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Matlab-assisted thermodynamic analysis

Part 2: Two-zone model

Introduction

Engine modelling activities, at least in recent decades, have largely been concentrated in the direction of designing better performing engines with lower emissions. In this regard, modelling of engine combustion processes assumes importance.

The various engine combustion models that have been developed to date may be grouped into the following categories: Zero dimensional, Quasi-dimensional, Multi-dimensional. In the above classification, although the level of detail and proximity to physical reality increases with the model dimensions, so does the complexity of creating and using those models.

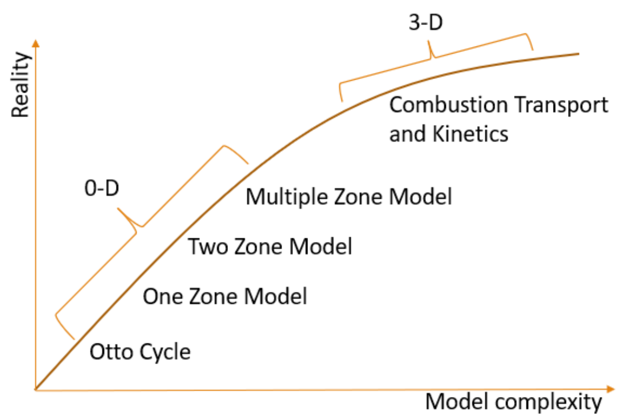


Fig. 1 Model complexity vs Physical Reality

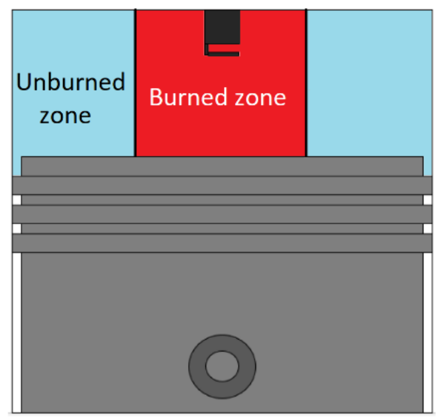


Fig. 2 A Two-zone model representation

Zero dimensional models (no flow field dimensions) are the simplest and most suitable to observe the effects of empirical variations in the engine operating parameters on overall heat release rates/cylinder pressure schedules.

Zero dimensional models are further sub-divided into:

1. Single zone models
2. Two zone models
3. Multi-zone models

In single zone models, the working fluid in the engine is assumed to be a thermodynamic system, which undergoes energy and/or mass exchange with the surroundings and the energy released during the combustion process is obtained by applying the first law of thermodynamics to the system.

In two zone models, the working fluid is assumed consisting of two zones (see Figure 2), an unburned zone and a burned zone. These zones are two distinct thermodynamic systems with energy and mass interactions between themselves and their common surroundings, the cylinder walls. The mass-burning rate (or the cylinder pressure), as a function of crank angle, is then numerically computed by solving the simplified equations resulting from applying the first law to the two zones.

1 Model assumptions

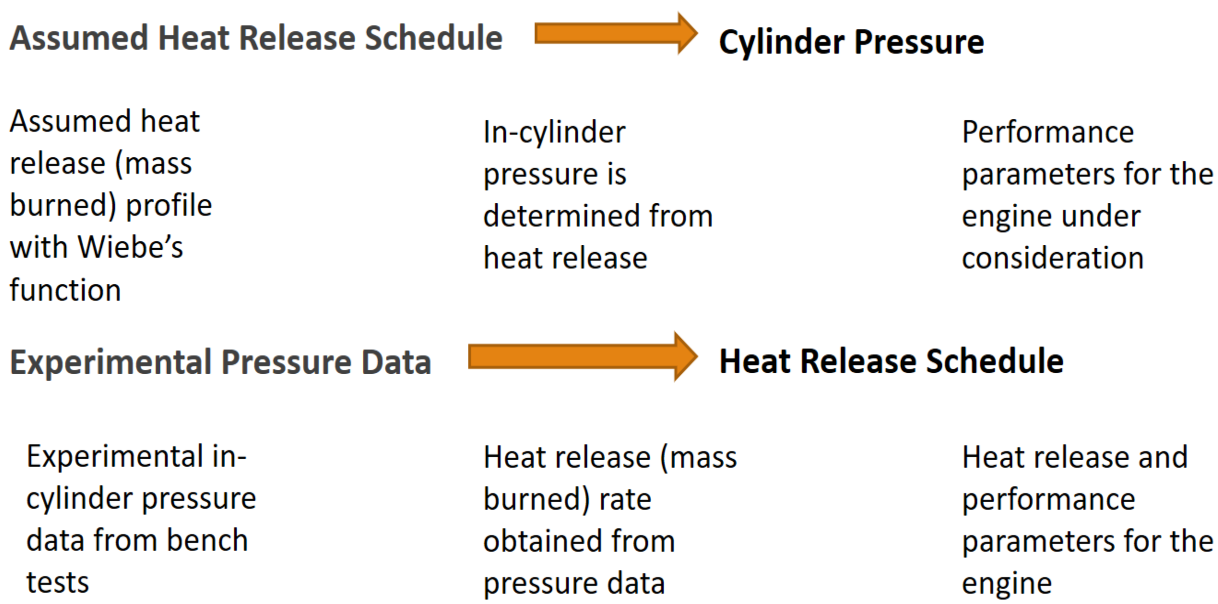


Fig. 3 Directions of the Two zone model usage

When referring to the nature of single zone and two zone models, these models have been traditionally used in two different directions (see Figure 3):

1. Predict the in-cylinder pressure as a function of crank angle from an assumed energy release or mass burned profile (as a function of crank angle).

2. Determine the energy release/mass burning rate as a function of crank angle from experimentally obtained in-cylinder pressure data.

Multi-zone models take this form of analysis one step further by considering energy and mass balances over several zones, thus obtaining results that are closer to reality.

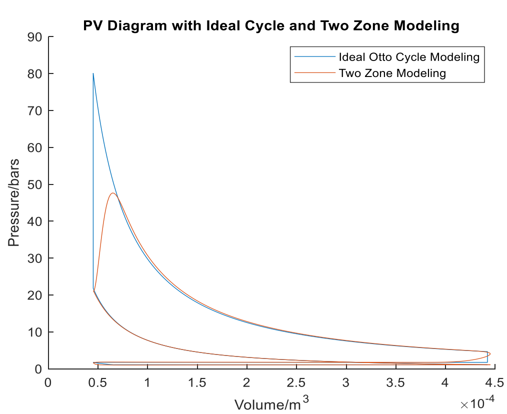


Fig. 4 Comparison: 2 zone model vs Ideal Cycle

A single-zone heat release model was developed in MATLAB. The single-zone model was then split into a two-zone model using burned and unburned volume fractions with an assumed mass burning rate and other suitable simplifications (see Table II) to predict engine performance parameters. The impact of turbocharger on global parameters was studies and the elevated burned zone temperature provided by the two-zone model was used to predict knock behaviour.

Table II. Assumptions of the Two-Zone Model

Dimensions	- 0-D: Time
Combustion	- Flame front propagation - Wiebe law
Two Zones	- Fresh gases/Burnt gases
Mixture	- Homogeneous and stoichiometric in each zone - No change in mixture composition
Pressure	- Homogeneous inside the cylinder
Fresh gases/Burnt gases	- First law for open systems - Ideal gas law - Mass conservation law - Isentropic ducted flow
Heat loss	- A semi-empirical model (Woschini)

2 Main results

The basic idea behind the two-zone model is to utilise the conservation of mass and energy (1st law of Thermodynamics) and the ideal gas equations to obtain an "apparent" rate of heat release curve or mass burned fraction curve, whose accuracy is limited by the assumptions of the model.

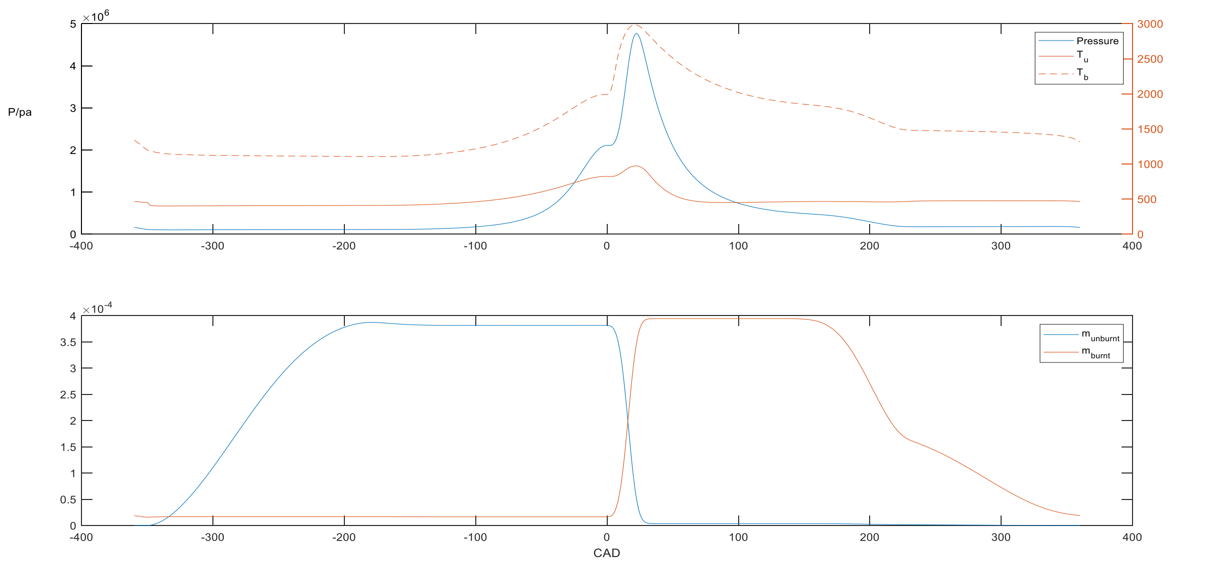


Fig. 5 Evolution of in-cylinder Pressure, Temperatures and Masses computed by a 2-zone model through a complete engine cycle

The results of the two-zone model can be compared with that of an ideal gas cycle model (see Figure 4) to observe the expected decrease in performance parameter values, since a two-zone model relatively models the reality better (see Tables 3 and 4).

Table III. Results from an ideal cycle model

Parameter	Value
Wcycle [J]	444.191
Pmax [bars]	80.062
Tmax [K]	3318.889
IMEP [bars	11.18
mFuel	0.00239
η [%]	43.2
Pout [kW]	14.806
Spec. Pout [kW]	37.285

Table IV. Results from a 2-zone model

Parameter	Value
Wcycle [J]	396.798
Pmax [bars]	47.636
Tmax [K]	2981.086
IMEP [bars	9.930
mFuel	0.0254
η [%]	36.4
Pout [kW]	13.226
Spec. Pout [kW]	33.100

One of the most important aspect of modeling is to validate the model for its correctness to expect reliable conclusions from the model, which is usually done with some experimental results. In the course of the project, the only source of validation was to compare the trends of the results with what one would expect from intuition/understanding of the physics of the system. One of the main feature of the two-zone model is the curve of the mass burnt fraction which is used to find the in-cylinder pressure and thus, the performance parameters. The trend of the curve was compared to that obtained from experimental results (see figure 6) to assume suitable validity of the model to move on with further analyses.

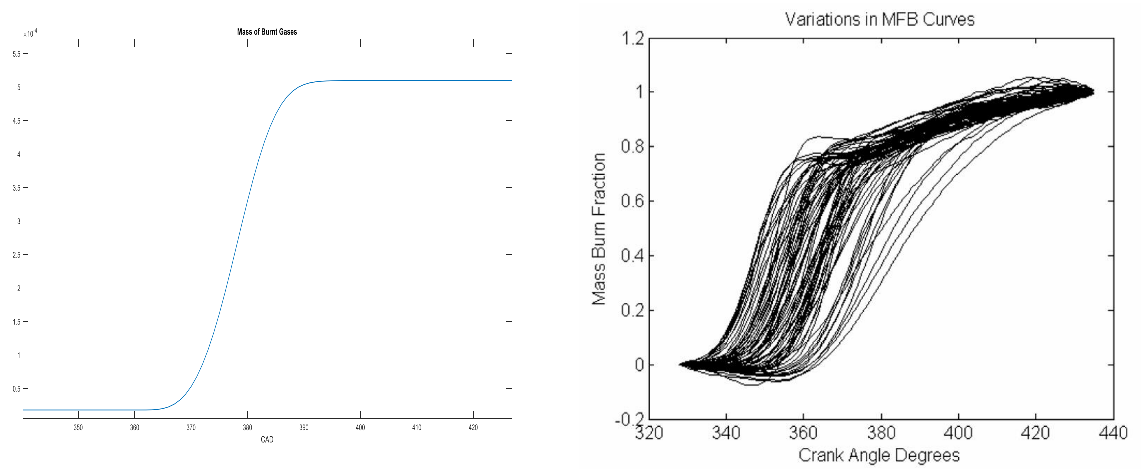


Fig. 6 Validation of the Wiebe Function [1]

3 Influence of design parameters on SI IC-engine performance

3.1 Influence of compression ratio on engine performance

Figure 7 shows that the fuel conversion efficiency increases continuously with compression ratio.

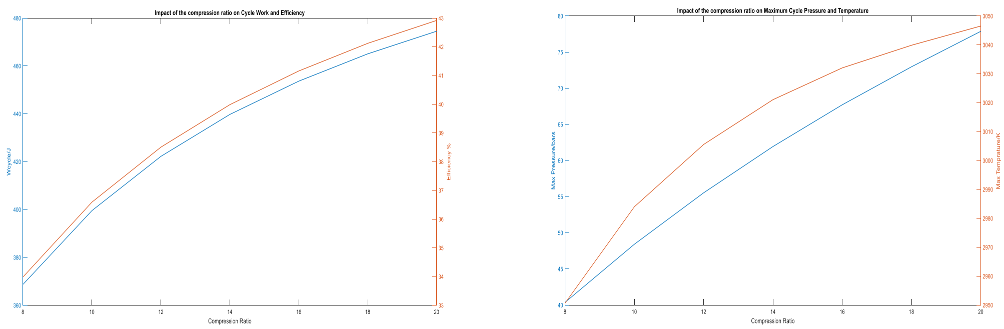


Fig. 7 Engine performance parameters variation with compression ratio

However, in actual engines, other processes which influence engine performance and efficiency vary with changes in compression ratio and start decreasing at very high compression ratios (see Figure 8).

The difference between the calculated and indicated efficiency represents losses due to combustion rate and stability, valve timing, non-constant volume combustion, and direct heat and friction losses [2].

Various simplifications in the model fail to capture these details and thus the behavior. Although, for the purpose of the study, this is not required since nominal values for compression ratios ranges from 9 to 11 for most vehicles, and around that range, the trend is as seen in the experimental results, therefore justifying the model fidelity.

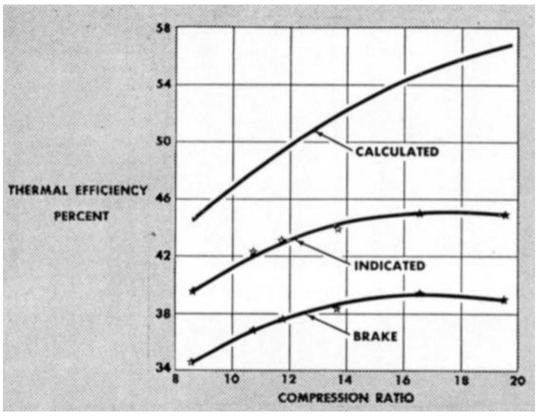


Fig. 8 Experimental results on effect of compression ratios in an SI Engine, Carin *et al*

3.2 Influence of combustion duration and spark advance on engine performance

In Otto's ideal thermodynamic cycle, the combustion process is instantaneous, making the in-cylinder pressure rise infinitely fast when the piston is at top dead centre. This instantaneous pressure increase maximises the cycle work, and so the efficiency. In real spark ignition engines, things are different.

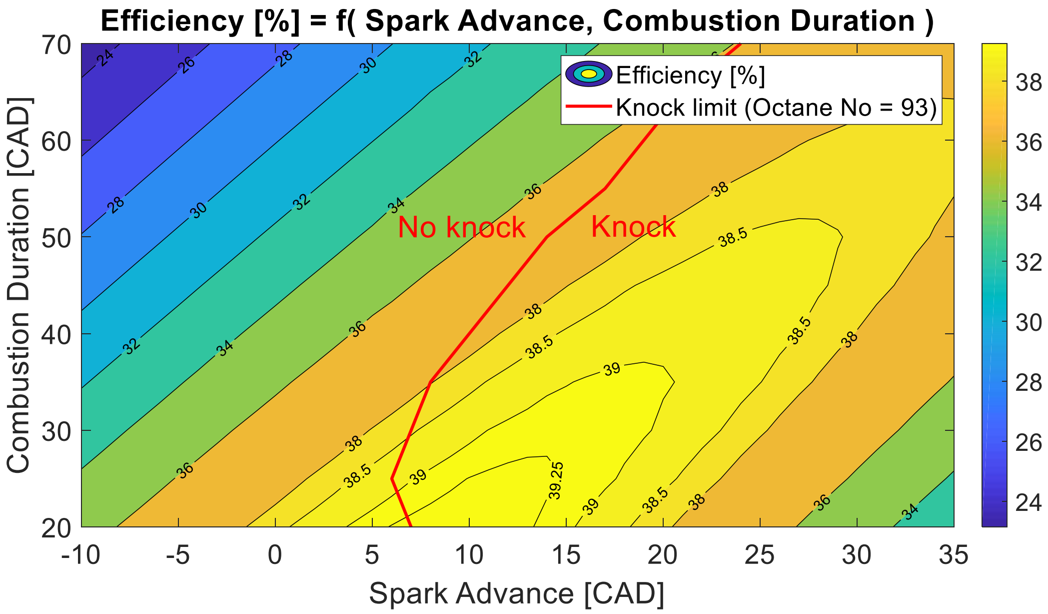
3.2.1 Combustion duration

In a real spark ignition engine, the turbulent flame speed is not infinite but usually ranges from 20 to 40 m/s, so combustion duration can go from 1 to 2 milliseconds if the cylinder has an 80 mm bore and is fitted with a centred spark plug. These time durations will correspond to angular durations ranging from around 10 to 70 CAD, depending on the engine speed.

By increasing the combustion duration, the real cycle gets more and more away from the Otto cycle (giving the maximum cycle work), so is the efficiency is decreased.

3.2.2 Spark advance

In a real spark ignition engine, combustion takes time due to the finite turbulent flame speed, which usually ranges from 20 to 40 m/s, making the idea of burning the complete air-fuel mixture at TDC unrealistic. Consequently, the combustion has to start before the moment when piston reaches TDC, hence the ignition advance value that has to be found by the calibrator of the engine.



For a given combustion duration, if we progressively increase the spark advance starting from -10°, we can see the efficiency increasing until it reaches its maximum (which is usually beyond the knock limit at full load). From this point, more spark advance will increase the negative work of the compression stroke more than it will increase the positive work of the expansion stroke (which can even be reduced). Therefore, the overall cycle work is decreasing, and so does the efficiency.

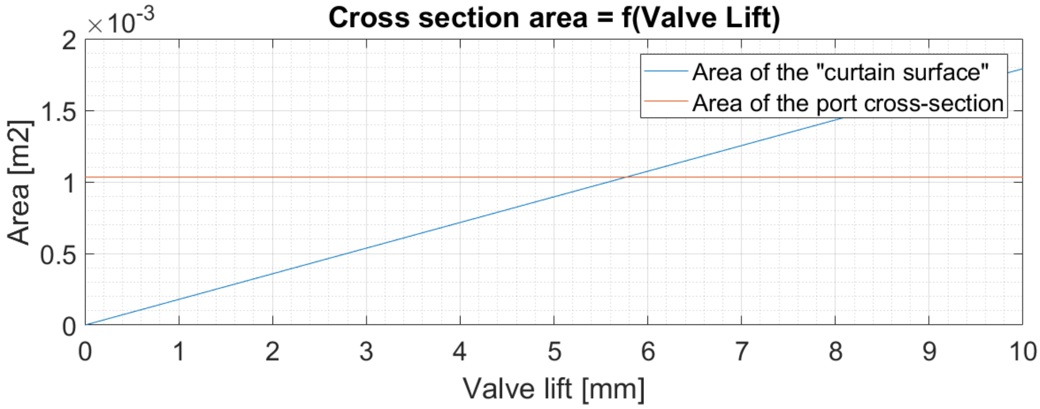
3.2.3 Conclusion about combustion duration and spark advance

Although we noticed that combustion duration should be kept as low as possible, there is no direct way to control the flame speed, as it depends on temperature (mainly linked to the load, but also to the spark advance) and turbulence level (mainly linked to the engine speed).

On the contrary, spark advance is directly controllable. When there is no limitation due to knock (at part load or when using a fuel with a high octane number) or to maximum in-cylinder pressure, the optimum value has to be found.

3.3 Influence of valve lift and valve timing on engine performance

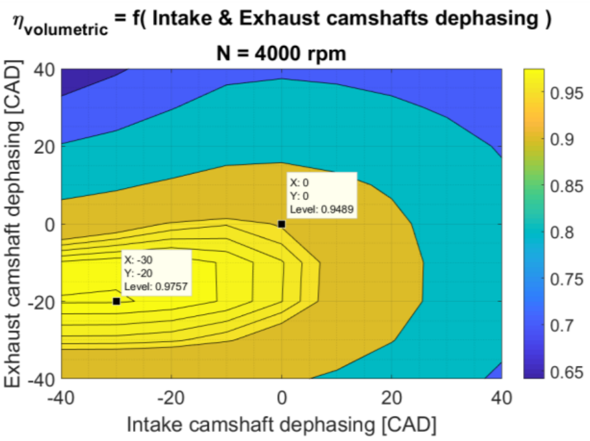
3.3.1 Valve lift

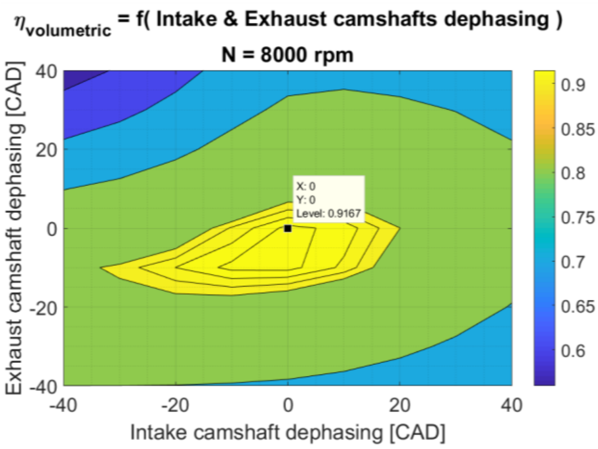


From a certain valve lift value (in this case from around 5.8 mm), the minimal cross section of the flow (going through the valve) is not anymore the surface between the valve and the valve seat ("curtain surface") but is port cross section. As a consequence, increase the valve lift has almost no effect on the volumetric efficiency (the small benefit is due to the increase of the lift values that were below 5.8 mm).

3.3.2 Valve timing

To study the influence of valve timing on volumetric efficiency, we made vary the phasing of the intake and exhaust camshafts from -40 to +40 CAD at two different engine speeds: 4000 and 8000 rpm.





At 4000 rpm, a 3% increase of the volumetric efficiency can be obtained by dephasing the intake camshaft by -30 CAD and the exhaust camshaft by -20 CAD.

At 8000 rpm, the phasing of the camshafts is already maximising the volumetric efficiency, so it would be unfavourable to modify it.

The valve timing directly affects the mass of air-fuel mixture going in and the mass of burned gases going out of the chamber during each cycle.

To maximise the mass of air/mixture going in during the intake phase:

- Intake Valve Opening (IVO) should occur as early as possible without reaching the point from which unburned gases start going from the chamber to the intake runner.

- Intake Valve Closing (IVC) should occur as late as possible without reaching the point from which a reverse flow is created ($P_{chamber} > P_{plenum}$)

To maximise the mass of expelled burnt going out during the exhaust phase:

- Exhaust Valve Opening (EVO) should occur as early as possible, but a compromise has to be found because an earlier EVO implies a reduction of the work produced by the expansion stroke.
- Exhaust Valve Closing (EVC) should occur as late as possible without reaching the point from which a reverse flow is created ($P_{exhaust\ manifold} > P_{chamber}$)

3.3.3 Conclusion about valve timing and valve lift

Valve timing and, to a lesser extent, valve lift have an influence on the volumetric efficiency. For a given energetic efficiency, the IMEP is directly proportional to the mass of injected fuel, which in turn is proportional to the mass of air trapped in the cylinder. Consequently, the link between volumetric efficiency and engine performance becomes obvious, hence the importance of choosing intelligently both valve timing and valve lifts.

3.4 Influence of addition of a supercharger/turbocharger on engine performance

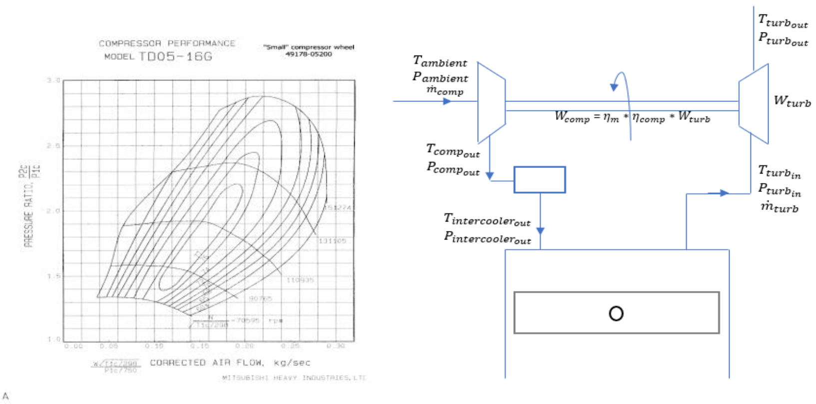


Fig. 9 Reperesentative map of a tubocharger and schematics of the model used

The effect of adding a turbocharger was studied. The following simplifications were made to model the turbocharger:

1. The mean temperature and pressure was considered for the turbine inlet. The mass flow modeled by the gas exchange through valves was used.
2. The turbine efficiency which depends on the mass flow rate (see Figure 9) was assumed to remain constant. The losses during compression was accounted for with another constant efficiency term.
3. For the operating engine speed of 4000 rpm, a pressure ratio of 1.5 was assumed.
4. The pressure drop across the intercooler was assumed.

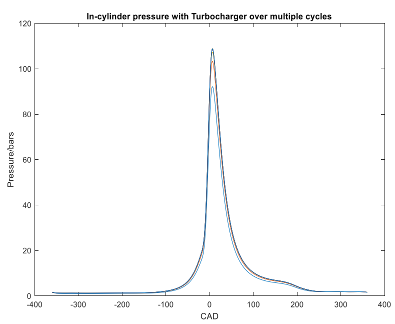


Fig 10 . In-cylinder pressure over multiple cycles with turbocharger

Table 5 depict the results of the addition of the turbocharger. The plenum pressure and temperature are now higher at intake which results in an increase of the engine performance parameters. Again, the trends observed in Table 6 were the only source of validation.

Table V. Comparison of the engine performance parameters with and without turbocharger modeled

Performance Indicators	Wcycle/J	MaxP	MaxT	IMEP	mFuel	Efficiency	PwrOut	SpecPwrOut
Without Turbo	558.56	92.15	3069.07	13.98	0.04	0.37	18.62	46.59
With Turbo	666.11	108.66	3099.33	16.67	0.04	0.39	22.20	55.57

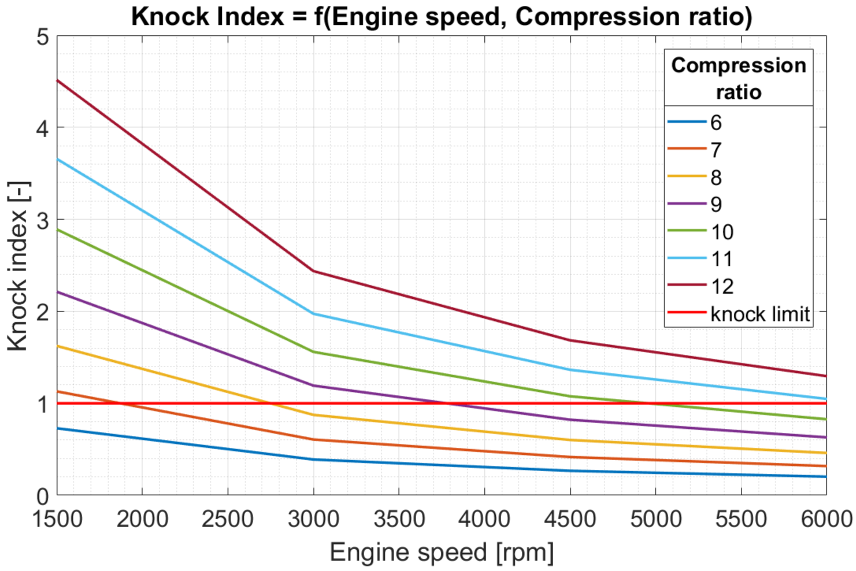
Table VI. Relevant parameters with the turbocharger over multiple cycles

Cycle No.	1	2	3	4	5	6	7	8
Pturb_in/bars	2.26	2.39	2.44	2.46	2.46	2.46	2.46	2.46
Tturb_in/K	1357.88	1345.69	1341.92	1340.76	1340.56	1340.56	1340.68	1340.78
MassFlow_turb/kgs^{-1}	1545.24	1733.92	1803.44	1825.42	1829.98	1829.33	1827.64	1825.67
MassFlow_comp/kgs^{-1}	1546.51	1733.68	1804.76	1826.61	1830.98	1829.40	1828.56	1826.94
Tcomp_in/K	298.00	307.88	311.34	312.41	312.64	312.61	312.51	312.42
Pcomp_in/bars	1.10	1.26	1.32	1.34	1.34	1.34	1.34	1.34
Tcomp_out/K	330.94	342.46	346.02	346.80	346.69	346.38	346.07	345.81
Pcomp_out/bars	1.70	1.95	2.04	2.06	2.05	2.05	2.04	2.03
Tintercooler_out/K	307.88	311.34	312.41	312.64	312.61	312.51	312.42	312.34
Pintercooler_out/bars	1.26	1.32	1.33	1.35	1.34	1.34	1.34	1.33
ρcomp_out/kgm^{-3}	1.16	1.28	1.33	1.34	1.35	1.35	1.35	1.35
ρintercooler_out/kgm^{-3}	1.53	1.80	1.89	1.91	1.91	1.90	1.89	1.89

4 Knock prediction

We already saw in part 3.2 the influence of combustion duration and spark advance on knock. Therefore, in the next analyses, we will keep them constant (10 CAD before TDC for the spark advance and 30 CAD for the combustion duration). In reality, they both vary a lot along the engine map (speed and load). We will focus on the most critical case, which is full load (highest pressure and temperature values).

4.1 Influence of compression ratio on knock occurrence

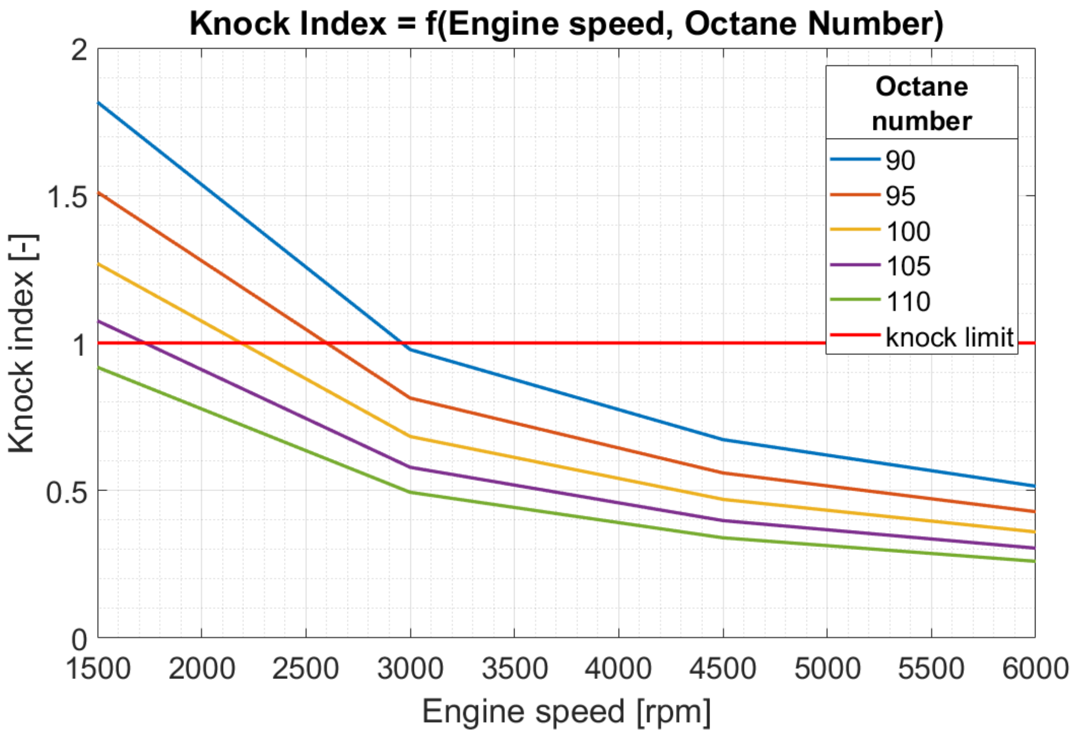


Higher is the compression ratio, higher are temperature and pressure, and so higher is knock occurrence. Higher is the engine speed, shorter is the time duration of the compression and combustion, and so lower is knock occurrence.

At full load and with a spark advance of +10 CAD, the compression ratio can reach 10 only if the engine speed is above 5000 rpm. It means that below this speed, spark advance will have to be reduced if this compression ratio is chosen.

4.2 Influence of octane number on knock occurrence

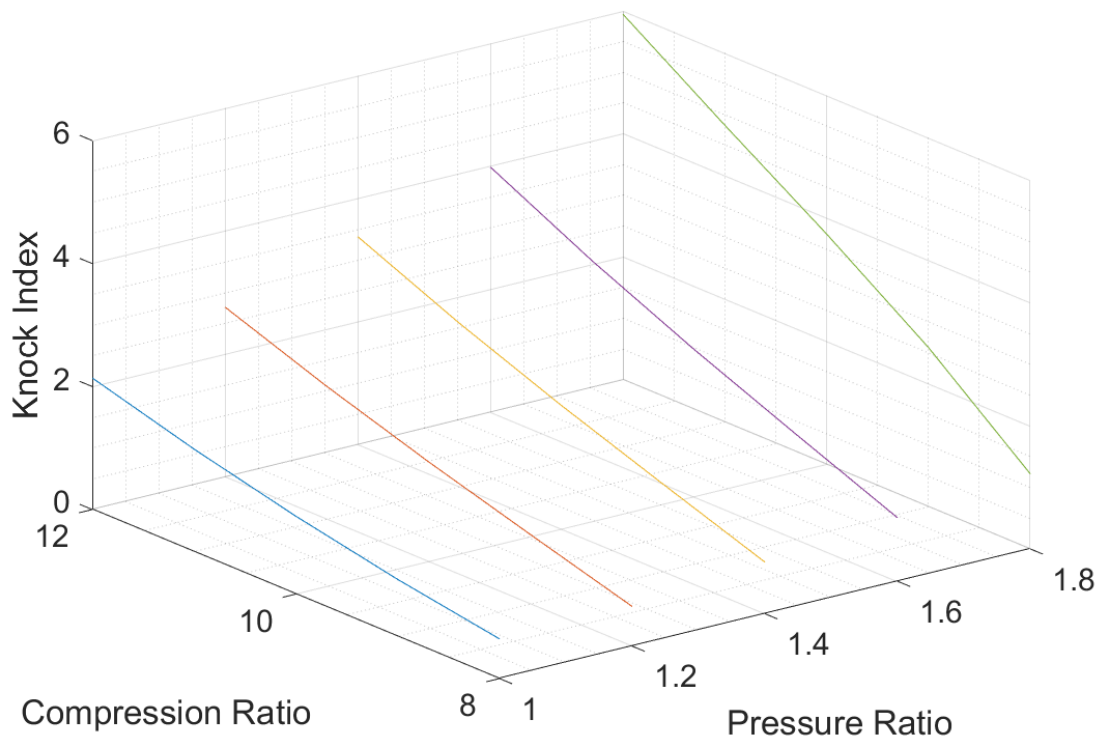
For this analysis, we consider a compression ratio equal to 8.



By definition, lower is the octane number, higher is the knock occurrence. With our combustion timing, the octane number has to be equal or higher than around 108 to avoid knock on the complete RPM range. If it is lower than 108, then the spark advance will have to be reduced at low engine revs in order to avoid knock.

4.3 Influence of turbocharging on knock occurrence

In the case of a turbocharged engine, we used a negative spark advance (-15 CAD) to reduce the knock index values, but this reduction was not sufficient to make the knock index go below 1 on a lot of conditions.



Indeed, by increasing the intake pressure thanks to a turbocharger, the temperature and pressure during compression and combustion are increased, and so is the knock occurrence.

Conclusion

This two-zone model created using Matlab has allowed us to carry out different parametric analyses to study the impact of key specifications (compression ratio, ignition advance, valve lift laws…) on the performance of our virtual internal combustion engine.

This 0-D model is a powerful tool which can be very helpful to do a pre-sizing of the engine before producing it, and a pre-calibration before testing it on a real engine test bench.