 kW
BSFC	71.985 g/MJ
Indicated efficiency	35.2%

The engine performance obtained before are a very clear example of the downsizing concept: the engine under analysis is a four-cylinders 1400 c.c. turbocharged engine, which provides a torque and a power higher by 29.14% and by 26.33% with respect to the ones given by a 2000 c.c. four-cylinders naturally aspirated engine, with a decrease of specific fuel consumption by 3.28%.

The lower displacement of the engine allows to reduce the size and consequently the mass, leading to a lower global fuel consumption and to an improvement of the vehicle's drivability.

These values will be used as a reference in all the following comparisons.

5.2 Engine simulated at 70% of full load

The study of the engine's behaviour at partial load will be performed in this paragraph and in the following ones: three levels of load reduction will be analysed, considering the main performances at 70%, at 48,7% and at 37.8% of full load; in this paragraph, the first case will be considered.

Each load reduction will be obtained by means of the systems previously described, that are the throttling valve, the early intake valve closing, the late intake valve closing and the lowering of the boosting pressure; their comparison will be performed at the end of each paragraph.

As for the case of the naturally aspirated engine, the perfect coincidence of the outputs given by the different control strategies is extremely hard to obtain, since the engine model is very sensitive to low variations of the control parameters (effective intake valve closing, valve lift profile, throttling valve's opening angle and boosting pressure), which cannot be varied in a continuous way; this is the reason why the engine is not simulated at 50% and at 40% of full load.

5.2.1 Load reduction by means of the throttling valve

The engine can express the 70% of its maximum potential if throttled with the throttle plate inclined by 32.6° with respect to the vertical position, while the valve lift profile and the boosting pressure are fixed at their original value.

The strong similarity with the analogous load configuration simulated on the naturally aspirated engine can be noticed: a reduction of the engine load by 30% requires a decrease of the butterfly valve's opening angle by 63.78%, confirming that the pressure losses caused by the passage of the air through a restricted section become important only when the throttle plate's angle of aperture becomes lower than 45°, considering that Gasdyn is equipped with an algorithm which takes into consideration a partial pressure recovery downstream of the contraction when the angular position of the valve is above 50°.

The engine set-up previously described leads to the following main performances:

Volumetric efficiency	138.406%
Brake torque	153.172 Nm
Brake power	72.18 kW
BSFC	76.307 g/MJ
Indicated efficiency	34.2%

The volumetric efficiency is lowered by 24.9% with respect to the full load configuration but remains over the 100% because of the turbocharging, while the indicated efficiency decreases by 2.84% and the specific fuel consumption increases by 6%.

The percentages of variation of the indicated efficiency and the BSFC are lower than the ones which resulted from the same kind of load reduction in the naturally aspirated engine, since the turbocharging contributes to compensate the pressure drops due to the throttle plate.

5.2.2 Load reduction by means of EIVC strategy

The effective mass of air drawn into the cylinder can be reduced anticipating the closure of the intake valve, as for the naturally aspirated engine.

The effective opening period and the lift of the intake valve have been determined by trials and errors, using the same method explained in the overview of the pervious chapter, which requires the cutting of the valve lift profile at a precise abscissa in the ascendant part, the joining of the symmetric portion in the descendant part with the previous one and the rounding of the edge resulting from the junction.

Observing the valve lift profile which implements the EIVC strategy compared with the original one, it can be appreciated how remarkable the reduction of the profile's area is: given the higher density of the air due to the turbocharging, to obtain the same load reduction gained in a naturally aspirated engine, a more severe reduction of the intake valve's opening period must be performed.

This explains why the opening period and the maximum lift are respectively reduced by 41.53% and by 25.72%: the maximum lift is now equal to 7.71 mm instead of 10.38 mm and its reached after a crankshaft rotation equal to 99° c.a. instead of 140°, while the valve is closed after 179.5° instead of 307° c.a.; due to the clearance between the valve and the cam, the valve effectively touches its seat after 164.5° c.a.

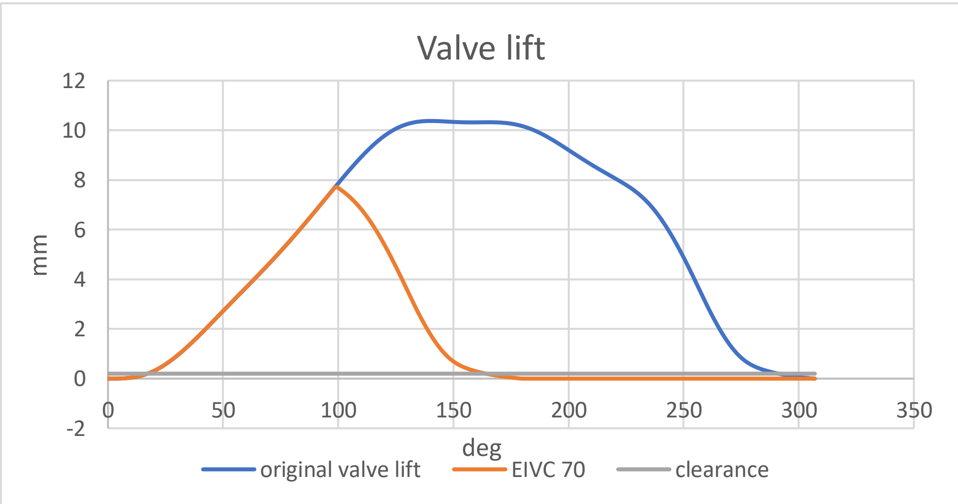

Figure 5.5 EIVC for 70%

The EIVC strategy previously defined makes the engine express the following performances:

Volumetric efficiency	135.179%
Brake torque	153.393 Nm
Brake power	72.285 kW
BSFC	74.421 g/MJ
Indicated efficiency	35%

The EIVC strategy implemented to obtain a reduction of the load by 30% allows to decrease the BSFC by 2.47% and to increase the indicated efficiency by 2.34% with respect to the throttled engine: the little improvement demonstrates that the pumping losses due to the throttling valve still have a low importance, as for the case of the naturally aspirated engine.

The decrease of volumetric efficiency by 2.18% is due to the necessity of a lower air mass to obtain the same power output, due to the reduction of the pumping losses.

5.2.3 Load reduction by means of LIVC strategy

The air mass entering into the cylinder can be reduced by means of a delayed closure of the intake valve which allows to push the excess of air back to the intake manifold, as in the naturally aspirated engine.

The valve lift profile which implements the LIVC strategy has been defined with a procedure different from the one used to obtain it on the naturally aspirated engine, since the shape of the profile has been modified: given the high sensitivity of the numerical model on low variation of the control parameters, the optimization of the intake valve's opening period was no more sufficient to guarantee the perfect coincidence of the resulting torque and power with the ones provided by the other methods of load reduction, so the shape of the curve has been simplified, making it symmetric and deleting the second lift; after this change in the profile's form, the angle of effective valve closure has been set by trials and errors.

The maximum lift, equal to 10.38 mm, is kept constant for 63.5° c.a., increasing the opening angular period by 11.7%, which is almost the same percentage registered in the respective case in the analysis of the naturally aspirated engine: given the equality of the pressure in the cylinder and in the intake manifold, the expulsion of the air excess is not helped by a blowdown process, so the piston has to make the same effort to push the same quantity of air back to the intake manifold.

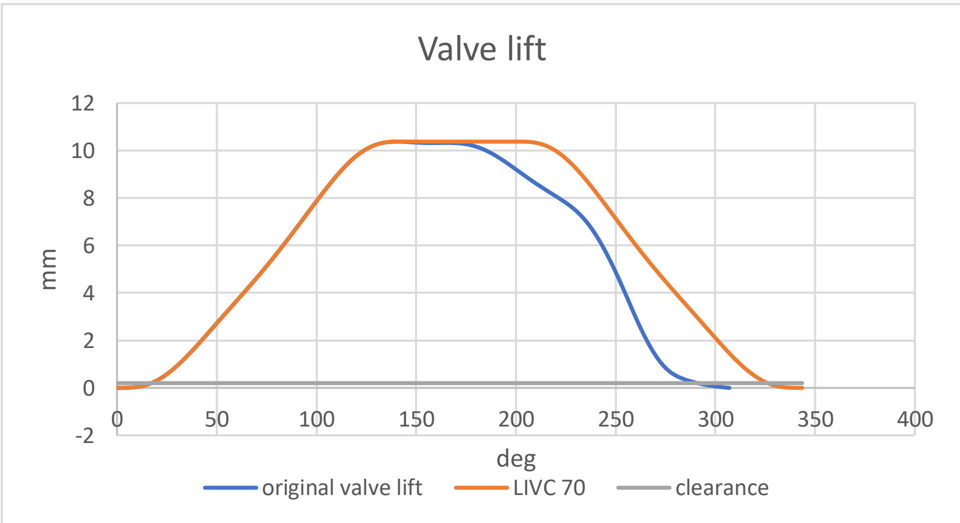

Figure 5.6 LIVC for 70%

The LIVC strategy previously shown guarantees the following engine performances:

Volumetric efficiency	135.551%
Brake torque	153.193 Nm
Brake power	72.191 kW
BSFC	74.72 g/MJ
Indicated efficiency	34.9%

The improvements in the specific fuel consumption and in the indicated efficiency are slightly lower than the ones guaranteed by the EIVC strategy, with a decrease of BSFC by 2% and an increase of indicated efficiency by 2% with respect of the throttled engine.

The lower improvements with respect to the EIVC strategy are due to the reduction of the useful work due to the expulsion of the air excess, which leads to a higher air requirement to get the same performances (underlined by the higher volumetric efficiency) as it has been seen for the naturally aspirated engine.

5.2.4 Load reduction by means of the lowering of the boosting pressure

The last method which allows to reduce the engine load by 30% is the lowering of the boosting pressure: a wastegate valve deviates a part of the exhaust gases far from the turbine, which consequently can exchange a lower power with the compressor and so makes it capable to provide a lower increment of pressure; given the compressibility of the air, a lower pressure implies a lower density (if the temperature remains constant), which is traduced in a reduction of the mean effective pressure, whose formula is reported below:

$$p_{me} = \eta_g \frac{H_i}{\alpha} \lambda_v \rho_a$$

The numerical model of the engine implemented in Gasdyn is equipped by an algorithm which simulates the presence of an intercooler, a heat exchanger which reduces the inlet air's temperature to increase its density, which in this case ensures an air's temperature equal to 323.15 K in the speed range from 4000 to 4750 rpm whichever the engine load.

Being the temperature of the intake manifold constant, the air density linearly depends on its pressure, but the direct calculation of the boosting pressure which allows to reach the desired load reduction is not possible, since the brake mean effective pressure is influenced by the engine's organic efficiency, which decreases at the lowering of the load and is determined a posteriori like the indicated one, recalling that:

$$\eta_g = \eta_o \eta_i$$

Since the problem of the boosting pressure's calculation is implicit, the only way to find the one which gives as output the required brake torque and brake power at the end of the simulation is the trials and errors approach.

The boosting pressure which makes the engine run at 70% of the full load resulted to be equal to 1.591 bar, which means a decrease by 29.29% with respect to the original one, equal to 2.25 bar; since the brake torque is linearly dependent on the brake mean effective pressure, which linearly depends on the boosting pressure, this means that the decrease of the indicated and organic efficiencies reduce the load by 0.71%, with an overall contribution in the load reduction of 2.37%.

The engine simulated with a boosting pressure equal to 1.591 bar provides the following performances:

Volumetric efficiency	135.63%
Brake torque	153.451 Nm
Brake power	72.312 kW
BSFC	74.639 g/MJ
Indicated efficiency	34.9%

The table above shows that this method guarantees an improvement in terms of indicated efficiency and specific fuel consumption similar to the one provided by the delayed closure of the intake valve: the first one increases by 2% while the second one decreases by 2.19% with respect to the throttled engine.

These enhancements allow the engine to provide the 70% of the maximum torque and power using a lower air quantity: the volumetric efficiency is lowered by 2%.

The analysis performed so far demonstrated that the best method to efficiently reduce the engine load by 30% is the early intake valve closing, which avoids the pressure losses generated by the passage of the air through a restricted section which are typical of the throttling valve and also avoids the reduction of the useful work due to the expulsion of the air excess which affects the LIVC strategy.

5.2.5 Graphical comparison between the control strategies

The results previously obtained are now synthetized in a graphical comparison: the diagrams of volumetric efficiency, indicated efficiency and specific fuel consumption are plotted to visualize the order of magnitude of the benefits given by each method of load reduction; the main reasons of the differences between the control systems are discussed in the analysis of the P-V diagram.

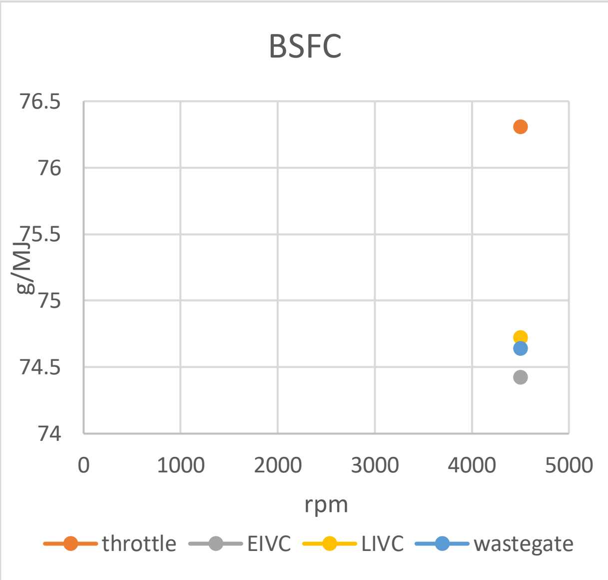

Figure 5.7 BSFC at 70%

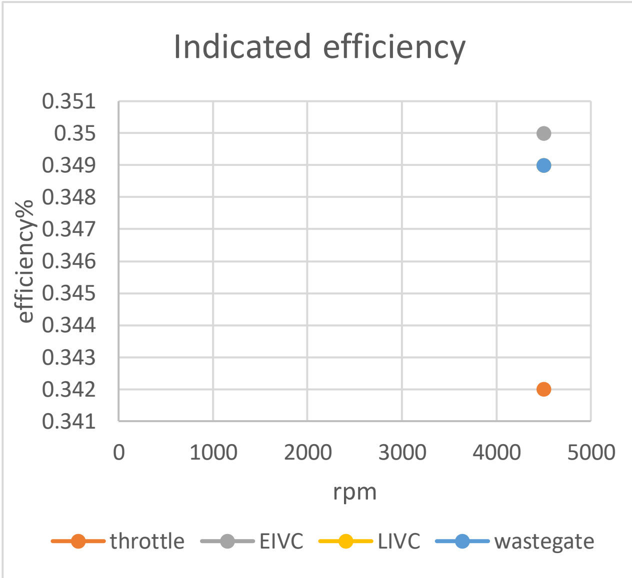

Figure 5.8 Indicated efficiency at 70%

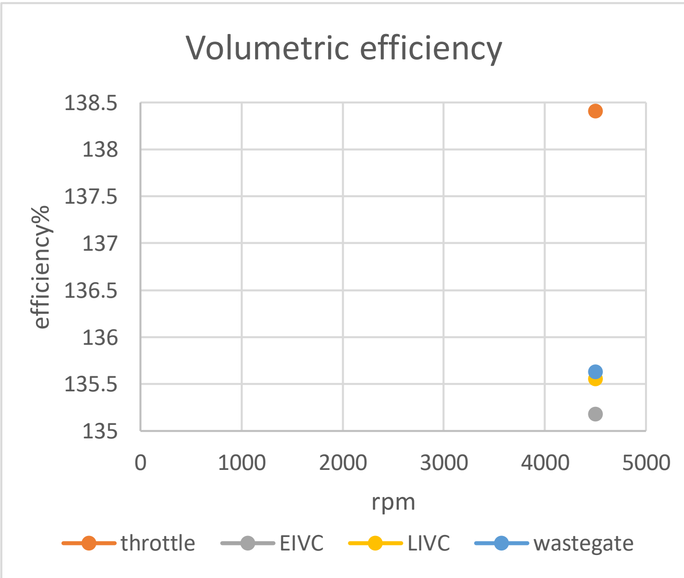

Figure 5.9 Volumetric efficiency at 70%

The graphs of indicated efficiency and BSFC remark that the early intake valve closing gives the best results, providing the highest volumetric efficiency and the lowest BSFC; the increase of specific fuel consumption is higher than the decrease of volumetric efficiency because of the reduction of the organic efficiency.

This trend is confirmed by the graph of the volumetric efficiency, which shows that the EIVC strategy provides the lowest one, because of the lowest mass of air required to generate the same power output.

The pressure-volume diagram expressed in a logarithmic scale allows to understand the reason why the EIVC is the best method for a low load reduction: the diagram generated by the engine simulated by a lower boosting pressure results to be the perfect downward translation of the full load one and its negative area is lower than the one which characterizes the curve generated by the throttled engine, but the positive area is lower than the ones resulting from the other control strategies, so the net useful work generated during the cycle results to be slightly lower than the one provided by the early inlet valve closing.

The edge in the clockwise part is due to the fact that the combustion starts after the top dead centre.

The EIVC strategy is characterized by a higher pressure drop at the middle of the intake phase, when the valve opening is maximum, with respect to the LIVC one, because of the lower lift which forces the air to pass through a more restricted section, as it happens in the case of the naturally aspirated engine; the overall pumping area is however lower thanks to the anticipated valve closure which generates an edge in the diagram: the curve tracks the same path in opposite verse, so the resultant work is null.

The quantitative comparison of the P-V diagrams expressed in a linear scale will be made in the last paragraph, during the discussion of the open cycle efficiency.

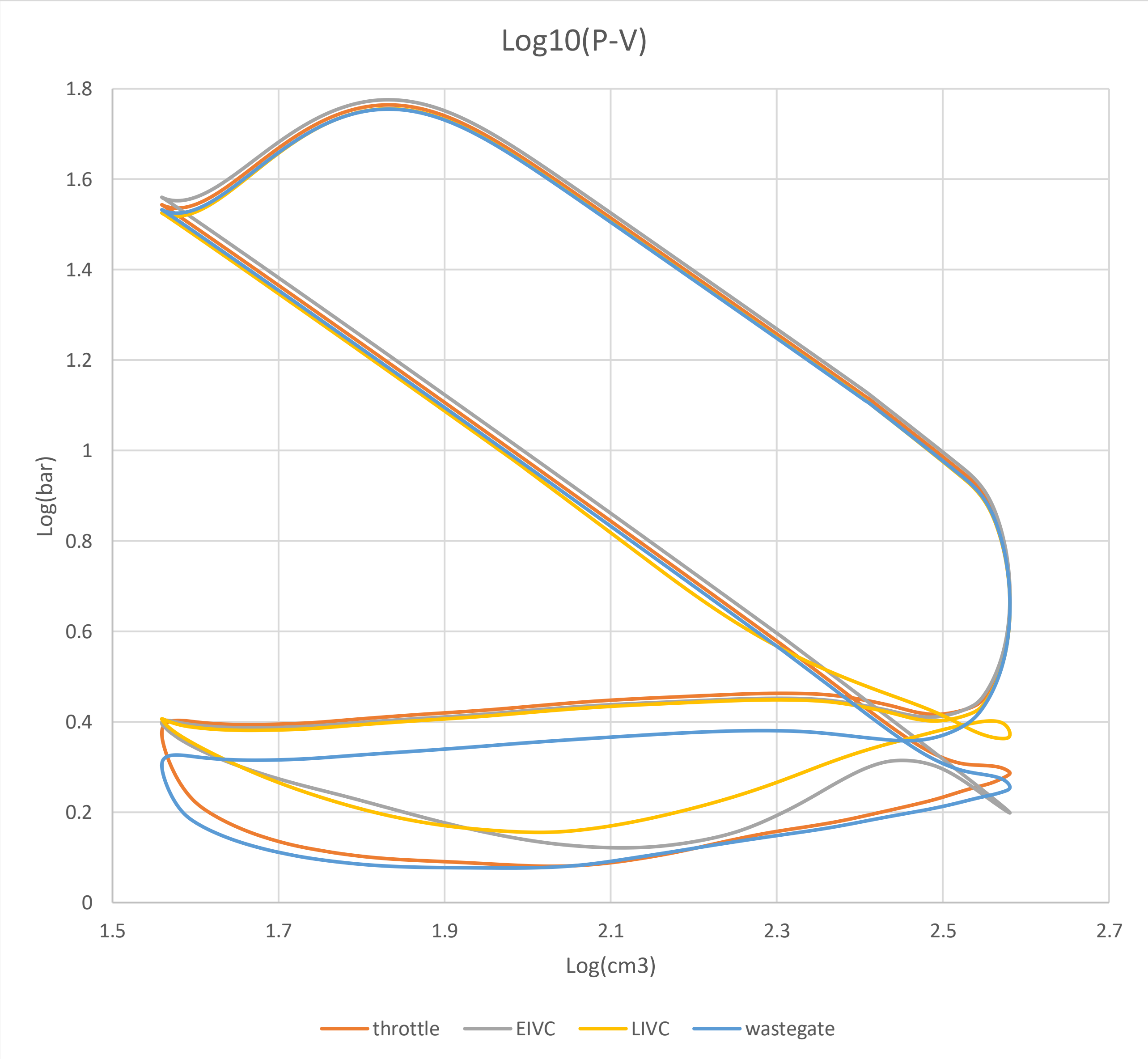

Figure 5.10 comparison between the P-V diagrams at 70%

5.3 Engine simulated at 47.8% of full load

The engine is now run at 47.8% of its full load, with the same procedure as before: each method of load control will be analysed independently from the other ones, with a comparison at the end of the paragraph.

The intake valve will be closed earlier in case of EIVC strategy or later in case of LIVC one, to entrap a lower mass of air during the intake stroke or to push back a higher excess during the compression one, while the angle of aperture of the throttle plate and the boosting pressure will be reduced; the effect of the pumping losses will be emphasized with respect to the previous case.

5.3.1 Load reduction by means of the throttling valve

The throttling valve's opening angle which halves the engine load is equal to 26°, this means that a decrease of the plate's aperture by 20.45% causes a load reduction by 31.3% with respect to the previous case; the valve is now operating in the range in which even a single grade of the opening angle makes a remarkable difference in the engine's power output: if the angle is set to 27°, the resultant brake torque is 114 Nm instead of 105.2 Nm, so a single degree of aperture causes an increase of brake torque by 8.37%.

It must be pointed out that the relationship between the angle of aperture of the butterfly valve and the cross section available for the air passage is highly non-linear, as it's shown in [9]:

$$\frac{4A_{th}}{\pi D^2} = 1 - \frac{\cos\varphi}{\cos\varphi_0} + \frac{2}{\pi}\left[\frac{a}{\cos\varphi}\sqrt{\cos^2\varphi - a^2\cos^2\varphi_0} - \frac{\cos\varphi}{\cos\varphi_0}\arcsin\left(\frac{a\cos\varphi_0}{cos\varphi}\right) - a\sqrt{1-a^2} + \text{arcsin}a\right]$$

Where A_{th} is the theoretical restricted area, D is the plate's diameter, $a = \frac{d}{D}$ where d is the diameter of the throttle's shaft, φ_0 is the angle of complete valve closure and φ is the effective opening angle.

The theoretical restricted area must be multiplied by the coefficient of discharge to obtain the effective area through which the air passes:

$$C_D = 1 - \left[1 - \left(\frac{d_{area}}{D}\right)^4\right] * \left[0.2 + 0.2 * \left(1 - \left(\frac{d_{area}}{D}\right)^4\right)\right]$$

Where d_{area} is the restricted area 's equivalent diameter.

The engine simulated with this throttle opening can provide the performance listed in the table:

Volumetric efficiency	102.598%
Brake torque	105.261 Nm
Brake power	49.603 kW
BSFC	82.313 g/MJ
Indicated efficiency	33.1%

The enforcement of the pumping losses makes the indicated efficiency decrease by 3.22% with respect to the previous case and by 5.97% with respect to the full load one, while the percentages of increase of the specific fuel consumption are respectively equal to 7.87% and 14.37%; the volumetric efficiency has been lowered by 25.87% with respect to the previous load configuration, because of the lower mass of air required to reach the 48.7% of the full load.

5.3.2 Load reduction by means of EIVC strategy

The halving of the engine load by means of the EIVC strategy obviously requires a more severe decrease of the intake valve's opening period, which makes the piston suck a lower mass of air into the cylinder.

While the angle among which the valve opens remains unvaried, the one among which its closure happens is reduced by 3.9% with respect to the configuration needed to run the engine at 70% of full load, since now the lift profile becomes equal to zero after 172.5° c.a. while the effective complete valve closure is obtained after 149.5°, with a reduction of the real opening period by 8.8%; the maximum peak, now equal to 6.68 mm, is decreased by 13.36% with respect to the previous case and is reached after 90° c.a.

Differently from the previous case and all the ones simulated in the model of the naturally aspirated engine, a slight modification of the profile's shape was needed to match the output torque and power with the ones provided by the other methods of load reduction: since even a half of crank angle degree in the position of the profile's maximum peak caused a remarkable difference in the main results, a distributed increase of the descendant part of 0.05 mm has been performed to obtain the little increment of torque and power needed to reach the target values.

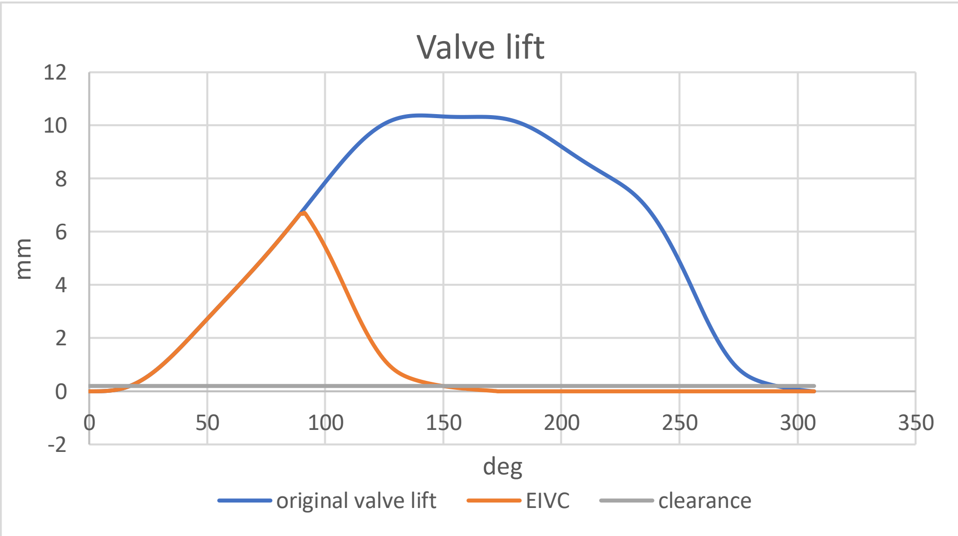

Figure 5.11EIVC for 48.7%

The valve action explained before gives the engine the following performances:

Volumetric efficiency	98.847%
Brake torque	105.259 Nm
Brake power	49.602 kW
BSFC	79.308 g/MJ
Indicated efficiency	34.3%

The effect of the EIVC strategy in the avoiding of the pumping losses generated by the throttle plate is now clearly expressed by the decrease of the specific fuel consumption by 3.7% and by the increase of the indicated efficiency by 3.63%, while the reduction of the volumetric efficiency by 3.7% confirms that a lower mass of air is required by the engine to express the same torque and power.

The organic efficiency of the engine equal to 85.5% makes the brake quantities be very close to the indicated ones, so the percentages of enhancement seen above are almost coincident, as it happened when the engine has been simulated at 70% of full load with an organic efficiency equal to 89.2%.

5.3.3 Load reduction by means of LIVC strategy

The engine load can be halved also by means of a delayed closure of the intake valve which allows to push a higher excess of air back to the intake manifold.

After some trials and errors, the theoretical angle at which there is the complete valve closure resulted to be equal to 359°, while the real one imposed by the clearances is equal to 345° c.a., this means that the theoretical opening period has been increased by 16.94% with respect to the original valve lift, while the effective one has been increased by 16.55%; the maximum peak of the lift profile has been kept constant for 82° c.a., increasing the angular period of maximum valve aperture by 29.13% with respect to the same profile needed to simulate the engine at 70% of full load.

Given the simplicity of implementation, the modification of the profile's shape previously derived has been used also for this load configuration, even if a slight decreasing of the descendant part has been necessary to gain a little increase of torque and power needed to obtain the perfect coincidence with the ones provide by the other control strategies: a lower lift in the descendant part of the profile allows to push a lower amount of air back to the inlet duct, allowing an increase of performance.

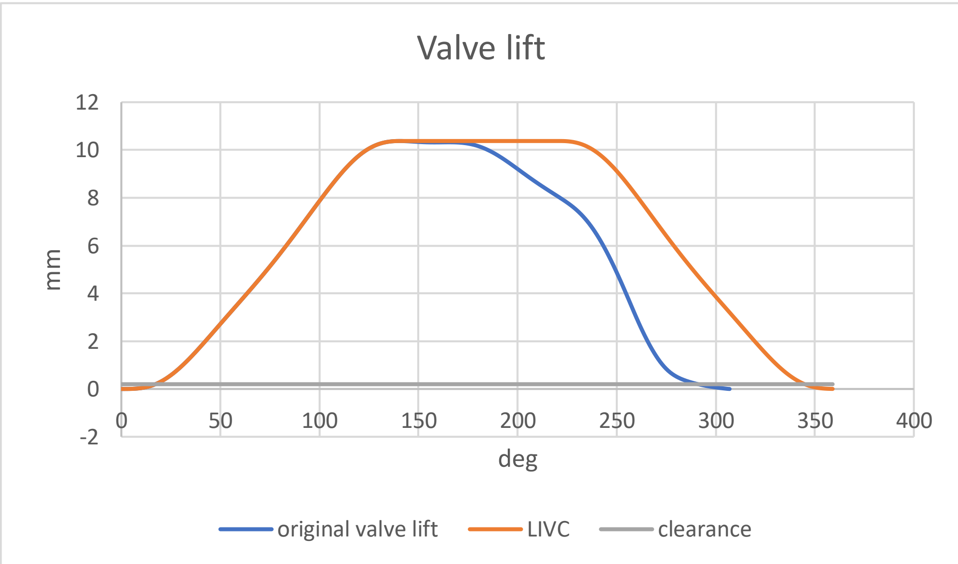

Figure 5.12 LIVC for 48.7%

The engine equipped with this valve acting is capable to give the following main performances:

Volumetric efficiency	100.351%
Brake torque	105.281 Nm
Brake power	49.612 kW
BSFC	80.491 g/MJ
Indicated efficiency	33.8%

The improvement in indicated efficiency and specific fuel consumption with respect to the throttled engine is slightly more remarkable, since the first one is increased by 2.11% and the second one is decreased by 2.21%: the load reduction has not such a severity to make the throttling valve generate high pumping losses; the percentage of enhancement are almost equal because of the high organic efficiency, equal to 85.5%.

The lower benefit given by this strategy with respect to the EIVC one is explained by the reduction of useful work due to the expulsion of the air excess, as it will be seen in the analysis of the P-V diagram: a higher mass of air is needed to reach the desired power output, so the volumetric efficiency increases by 1.52%.

5.3.4 Load reduction by means of the lowering of the boosting pressure

The engine load can be reduced to the half of the maximum one also by the lowering of the boosting pressure, with a wider aperture of the wastegate valve which makes a higher mass of burned gases bypass the turbine, that becomes capable to generate a lower power and so reduces the maximum boosting pressure which can be provided by the compressor.

The air temperature at the outlet of the intercooler is equal to 323.2 K, the same value registered in the analysis of the engine run at 70% of the full load, this means that there is a linear relationship between the decrease of the air density and the load reduction.

The optimal boosting pressure which allows the engine to run at 48.7% of its maximum load resulted to be equal to 1.439 bar: a decrease by 10.56% of the pressure of the air entering in the cylinder generates a reduction of the engine load by 31%, this confirms that the linear dependence of the brake mean effective pressure on the air pressure cannot be used to predict the boosting pressure unless a model which estimates the reduction of the organic and indicated efficiencies is implemented.

This boosting pressure makes the engine provided the following performances:

Volumetric efficiency	99.12%
Brake torque	105.651 Nm
Brake power	49.787 kW
BSFC	79.228
Indicated efficiency	34.3%

As for the case of the low load reduction, the percentages of improvement of specific fuel consumption and indicated efficiency are almost coincident with the ones guaranteed by the EIVC strategy: the second one is actually coincident, while the first one is decreased by 0.1%, which is a negligible percentage.

Even the volumetric efficiency remains substantially constant, since it has been lowered by 0.28%.

The coincidence of the results provided by the EIVC strategy and the lowering of the boosting pressure are due to the balancing between these phaenomena: the EIVC strategy can guarantee lower pressure losses while the lowering of the boosting pressure provides a wider positive area in the P-V diagram.

The analysis of engine run at its mid load demonstrated that the best methods which partialize the engine in an efficient way are equivalently the early intake valve closing and the lowering of the boosting pressure, thanks to their avoiding of the pressure drops which would be generated by a throttle plate and the loss of useful work due to the expulsion of the air excess back to the intake manifold.

The wastegate valve is cheaper than the variable valve train system but it cannot perform high load reduction, since the boosting pressure can't be lowered below the atmospheric one; the two control strategies are often both installed on the turbocharged engine for safety shake, while their combined use is also possible.

The wastegate valve is however necessary, since the turbochargers are normally designed to give their highest supercharging rate in the range of 20-40% of the maximum engine speed to favourite the vehicle's drivability positioning the torque peak in that range: given the direct proportionality of the turbine's power on the mass flow rate, in order to avoid too high boosting pressure at high regimes, the wastegate valve is open when the engine speed overcomes an imposed limit.

5.3.5 Graphical comparison between the control strategies

The results previously obtained are now compared in a graphical way, plotting the volumetric efficiency, the indicated one and the specific fuel consumption provided by the four strategies, to better understand the order of magnitude of the efficiency gap between each method of load reduction.

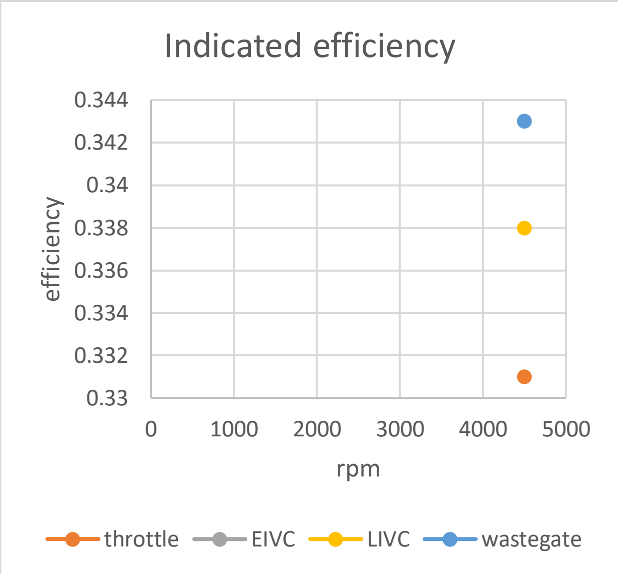

Figure 5.13 Indicated efficiency at 48.7%

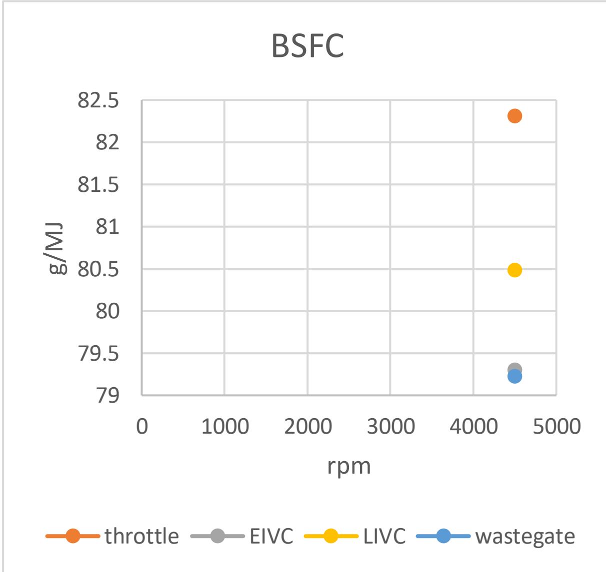

Figure 5.14 BSFC at 48.7%

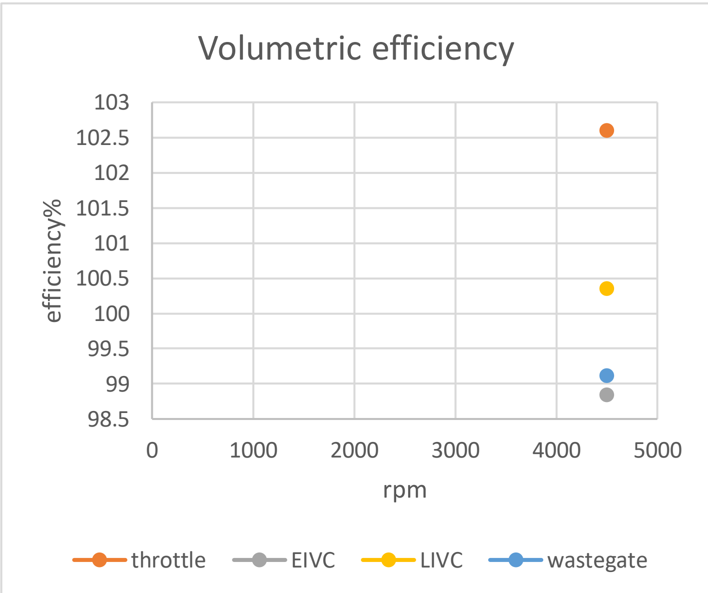

Figure 5..15 volumetric efficiency at 48.7%

The graphs about indicated efficiency and specific fuel consumption confirm that the use of the wastegate valve is absolutely equivalent to the implementation of the EIVC strategy, since their marker points are perfectly superimposed, preventing the visibility of the ones generated by the second method.

The plots also confirm that the throttling valve is the worst partializing method because of the pumping losses it suffers: its indicated efficiency is the lowest one, while its BSFC is the highest one.

The distance between the marker point representing the throttled engine and the other ones is higher on the graph of BSFC, because of the multiplication by the organic efficiency.

The analysis of the logarithmic P-V diagrams provided by each control strategy is useful to clarify the motivations of the differences in the resulting performances, as for all the cases considered so far; a focus on the comparison between the one given by the EIVC strategy and the one provided by the use of the wastegate valve, since that methods have proved to be the best ones in the halving of the engine load.

Differently from the case of low load reduction, the pressure-volume diagram generated by the reduction of the boosting pressure is no more the rigid downward displacement of the one provided by the engine running at full load, since the lower boosting pressure makes the pumping losses become more relevant because of the reduction of the air density which decreases the cylinder filling.

The pumping area of the diagram resulting from the simulation of the engine controlled by means of an early intake valve closing is lower than the one which characterizes the diagram provided by the engine analysed with a partial opening of the wastegate valve, thanks to the expansion and compression through which the air undergoes before starting the effective compression phase, tracing a path with null resultant area, but the diagram resulting from the second method is characterized by a higher positive area, since the air starts the compression phase with a higher pressure, which makes the positive area start from a lower pressure, while the remaining part of the clockwise diagram is almost equal between the two strategies, given that the air mass entrapped in the cylinder is the same.

The phaenomena explained above are perfectly counterbalanced, so the two partializing strategies give the same performance, while in the previous case the EIVC strategy was slightly superior.

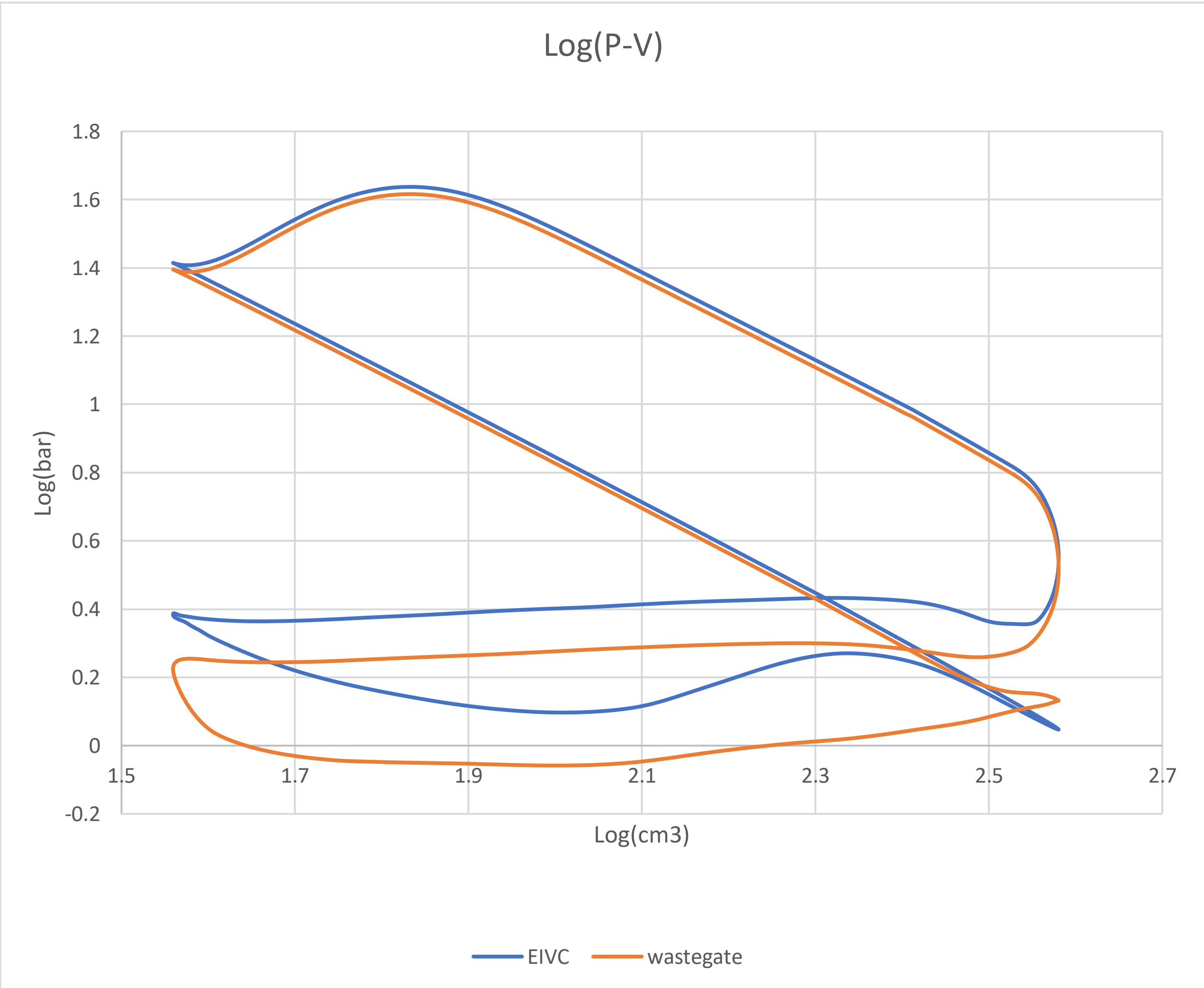

Figure 5.16 comparison between the P-V diagrams of EIVC and wastegate valve at 48.7%

The main disadvantage which chartcterizes the LIVC strategy can be seen also in this case: the process of expulsion of the air excess back to the intake manifold causes a remarkable reduction of the useful work the engine can generate since the fact that the pressure doesn't remain perfectly constant makes the first segment of the clockwise curve be not horizontal, generating a triangular area subtracted to the global positive part of the cycle.

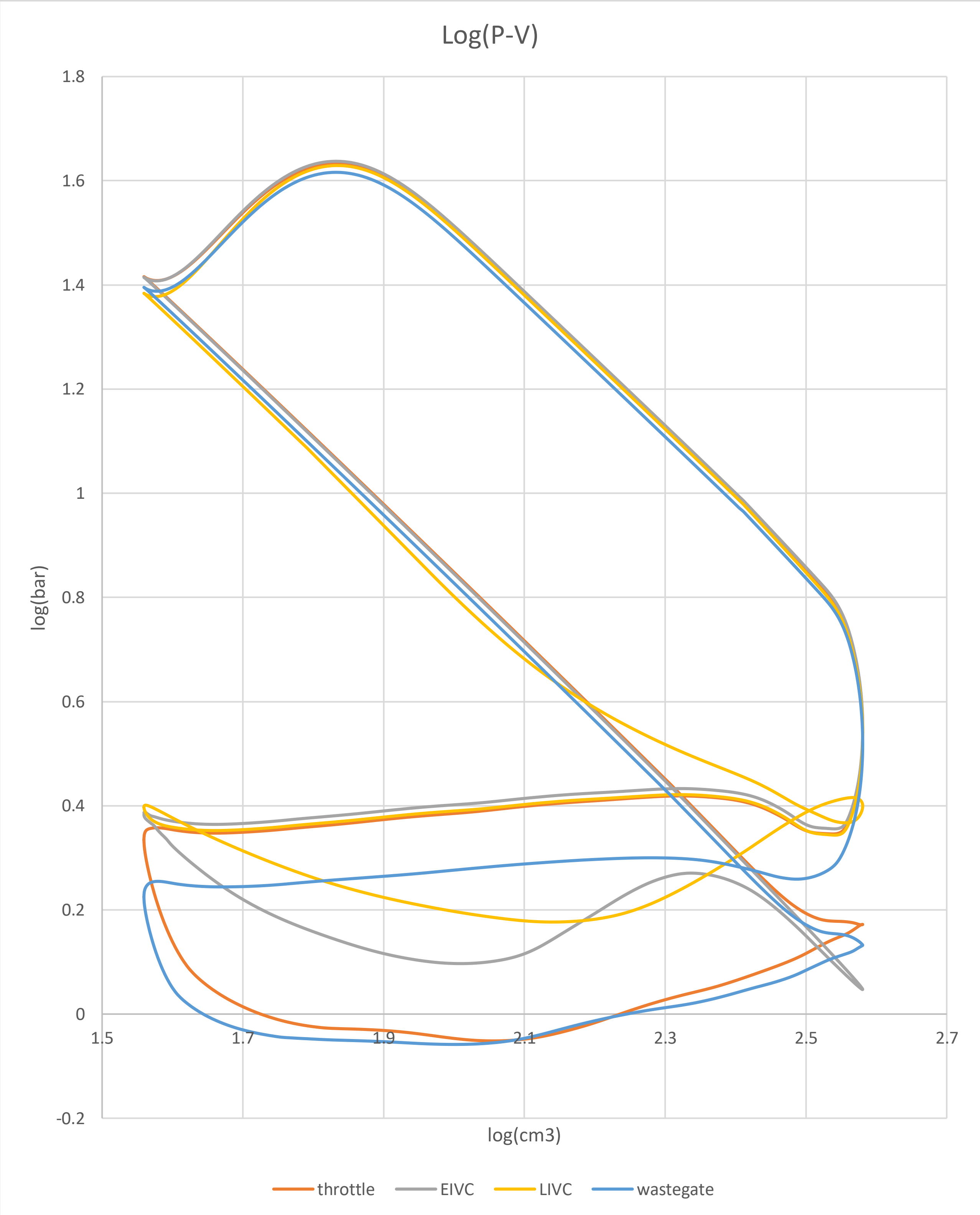

Figure 5.17 comparison between the P-V diagrams at 48.7%

5.4 Engine simulated at 38.7% of full load

The last simulation performed on the turbocharged engine regards the engine run at 37.8% of its full load; this level of partializing has been chosen since it is almost the lowest one which can be reached by means of a reduction of the boosting pressure: if that parameter is set equal to 1 bar, transforming the engine in a naturally aspirated one given the absence of pressure increase with respect to the atmospheric value, the load becomes equal to the 36% of the maximum one.

As previously said, the load configuration under analysis is not precisely equal to the 40% because of the discrete variations of the control parameters which prevent the coincidence of the torque and the power provided by the four partializing methods.

Given the increased severity of the load reduction, all the control strategy must be implemented in a harder way: the opening angular period of the intake valve must be furtherly restricted in case of EIVC strategy or enlarged in case of LIVC one, the angle of aperture of the throttling valve must be lowered and the boosting pressure must be set almost to the atmospheric value; the negative effects of the pumping losses generated by the throttle plate will be obviously enlarged.

5.4.1 Load reduction by means of the throttling valve

The throttle plate's angle of aperture which reduces the engine load by 62.2% is equal to 24.41°: a reduction of the opening angle by 6.12% causes a load decrease by 22% of the engine load with respect to the previous load configuration; this is not surprising since the angular position of the plate is now in the range in which even a single tenth of degree causes a non-negligible difference in the performances.

The performance expressed by the engine throttled with this angle are the following:

Volumetric efficiency	85.101%
Brake torque	81.759 Nm
Brake power	38.528 kW
BSFC	87.903 g/MJ
Indicated efficiency	32.2%

The indicated efficiency is decreased by 2.72% with respect to the previous case and by 8.52% with respect to the full load one, while the specific fuel consumption is increased respectively by 6.8% and by 19.34%.

These percentages are sensibly lower than the ones obtained in the simulation of the medium and high load reduction in the naturally aspirated engine, which has been analysed at 27% and at 11% of its maximum performance: the reduction of the opening angle was more severe and consequently the pressure drops were much higher, leading to a huge efficiency drop.

The percentage of variation of the BSFC is now remarkably higher than the one of the indicated efficiency, due to the reduction of the organic efficiency, which is now equal to 82.3% and so has been reduced by 3.74% with respect to the configuration of medium load and by 7.74% with respect to the maximum load one.

The volumetric efficiency is decreased by 17% with respect to the case of load reduction by 51.3% and by 53.83% with respect to the full load configuration, given the reduction of the required mass needed to lower the output power.

5.4.2 Load reduction by means of EIVC strategy

The reduction of the engine load by 62.2% obtained by means of an EIVC strategy requires a further decrease of the opening period and the maximum lift with respect to the other load levels analysed before, since a lower air mass must be entrapped into the cylinder.

It must be noticed that the mass of air passing through the intake port is proportional to the permeability of the intake valve, which is defined as the integral calculated on the opening angular period of the effective area of the valve itself, given by the product between its circular cross section and the coefficient of discharge, which is a function of the ratio between the lift and the valve's diameter [1]:

$$\mathrm{m} \sim \int_{\mathrm{t_f}}^{\mathrm{t_i}} \mathrm{A_{ef}(t)dt} = \frac{1}{\omega}\int_{\Theta_i}^{\Theta_f} \mathrm{A_{ef}(\Theta)d\Theta}$$

$$A_{ef} = C_d \frac{\pi d_v^2}{4}$$

$$C = C(\frac{h}{d_v})$$

Where ω is the engine's angular speed, since $\Theta = \omega t$

The integral can be reduced lowering the length of the integration domain, which corresponds to the opening angular period, or decreasing the effective area by means of a reduction of the lift which cause a decrease of the discharge coefficient.

The angle at which the profile returns to be equal to zero is now equal to 163.5° c.a., but the valve effectively touches its seat after 145.5° because of the clearance: the opening period is decreased by 5.22% with respect to the lift profile which implements this strategy to obtain a medium load reduction.

The maximum lift is now equal to 6.65 mm and is reached after 89.5° c.a., so these values are decreased by 0.45% and by 0.56% with respect to the previous profile, the curve from 130° to 163.5° c.a. has been lowered by 0.002 mm: given the high sensitivity of the numerical model on the control parameters, these slight modifications have been sufficient to decrease the load by 22%

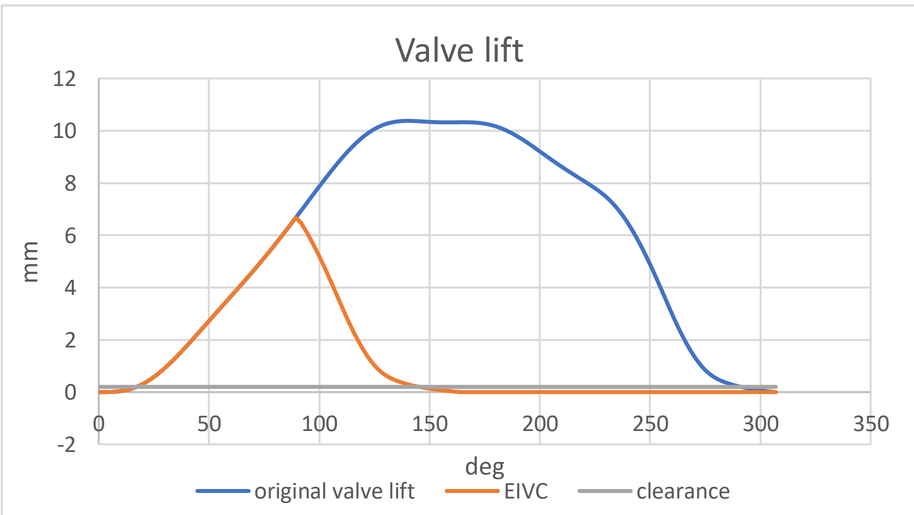

Figure 5.18 EIVC for 37.8%

The valve lift profile defined before makes the engine express the following performances:

Volumetric efficiency	80.754%
Brake torque	81.779 Nm
Brake power	38.537 kW
BSFC	83.394 g/MJ
Indicated efficiency	33.9%

The efficiency improvement given by this strategy with respect to the throttled engine is now highly remarkable: the indicated efficiency is increased by 5.3% while the specific fuel consumption is decreased by 5.13%, saving 4.509 g/MJ of fuel.

The volumetric efficiency is decreased by 5.1%, so a remarkably lower mass of air is necessary to generate the same torque and power expressed by the throttled engine.

As for the other studied load configurations, these results are due to the avoiding of the pressure losses generated by the passage of the air through the restricted section imposed by the throttling valve, so, as it has been registered in the naturally aspirated engine, the advantage of the early intake valve closing on the simple engine throttling increases at the decreasing of the load.

5.4.3 Load reduction by means of LIVC strategy

The engine load can be reduced by 62.2% also with a late intake valve close strategy, but the closure delay must be characterized by a strong severity, given the high quantity of air to be pushed back to the intake manifold: the expulsion process will last for a wide part of the cycle and it will subtract consistent part of the useful work, as it will be seen in the pressure-volume diagram.

The imposed theoretical opening period is equal to 359.5° c.a., so it increased only by a half of angular degree, that is by 0.14%, while the real angle in correspondence of which the intake valve becomes fully closed is equal to 346.5° c.a. because of the clearance between the valve's stem and the cam, so the real angular period during which the valve is open is increased by 0.435%.

The valve lift profile is almost identical to the one implemented for the halving of the engine load, only the descendant pat has been slightly increased to obtain a bit higher effective area needed for the air expulsion in the last part of the valve's opening period.

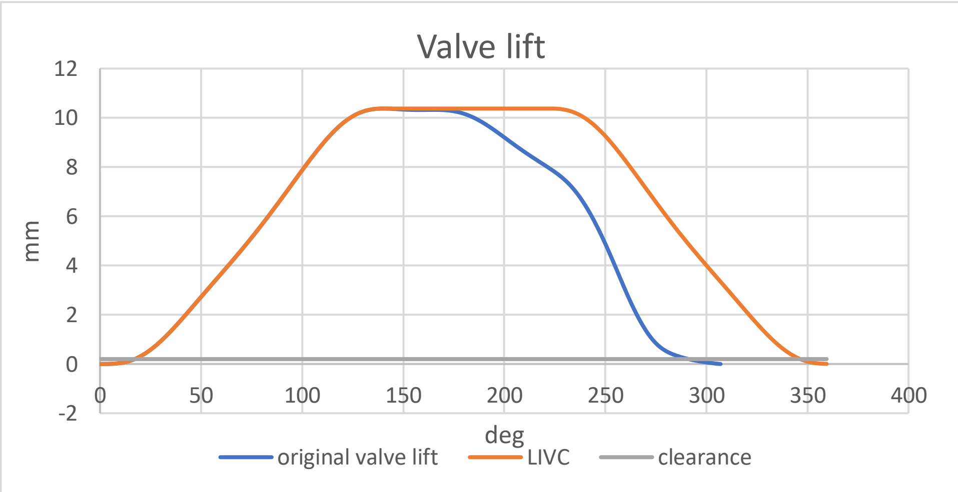

Figure 5.19 LIVC for 37.8%

This valve lift profile makes the engine capable to express the following performances:

Volumetric efficiency	82.507%
Brake torque	81.694 Nm
Brake power	38.498 kW
BSFC	85.285 g/MJ
Indicated efficiency	33.1%

The effect of the efficiency loss due to the lowering of the useful work caused by the expulsion of the air excess can be immediately seen looking to the data about the indicated efficiency and the specific fuel consumption: the first one increases by 2.8% with respect to the throttled engine, while the second one decreases by 3%, these percentages of improvement are remarkably lower than the ones guaranteed by the EIVC strategy implemented to obtain the same load reduction.

This efficiency reduction obliges the engine to entrap a higher mass of air to generate the desired torque and power, this is demonstrated by the increase of volumetric efficiency by 2.2% with respect to the one provided by the EIVC strategy.

5.4.4 Load reduction by means of the lowering of the boosting pressure

The last method by means of which the engine can be run at 38.7% of its full load is the lowering of the boosting pressure, which can reduce the effective air mass entering in the cylinder decreasing its density; given the severe load reduction, the compressor will provide a very low pressure increase, so the wastegate valve will be almost fully open to make the largest part of the exhaust gases bypass the turbine preventing it to use their residual energy to move the compressor.

As it has been already pointed out, the numerical model is equipped by an algorithm which simulates an intercooler which keeps the temperature at its outlet constant and equal to 323.2 K whichever the load, ensuring the linearity of the relationship between the pressure of the air and its density.

The boosting pressure which guarantees the desired load reduction resulted to be equal to 1.19 bar, a value very close to the atmospheric one which implies a decrease by 17.3% with respect to the boosting pressure required to run the engine at 48.7% of its maximum potential.

In spite of the linear relationship between the air density and the brake torque, the decrease of the boosting pressure is lower than the torque’s one, because of the distributed pressure drops along the whole intake manifold and the reduction of indicated and organic efficiencies: that losses contributes to the load reduction from the previous configuration to this one by 21.36%.

The performance generated by the engine boosted with this pressure are the following:

Volumetric efficiency	80.925%
Brake torque	81.762 Nm
Brake power	38.529 kW
BSFC	83.641 g/MJ
Indicated efficiency	33.8%

The phaenomenon observed in the cases of low and medium load reduction is now repeated: the lowering of the boosting pressure and the early intake valve closing are almost equivalent in terms of efficiency, since the specific fuel consumption generated by the first method is increased by 0.3% with respect to the one resulting from the second strategy, while the indicated efficiency is also decreased by 0.3%.

The reason of this almost total coincidence of the results is the same for all the load levels analysed so far: there is an almost perfect balancing between the lower pumping losses guaranteed by the EIVC strategy and the higher gross useful work ensured by the simple lowering of the boosting pressure.

As previously explained, the wastegate valve system is easier and cheaper to implement with respect to a variable valve train, but it can’t reduce the load below the value reached when the boosting pressure becomes equal to the atmospheric one; the two systems can be simultaneously used in order to avoid the pressure loss which affects the EIVC strategy because of the reduced lift which decreases the coefficient of discharge, since a reduction of the pressure of the air entering into the cylinder lowers its density and so the entrapped mass, allowing to impose a valve lift profile with a higher lift and a wider opening period.

A more accurate analysis of this phaenomenon can be made during the study of the open and close cycle efficiency of each control strategy derived so far, which requires the calculation of the integral of the positive and negative parts of each pressure-volume diagram.

5.4.5 Graphical comparison between the control strategies

The graphical comparison between the partializing strategies explained before will be now performed, following the same procedure as before: the plots of volumetric efficiency, indicated efficiency and specific fuel consumption will be shown, to visualize the order of magnitude of the differences between each method, whose causes will be clarified by the analysis of the pressure-volume diagram.

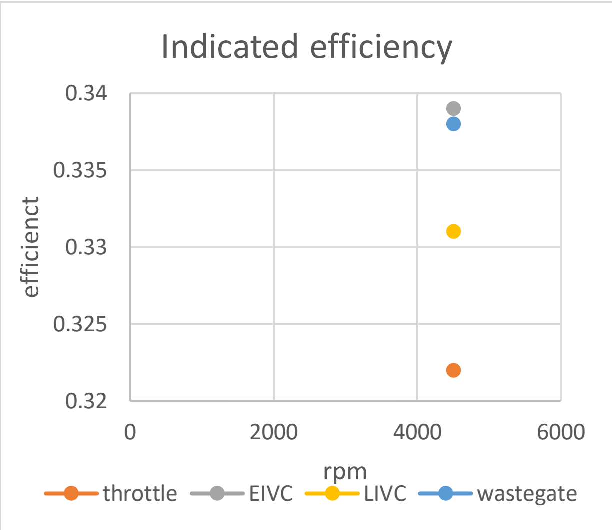

Figure 5.20 Indicated efficiency at 37.8%

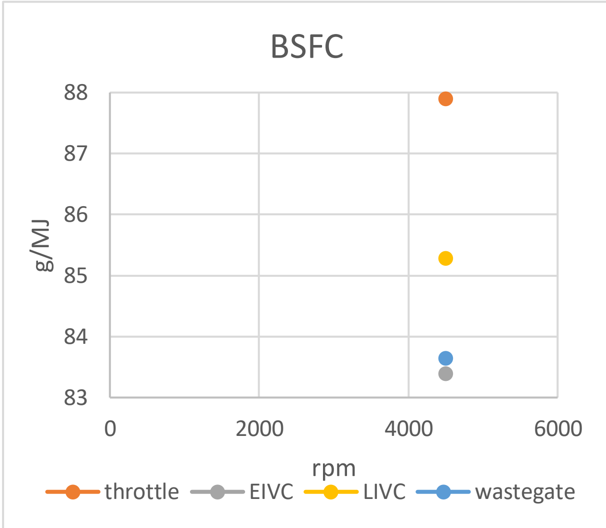

Figure 5.21 BSFC at 37.8%

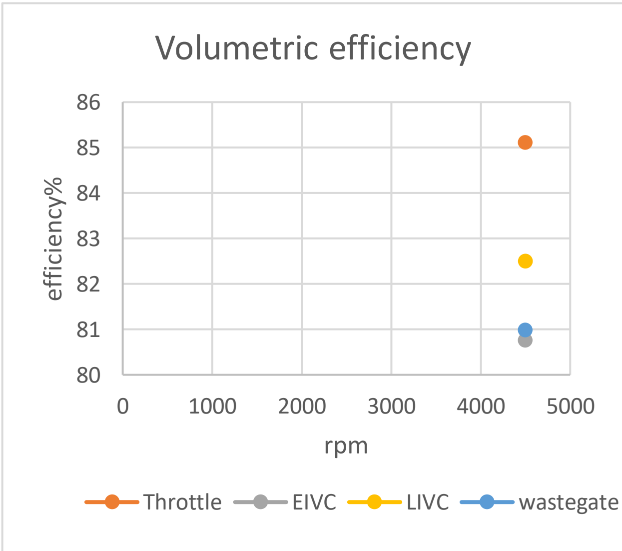

Figure 5.22 Volumetric efficiency at 37.8%

The graph of indicated efficiency and specific fuel consumption confirm what has been derived during the previous analysis: the throttling valve is the worst system of load reduction because of the huge pressure losses it causes, while the early intake valve closure and the lowering of the boosting pressure are substantially equivalent, giving the best improvement in terms of fuel saving in case of load reduction.

Even in this case the reduction of performance with respect of the full load in terms of BSFC is higher than the decrease in terms of indicated efficiency because of the lowering of the organic efficiency, which is now equal to 82.3% and so it has been reduced by 3.74% with respect to the previous case.

The EIVC strategy generates the lowest volumetric efficiency, given its lowest mass request to provide the target torque and power expressed by the other control strategies.

The pressure-volume diagrams provided by the control strategies previously analysed are now compared and discussed, to derive the main reasons of the differences between them; the phaenomena observed in the previous cases are expected to be seen even in this engine configuration, with a higher severity; given the almost perfect coincidence of the performance guaranteed by the lowering of the boosting pressure and the ones provided by the early intake valve closing, their diagrams are analysed separately from the other ones, to avoid confusion between the curves.

Even if the differences between the positive and negative areas are increased with respect to the case of medium load reduction, there is still a perfect balancing between the lower pumping losses which characterize the EIVC strategy and the higher useful work proper of the lowering of the boosting pressure: the decrease of the air pressure in the intake manifold reduces the capability of the air stream to spontaneously enter into the cylinder, so the piston has to spend a higher amount of work to draw it into the combustion chamber, but the exhaust gases are discharged at a lower pressure while the compression and expansion phases reach the same pressure as the EIVC strategy, so the energy deriving from the combustion is better exploited and so a higher gross indicated work is obtained.

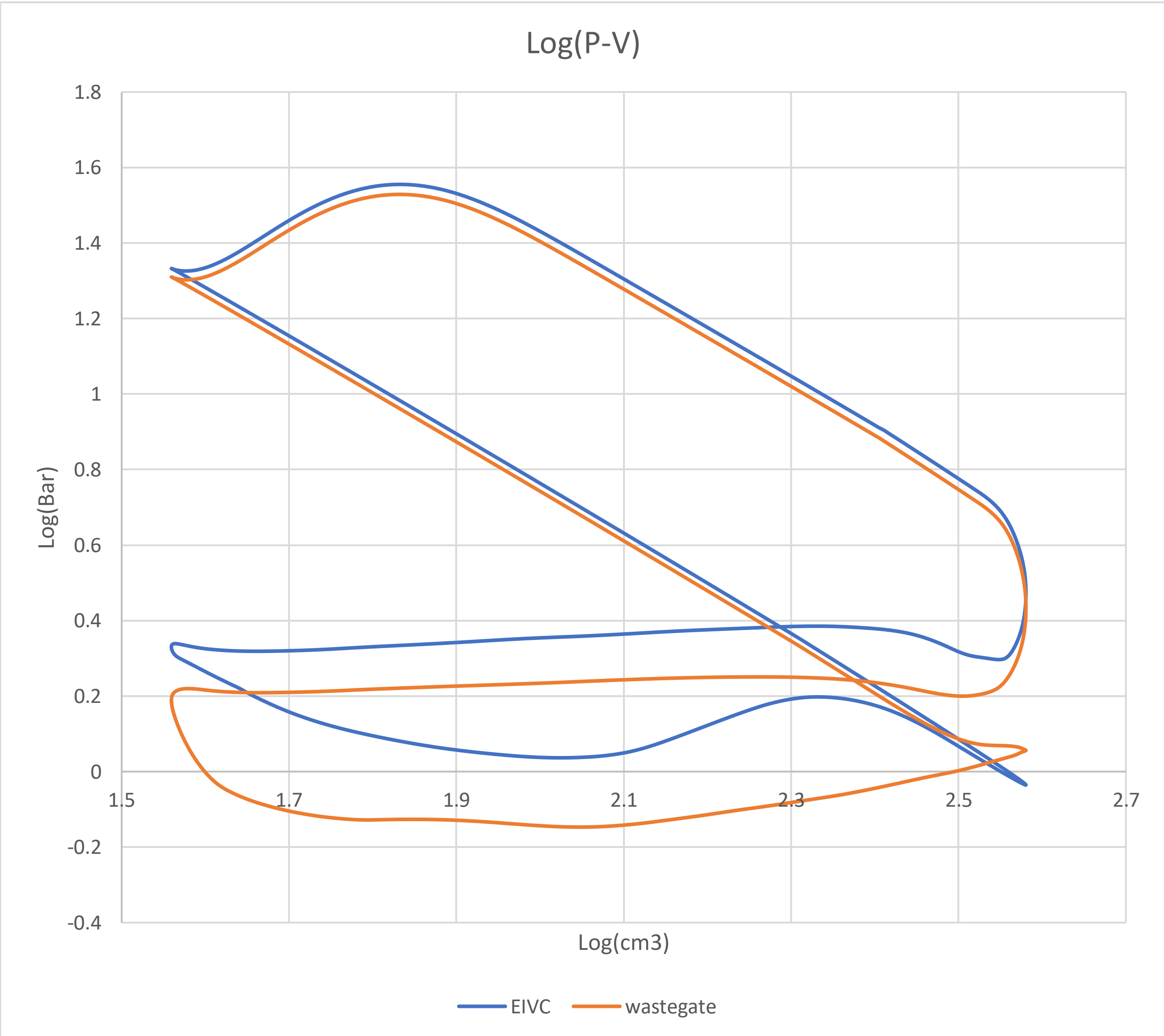

Figure 5.23 comparison between the P-V diagrams of EIVC and wastegate valve at 37.8%

The diagram resulting from the LIVC strategy confirms that the main disadvantage of this method occurs even at this level of load reduction: the pumping area is even lower than the one resulting from the EIVC strategy, but the process of expulsion of the air excess causes a reduction of the positive area of the diagram according to the process explained before, which means a reduction of the gross useful work generated by the cycle; the diagram is characterized by a clockwise curl at the end of the intake phase, which is due to the growing of the air pressure above the one at which the exhaust gases are discharged and, being positive, reduces the pumping area.

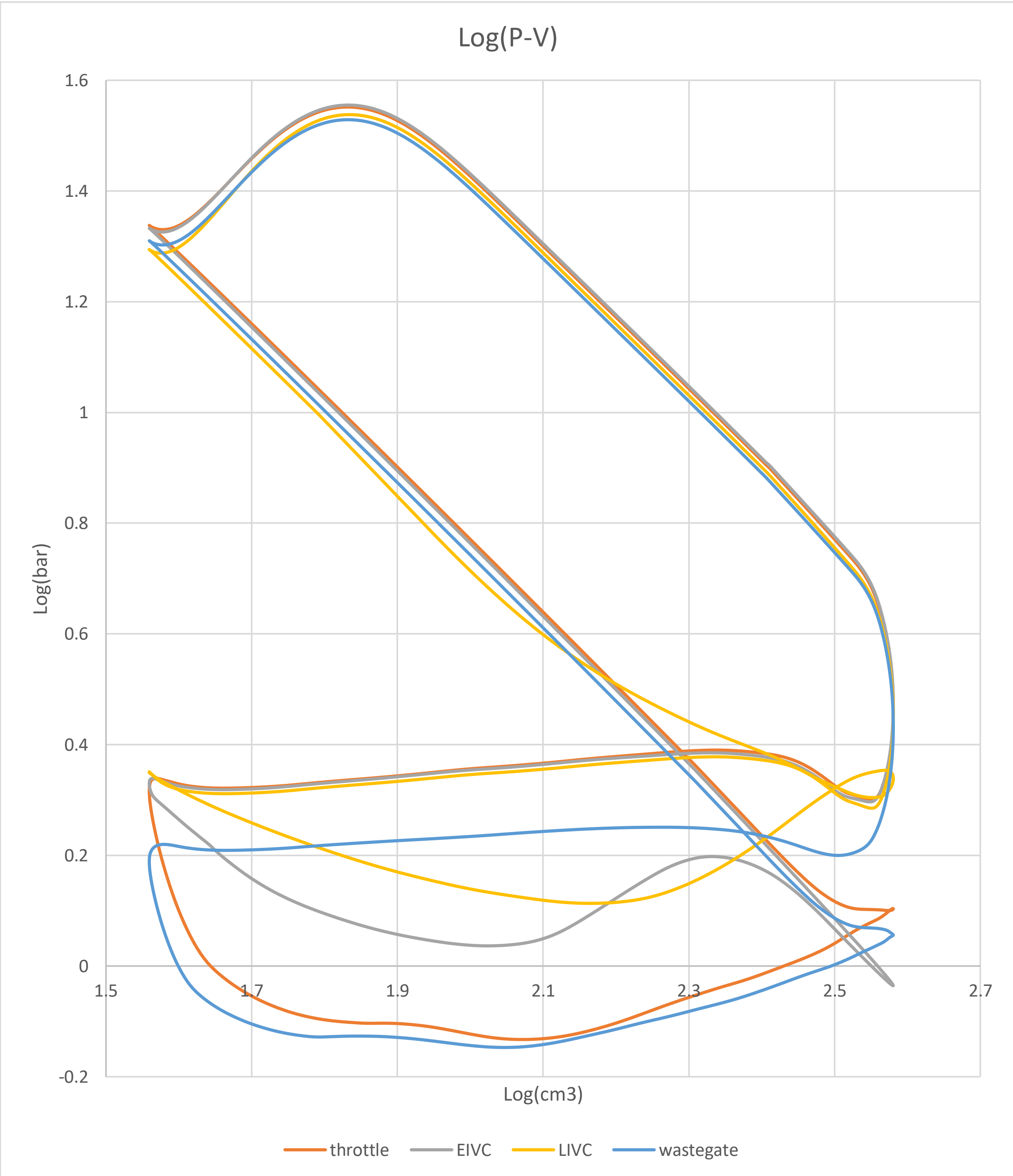

Figure 5.24 comparison between the P-V diagrams at 37% of full load

5.5 Global comparison of all the control strategies for all the load configurations

In this paragraph, the synthesis of all the analysis carried out so far will be performed: the volumetric efficiencies, the indicated ones and the brake specific fuel consumptions obtained by the implementation of all the methods of load reduction explained before in all the three load configurtion considered will be plotted on column diagrams, in order to definitively clarify the evolution of the main characteristics of each strategy at the progressive decreasing of the load and finally derive which is the best method.

In order to start the synthesis, all the operating points analysed before are now tabulated: the angles in correspondence of which there are the intake valve opening and closing are the absolute ones and the effect of the clearance is taken into account.

Load	Method	Throttle opening	P. boost [bar]	IVO	IVC	LIFT [mm]	ΔLift %	Duration	ΔDuration%
70%	**Throttle**	36.22 %	2.25	290°	565°	10.38	0	275°	0
	EIVC	100 %	2.25	290°	437°	7.71	-25.72	147°	-46.54
	LIVC	100 %	2.25	290	599.5°	10.38	0	309.5°	12.55
	wastegate	100 %	1.591	290°	565°	10.38	0	275°	0
48.7%	**Throttle**	28.89 %	2.25	290°	565°	10.38	0	275°	0
	EIVC	100 %	2.25	290°	423°	6.72	-35.26	133°	-51.63
	LIVC	100 %	2.25	290°	618°	10.38	0	328°	19.27
	wastegate	100 %	1.493	290°	565°	10.38	0	275°	0
38.7%	**Throttle**	27.12 %	2.25	290°	565°	10.38	0	275°	0
	EIVC	100 %	2.25	290°	418.5°	6.65	-35.93	128.5°	-53.27
	LIVC	100 %	2.25	290°	619.5°	10.38	0	329.5°	19.82
	wastegate	100 %	1.19	290°	565°	10.38	0	275°	0

The table shows how the opening angular period has been more and more reduced at the decreasing of the engine load in case of use of EIVC strategy or increased in case of use of LIVC one.

It can be noticed that, considering the EIVC strategy, the reduction from 70% to 48.7% of full load implies a decrease of maximum lift and opening angular period much higher than the one needed to furtherly partialize the engine to run it at 38.7%, and the same phaenomenon occurs considering the increase of the same quantities in case of LIVC strategy: the valve lift profiles which implement these partializing methods in the low load configuration have been derived by means of a slight modification of the shape on the descendant part on the profile together with a very little reduction of the opening period, while the profiles which allow to perform a low and medium load reduction have been reached after remarkable reductions of the lift duration, remembering that a simplified valve lift law has been used to implement the LIVC strategy whichever the condition (the profile has been made symmetric by means of the deletion of the minor peak which characterized the original one).

The reduction of the throttle valve's opening angle is also characterized by the phaenomenon explained before, but the reason is the non-linearity of the effective valve's area on the plate's angular position.

The column diagram of volumetric efficiency shows that the turbocharging ensures a lower reduction of volumetric efficiency with respect to the naturally aspirated engine: considering the load reduction by 30% obtained by means of the throttling valve, the volumetric efficiency is decreased by 24.9%, while a load decrease by 32% in the naturally aspirated engine provokes a drop of volumetric efficiency by 26.4%.

This is due to the increased density of the air which allows to entrap a higher mass of air in the same volume, moreover the pressure greater than the atmospheric one makes the air enter spontaneously in the cylinder.

The low difference between the four control methods is due to the fact that the load reductions and consequently the pressure losses are not as severe as the ones performed on the naturally aspirated engine, since the engine load remains always above the 30% of the maximum one.

The EIVC strategy provides the lowest volumetric efficiency whichever the load reduction, thanks to the lower air mass it requires to generate the same torque and power; its advantage with respect to the throttling valve increases at the decreasing of the load.

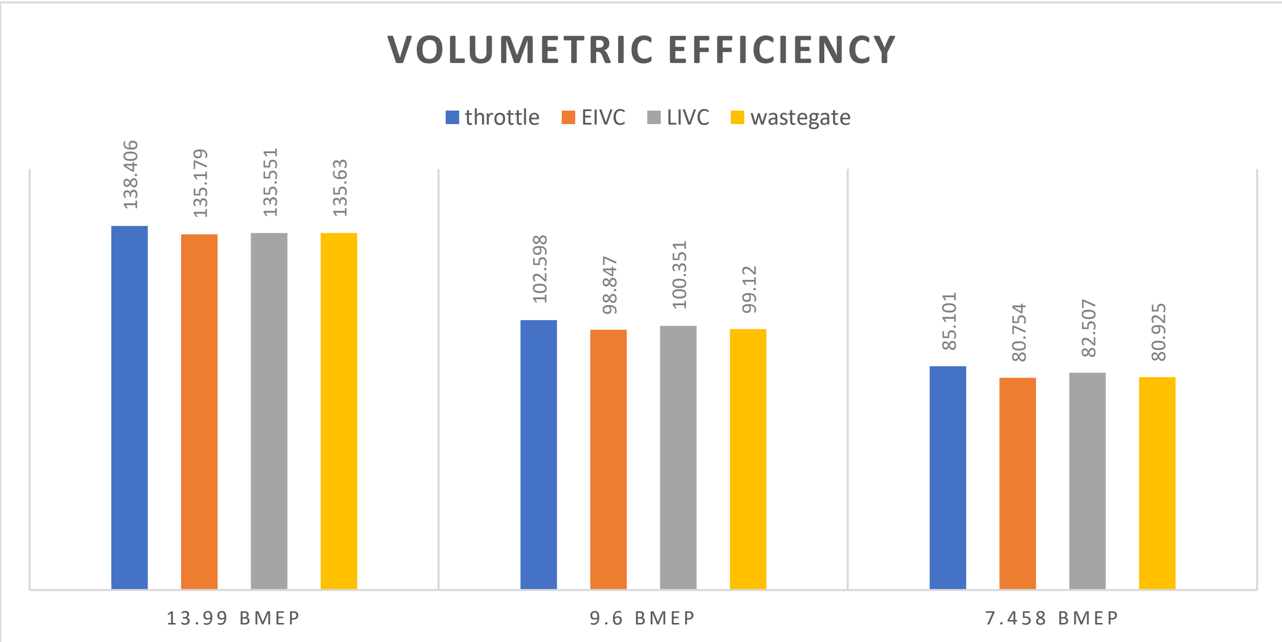

Figure 5.25 column diagram of volumetric efficiency

The column diagram about the indicated efficiency show that the early intake valve closing and the lowering of the boosting pressure are largely the best methods for engine partializing, since the indicated efficiencies they provide are almost equal and always higher than the ones given by the other control strategies; even in this case their advantage increases at the decreasing of the load: when the brake mean effective pressure is equal to 13.99, 9.6 and 7.458 bar, the EIVC strategy gives an indicated efficiency higher than the one provided by the throttled engine respectively by 2.34%, by 3.63% and by 5.3%.

The strongest the load reduction is, the lowest the benefits deriving from the use of the LIVC strategy are, because of the increase of the part of useful work which is spent to push the air excess back to the intake manifold: in the first load configuration the indicated efficiency it generates is lower than the one given by the EIVC strategy by 0.28%, in the second one it is lower by 1.48% while in the third one it's lower by 2.36%

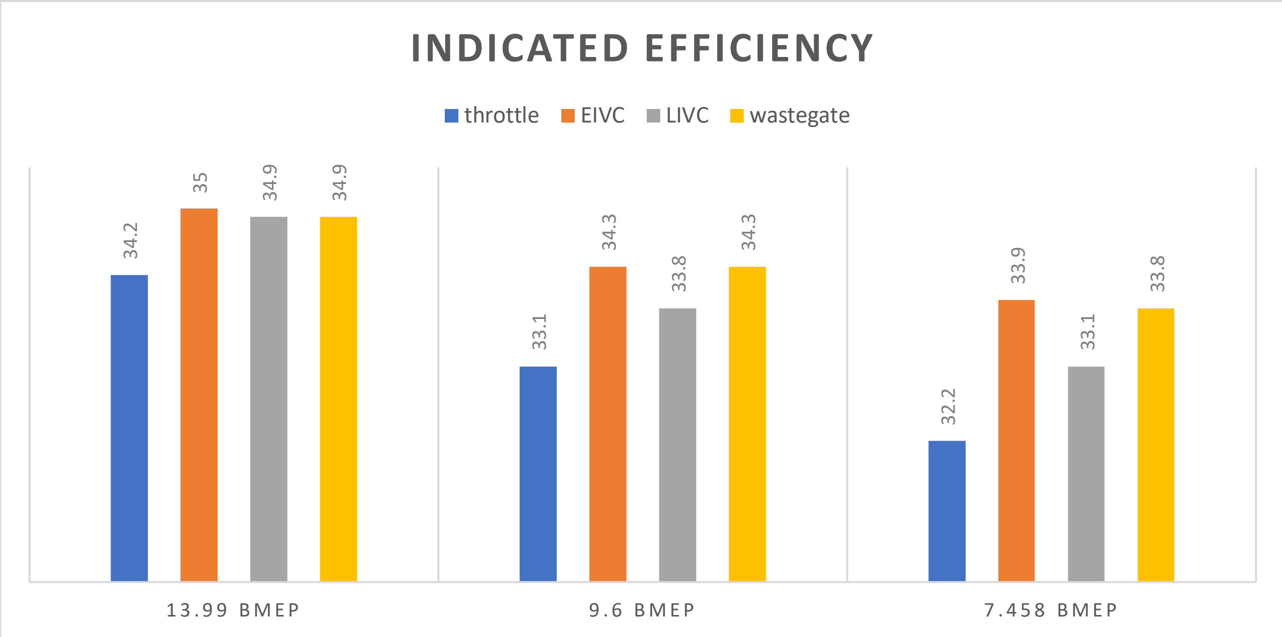

Figure 5.26 column diagram of indicated efficiency

The two best control strategies as such a capability to avoid pressure drops that the maximum decrease of indicated efficiency between the first and the last engine configuration is by 3.24%.

The column diagram of the specific fuel consumption gives the same information provided by the previous one being its dual, because of the relationship of inverse proportionality between the indicated efficiency and the specific fuel consumption: the EIVC strategy and the lowering of the boosting pressure are the partializing methods which guarantee the highest fuel saving whichever the engine load, in particular at the low ones: the early intake valve closing improves the BSFC by 2.47%, by 3.65 and by 5.13% when the load reduction is low, medium and severe respectively; the slight difference of these percentages with respect to the ones derived during the discussion of the column diagram about the indicated efficiency is due to the division by the organic efficiency which is lower than one, as explained in the previous chapter.

The progressive increase of BSFC and decrease of indicated efficiency are obviously due to the growing of the pumping work needed to draw the air into the cylinder.

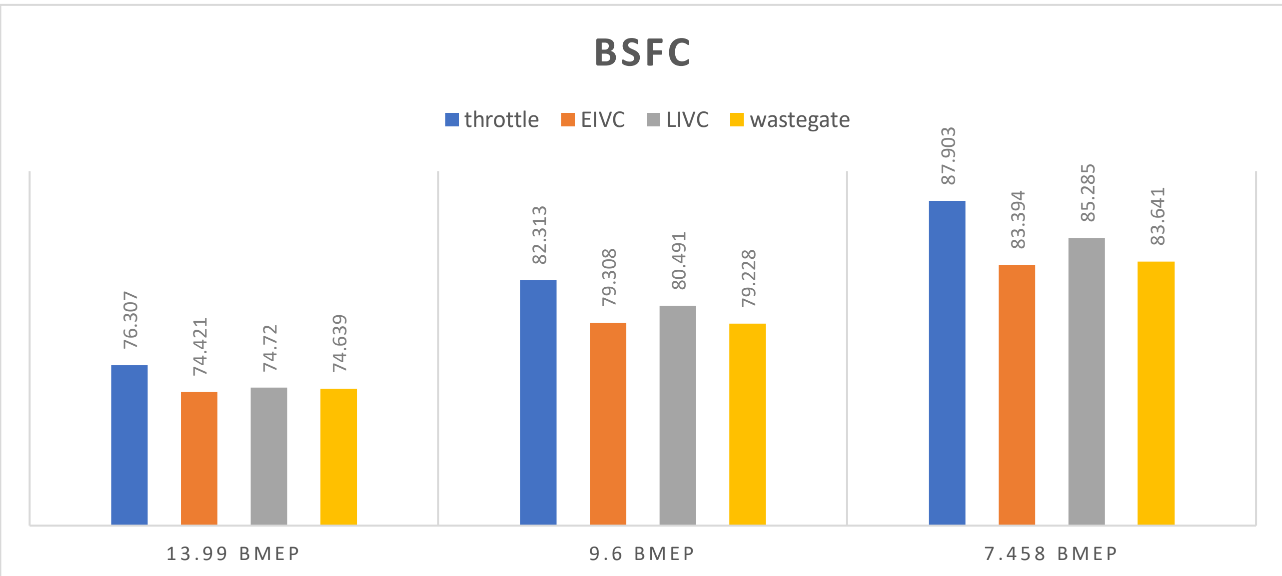

Figure 5.27 column diagram of BSFC

5.6 Open cycle efficiency analysis

The global comparison between the methods of load reduction is now completed by the quantitative analysis of the pressure volume diagram expressed in a linear scale: the calculation of the clockwise and anti-clockwise areas' integrals allows to derive the open cycle efficiency, which quantifies how good the engine is to perform the gas exchange process, and the close cycle one, which is related to the part of the cycle during which both the intake and the exhaust valve are closed.

The formulas for the calculation of open and close cycle efficiencies have been already explained in the chapter about the naturally aspirated engine, remembering that they're based on the indicated mean effective pressure of the positive and negative parts of the pressure-volume diagram.

The table below contains the data about the open cycle analysis, that is the Net indicated mean effective pressure, which corresponds to the indicated mean effective pressure properly named, the Gross one and the Pumping one; the last column contains the reductions of PMEP with respect to the throttled engine guaranteed by the other control strategies in the three load configurations.

BMEP (bar)		NIMEP (bar)	GIMEP (bar)	PMEP (bar)	$\eta_{open}\%$	PMEP REDUCTION %
19.721	**Full load**	21.49	22.32	-0.8102	96.37	-
13.99	**Throttle**	15.66	16.60	-0.9385	94.35	-
	EIVC	15.68	16.28	-0.6076	96.27	35.26
	LIVC	15.66	16.21	-0.5468	96.63	41.73
	wastegate	15.69	16.37	-0.6800	95.85	27.54
9.60	**Throttle**	11.24	12.14	-0.9133	92.48	-
	EIVC	11.23	11.70	-0.4668	96.01	48.89
	LIVC	11.23	11.59	-0.3551	96.94	61.12
	Wastegate	11.27	11.90	-0.6415	94.61	29.76
7.458	**Throttle**	9.064	9.95	-0.8862	91.09	-
	EIVC	9.052	9.48	-0.4288	95.48	51.61
	LIVC	9.047	9.49	-0.4442	95.32	49.87
	wastegate	9.058	9.68	-0.6180	93.61	30.26

Differently from the case of the naturally aspirated engine, the EIVC strategy is characterized by a higher open cycle efficiency with respect to the LIVC one only when the load reduction becomes severe, while the second method guarantees the best efficiency in the other engine configurations.

The reason of this phaenomenon is that the characteristics of the turbocharger installed on the engine and the delayed closure of the valve make the air pressure be higher than the one at which the exhaust gases are discharged when the piston approaches to the bottom dead centre, with a successive pressure decrease at the beginning of the compression stroke: this sequence of transformations is represented by a clockwise curl clearly visible in the P-V diagram and allows the air to exchange a positive work with the piston, lowering the anti-clockwise area of the diagram itself and so increasing the indicated efficiency.

When the engine is simulated at 50% of full load the phaenomenon is so remarkable that the pumping area resulting from the implementation of the LIVC strategy is lower than the one given by the full load configuration with such a difference that the open cycle efficiency becomes higher by 0.6% than the one obtained at W.O.T condition and by 0.96% than the one provided by the EIVC strategy, despite of the lower positive work generated; when the load is reduced by 62.2%, the loss of useful work due to the wide air excess to be drawn outside the cylinder overcomes the advantage given by the lower dimension of the pumping area, making the LIVC strategy less efficient by 0.17% than the EIVC one.

The wastegate valve opening provides a lower open cycle efficiency with respect to the EIVC strategy whichever the load reduction, the gap enlarges at the decreasing of the BMEP.

5.7 Close cycle efficiency analysis

The efficiency of the compression, combustion and expansion processes will be now quantified thanks to the calculation of the gross indicated power, derived from the calculation of the integral of the positive part of the pressure-volume diagram, and its division by the fuel power, according to the same procedure followed in the chapter about the naturally aspirated engine.

The fuel adopted to run the engine is the same used to simulate the naturally aspirated one and its lower heating value is equal to 43000 kJ/kg, remembering that:

$$fuel\ power = \dot{m}_c H_i$$

The fuel flow rate is derived by the multiplication between the brake specific fuel consumption and the brake power, whose values are obtained at the end of the simulation performed by Gasdyn, which also provides the data about the P-V diagram whose integration is solved by means of the same Matlab code as before.

The data about the gross indicated power, the fuel power and the closed cycle efficiency are listed below:

Bmep (bar)		GROSS IP (kW)	Fuel Power (kW)	η_{closed}%
19.721	**Full load**	115.31	315.39	36.56
13.99	**Throttle**	85.75	236.84	36.20
	EIVC	84.12	231.34	36.36
	LIVC	83.72	231.95	36.09
	wastegate	84.56	232.08	36.44
9.60	**Throttle**	62.74	175.57	35.74
	EIVC	60.45	169.15	35.74
	LIVC	59.88	171.71	34.87
	wastegate	61.50	169.61	36.26
7.458	**Throttle**	51.40	145.63	35.29
	EIVC	48.98	138.19	35.44
	LIVC	49.03	141.18	34.73
	wastegate	49.99	138.57	36.08

The first important information given by the table is that the boosting pressure reduction gives always the best closed cycle efficiency, and this quantity is shows a low reduction at the decreasing of the load: when the BMEP is equal to 7.458 bar, the engine partialized with the LIVC strategy provides the worst closed cycle efficiency, lower by 5% with respect to the one resulting from the engine run at W.O.T. condition.

As it occurred in the case of the naturally aspirated engine, the EIVC strategy provides the lowest fuel power whatever the engine load, since the lower pressure drops it generates reduce the mass of air needed to reach the desired power output, and consequently the required mass of fuel per cycle (it must be underlined that the imposed air to fuel ratio provides at this regime a slightly rich mixture, being it equal to 14.42).

The most important difference with respect to the naturally aspirated engine is that now the EIVC strategy guarantees a slight efficiency improvement with respect to the throttling valve: the increased pressure in the combustion chamber due to the turbocharging causes an increase of the combustion's velocity (directly proportional to temperature and pressure), which overcomes the effect of the lower air velocity which reduces the burning rate, as explained in [8].

The lowering of the boosting pressure ensures the best closed cycle efficiency whichever the load, because of the highest useful work it can provide, which counterbalance the pumping losses it generates that are higher than the ones resulting from EIVC and LIVC strategies, as it can be seen in all the comparisons of the P-V diagrams.

5.8 Summary

The aim of this chapter has been to analyse the behaviour of a 1400 c.c. four-cylinders turbocharged engine controlled by means of the throttling valve, the early and late intake valve closing and the lowering of the boosting pressure caused by the opening of the wastegate valve, in order to study the effect of the implementation of each strategy on the engine efficiency on three levels of load reduction, that is 70%, 48.7% and 38.7% of full load.

The overview contains a brief description of the engine under analysis and the explanation of the methodology according to which the work is carried out, while the first paragraph reports the description of the engine's performances at full load, with a focus on the plots about the brake torque and the brake power on the full speed range needed to choose the regime at which the whole analysis is carried out, that is the maximum power one (4500 rpm).

The paragraphs from 5.2 to 5.4 are related to the simulation of the engine at the three levels of load explained before, each of them is divided into five sub-paragraphs dedicated to the four methods of load control and to the final comparison between them: the angles of aperture of the throttling valve and the boosting pressures needed to reach each load configuration are explained and the valve lift profiles which implement the EIVC and the LIVC strategies are derived, then the graphs of volumetric efficiency, indicated efficiency and brake specific fuel consumption are compared to better understand the order of magnitude of the differences of performance between the partializing system, that are explained by the analysis of the logarithmic pressure-volume diagram.

All the results obtained from the second to the fourth paragraph are summarized in the fifth one, in which the data are collected on column diagrams which compare he volumetric and indicated efficiencies and the BSFC provided by each control strategy in all the three analysed load configurations.

The main result of the whole analysis is the same obtained in the previous chapter: the early intake valve closing is the best method of load reduction, since it generates lower pumping losses with respect to the throttling valve and the simple reduction of the boosting pressure, while it avoids the loss of useful work due to the expulsion of the air excess back to the intake manifold which affects the LIVC strategy.

The lowering of the boosting pressure provides however a performance which is extremely close to the one given by the EIVC strategy when the load reduction is medium or high, since the wider positive area on the P-V diagram counterbalances the higher pumping losses.

The paragraphs 5.6 and 5.7 are related to the open and closed cycle efficiency analysis, whose main result is the explanation of the reason why the indicated efficiencies and the specific fuel consumptions provided by the EIVC strategy and the lowering of the boosting pressure are so similar: the first method is characterized by a higher open cycle efficiency but also by a lower closed cycle efficiency with respect to the second one and the difference of open cycle efficiency is almost equal to the one of closed cycle efficiency, so they are almost perfectly balanced.

Despite of this equivalence of efficiency, the EIVC strategy has to be preferred to the wastegate valve opening since it can reduce the engine load below a limit that the second method can't overcome, since the boosting pressure can't be reduced under the atmospheric value and when that limit is reached, the engine is run at 36% of full load; the two strategies can be used together, since the lower air pressure allows to use a valve lift profile with a higher maximum lift which reduces the pressure drops.

In conclusion, the EIVC strategy largely demonstrated to be the best method of load reduction even in the case of the turbocharged engine, confirming that its implementation on the new generation engines is necessary to reduce the fuel consumption at partial load, that is the condition at which the engine is mostly used.

Chapter 6

Conclusion

This chapter concludes the whole thesis and is aimed to the final synthesis and discussion of all the analysis carried out so far.

As it has been explained in the introduction, the aim of this thesis is to find a system which allows to reduce the engine load avoiding the use of the throttling valve, which causes a severe concentrated pressure loss when its angle of aperture decreases below 30° because it forces the air to pass through a restricted section characterized by an extremely low discharge coefficient.

The importance of the reduction of the fuel consumption is extremely high nowadays because of the growing attention paid by the society to the problems of oil scarcity and global warming: since the vehicle are always used at low or medium load condition, the new generation engine must be equipped by a system which makes them able to reduce the load in an efficient way.

The alternative methods which allow to reduce the quantity of air entering in the cylinder consist in the early intake valve closure, which stops the air sucking before the intake stroke is completed, and in the delayed one, which allows the piston to push the air excess back to the intake manifold at the beginning of the compression stroke; the turbocharged engines can be controlled also by means of the lowering of the boosting pressure, which implements a reduction of the mean effective pressure avoiding the use of the throttling valve and the variable valve train.

The use of the EIVC or the LIVC improves the efficiency of the gas exchange process thanks to the reduction of the pumping losses they guarantee but also improves the thermal efficiency of the engine, since the anticipated or delayed closure of the intake valve make the effective intake stroke be lower than the compression stroke, so the Otto cycle normally realised by the engine is transformed into a Miller cycle, which is characterized by a higher thermal efficiency with respect to the first one, thanks to the better exploiting of the energy exchanged by the burning gases which can undergo to a higher expansion and so can exchange a higher amount of work with the piston.

In order to demonstrate the utility of these control strategies, they have been implemented on two different engines, a four-cylinders 2000 c.c. naturally aspirated engine and a 1400 c.c. turbocharged one, in order to reduce their load up to three target levels; the results provided by each method have been compared to understand which one guarantees the lowest specific fuel consumption and the highest indicated efficiency.

The analysis carried out on the naturally aspirated engines demonstrates that the EIVC strategy to reduce the engine load in the most efficient way, since it guarantees the highest indicated efficiency and the lowest specific fuel consumption whatever the load and also the highest open cycle efficiency, thanks to its capability to reduce the pressure drops without losing useful work.

The simulations performed on the turbocharged engine lead to the same result obtained on the other one, since the early intake valve closing provides the highest indicated and open cycle efficiency and the lowest specific fuel consumption; the reduction of the boosting pressure gives almost the same output as the EIVC strategy, and its implementation is cheaper than the variable valve train’s one, but it can’t reduce the load below the limit constituted by the load reached when the boosting pressure becomes equal to the atmospheric one and the engine returns to be naturally aspirated.

The late intake valve closing resulted to be worse than the early one in all the analysed configurations on both the engines, but it has been demonstrated in [2] that it reduces the knock at low and medium load.

To conclude, the main result obtained at the end of the work is that the EIVC strategy is the best way to reduce the load on an internal combustion engine, so its implementation through a variable valve action system is absolutely needed on the new generation engines to ensure the reduction of the fuel consumption.

This analysis can be further developed by means of the simulation of the engine run with the simultaneous use of the EIVC strategy and the lowering of the boosting pressure, in order to join the effects of the lower pumping losses produced by the first method and the higher positive work given by the second one.

The study can be completed by the analysis of the influence of the spark advance on the fuel consumption, since an advanced start of ignition improves the stability of the combustion when the turbulence of the mixture in the combustion chamber is low (this condition is typical of the EIVC strategy).

APPENDIX

Turbocharged engine's combustion parameters:

Engine speed [rpm]	A/F ratio [-]	Θ_{50} [deg. aTDC]	Θ_{10} [deg.]	Efficiency parameter [-]	Shape factor [-]
1000	14.54	18.8	16.1	6.9	4.1
1250	14.54	18.8	16.1	6.9	4.1
1500	14.54	34.4	20.2	6.9	3
1750	14.54	32.7	20.3	6.9	2.7
2000	14.54	34	20.7	6.9	2.9
2250	14.54	34	20.7	6.9	2.9
2500	14.54	30	22.5	6.9	1.9
2750	14.54	30	22.5	6.9	1.9
3000	14.54	27.8	20.3	6.9	3.3
3250	14.54	27.8	20.3	6.9	3.3
3500	14.54	25.6	19.7	6.9	3.2
3750	14.54	25.6	19.7	6.9	3.2
4000	14.55	25.3	20.7	6.9	2.9
4250	14.55	25.3	20.7	6.9	2.9
4500	14.42	24.9	21.5	6.9	2.6
4750	14.42	24.9	21.5	6.9	2.6
5000	14.15	23.8	20	6.9	3
5250	14.15	23.8	20	6.9	3
5500	13.8	21.8	20	6.9	2.4
5750	13.8	21.8	20	6.9	2.4
6000	13.53	22	20.3	6.9	2.1

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