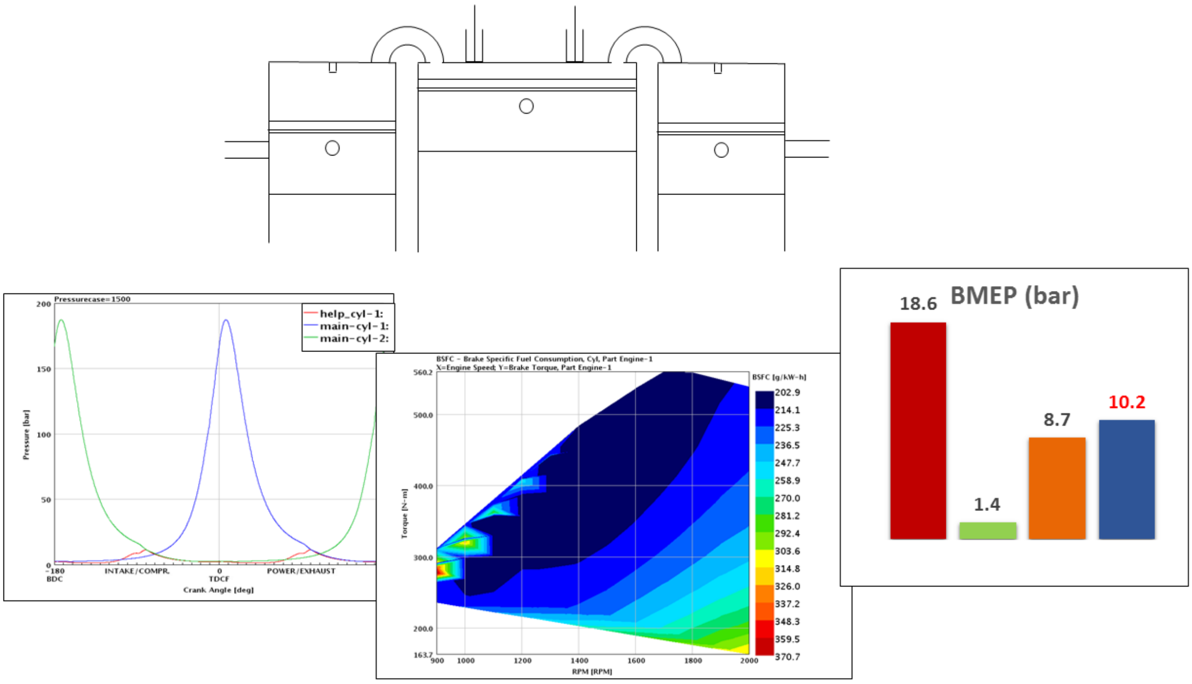

A novel approach to a two-stroke dual stage expansion engine concept

Master's Thesis in Automotive Engineering

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Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2016

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1

Abstract

The SICO engine concept was proposed by Per-Arne Sigurdsson. The engine comprises of two main cylinders for combustion implementing a two-stroke cycle operation and a single help cylinder running at twice the engine speed. At the end of combustion, the burned gases from the main cylinder are transferred to the help cylinder where the second stage expansion occurs simultaneously along with the main cylinder. The cylinders are considered to be thermally insulated and along with the parallel expansion cylinder aims to derive a higher engine efficiency. The charged induction of air is done by means of a compressor cylinder/radial compressor.

The engine concept is evaluated for a power generation application where the engine efficiency at a single operating speed is essential. Consequently, the scavenging and friction models were augmented to better suit the model. Different aspects like the heat insulation, valve timings, cylinder dimensions, cylinder phasing, compression ratio of the main and help cylinders, charged induction using a radial compressor and compression cylinder were evaluated and optimized for the necessary application. Additionally, the SICO concept was compared with existing concepts that implement similar strategies such as the five-stroke engine concept that aims in utilizing exhaust energy for a four stroke cycle operation.

The final SICO engine model is a 3.14L engine, producing a peak power of 81.5kW at 1500rpm. At this operating point the engine runs at an efficiency of 41.3%. Simulations show that the implementation of the help cylinder contributes to around a 7% brake efficiency increase and the effect of heat insulation is not of large significance for the break efficiency. Based on the preliminary evaluation of the concept by means of simulations, the engine efficiency of the concept is comparable to existing modern diesel engines and thus the SICO concept's operation and implementation can be further pursued.

2

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Kiran Subhramanya Banavathy Srinivasa & Prajwal Kagganagadde Shankaregowda

Gothenburg, September 2016.

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List of Abbreviations

η	Efficiency (%)
ρ	Density (kg/m^3)
λ	Ratio of actual Air Fuel Ratio to stoichiometric
5S	Five stroke model
AFR	Air Fuel Ratio
BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure (bar)
BSFC	Brake Specific Fuel Consumption(g/kWh)
CAD	Crank Angle Degree
CC	Compressor Cylinder model
CR	Compression Ratio
EVC	Exhaust Valve Closing (CAD)
EVO	Exhaust Valve Opening (CAD)
FMEP	Friction Mean Effective Pressure(bar)
IMEP	Indicated Mean Effective Pressure(bar)
IVC	Inlet Valve Closing(CAD)
IVO	Inlet Valve Opening (CAD)
MCO	Main Cylinder Only model
TDC	Top Dead Center(CAD)
ZDP	Zero Degree Phase model
V_d	Displacement Volume(Litres)

3

Introduction

This chapter gives a brief introduction about the motivation for the thesis to take form. It also gives details about the objectives, goals and limitations of the thesis.

3.1 Background

One of the main drawbacks of an internal combustion engine is that there are energy losses in the form of exhaust energy. In Diesel engines, around 40 % of the energy is obtained from the engine as usable energy as the rest is lost, either as heat energy transferred to coolants or as exhaust gas energy and the remaining is lost as friction losses, etc [2]. Losses due to heat energy rejection can be reduced to an extent by using some high insulating non-conventional materials. Losses due to exhaust gas energy can be reduced by implementing various technologies such as turbochargers and expansion cylinders.

Two stroke engines have been known for being lighter and simpler in construction in comparison to four-stroke engines since most configurations use intake ports instead of a valve-train. Additionally, since the power produced is once every revolution they could be more powerful than a four-stroke engine where the power produced is once every two strokes. They are not widely used since the smaller engines are not fuel efficient and they produce more emissions due to combustion of oil along with the fuel.

3.2 Aim

The main aim of this thesis is to evaluate the feasibility of the SICO diesel engine concept by analyzing the performance of the concept through simulations in GT- POWER. This thesis work intends to produce the simulation results of the SICO diesel concept and acts as the first implementation of the concept. It does not address the design and emission challenges. However, it acts as a basis to analyze these areas in the future.

3.3 Boundaries

The current thesis work offers the possibility of exploring a vast set of areas due the nature of the topic. However, considering the available resources and the time constraints, the thesis work is carried out under certain boundaries that are defined below.

- The thesis work only intendeds to present the simulation results of the topic and thus no experimental work will be accomplished.
- The thesis will be oriented towards developing only a few key factors identified after the analysis of the base model.
- The structural part of the engine, the NVH, the dynamics aspects of the engine would not be evaluated as part of the thesis.

3.4 Report Outline

This report gives a brief introduction and some theory about diesel combustion and two stroke engine along with a brief information about previous work on similar concepts. In the methodology section, the approach taken during the thesis is discussed. Further, the results of the work are presented along with some discussions and suggestions on possible applications. The report is then concluded along with some suggestions for further work.

4

Theory

4.1 SICO Diesel Engine Concept

The ever increasing demand to improve the efficiency of the existing IC engines and decrease the emissions it produces has driven research and development of new technologies. The need to increase the efficiency has been a key issue that is addressed globally. Some of the challenges faced while designing a diesel engine are the cost feasibility and its impact on the environment. An efficient engine would mean better utilization of the fuel due to consumption of a smaller amount of fuel for the same amount of power output. Most of the energy produced in an engine is lost in the form of heat energy either through coolants or to the atmosphere through the exhaust.

The SICO diesel engine concept was developed by Mr. Per Arne Sigurdsson with the aim of improving the engine efficiency through extra expansion of the exhaust gases and by insulation of the cylinders in order to reduce the heat transfer through the cylinder walls [1].

4.1.1 Basic Concept

The proposed concept is mainly a 2 stroke engine and is made up of four cylinders. A schematic representation of the concept is as shown in the figure 4.1 . It consists of a compressor cylinder that compresses air to higher pressures and transfers it to the two *Main Cylinders.* The main cylinders are made up of two combustion cylinders running on 2 stroke cycle and diesel fuel is burnt in the cylinder chamber. These two cylinders are phased at 180 degrees to each other. Finally, the burned gases are further expanded in the additional expansion cylinder, termed as the *Help Cylinder*, before being sent out through the exhaust valve. The compressor cylinder and the help cylinder run at twice the speed as that of the main cylinders. Transfer ports between the cylinders assist in transferring the gases between the cylinders.

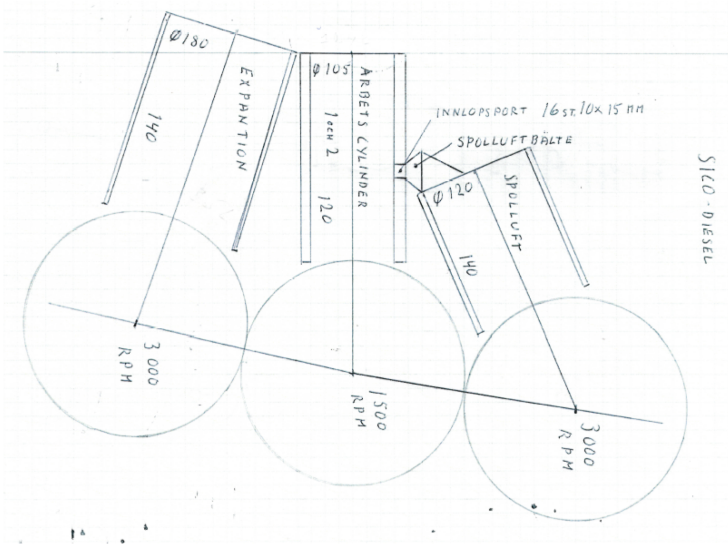


Figure 4.1: A schematic representation of the SICO diesel engine concept from the inventor

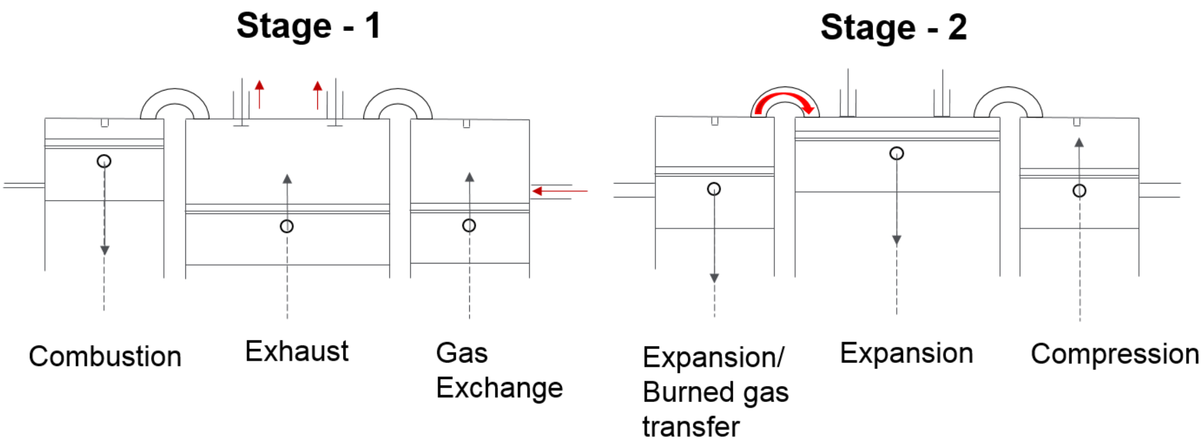
The cycle to cycle description of the concept is as represented in figure 4.2. The main cylinder and help cylinder movements and the corresponding variation in their positions is as shown. The two main cylinders are termed as cylinder 1 and cylinder 2 and the complete process is split into 4 stages for simplification of depiction.

- Stage 1 : As represented in Figure 4.2, when main cylinder 1 is at TDC fuel is injected. Consequently, main cylinder 2 is in the intake phase and is at BDC. The help cylinder is also at BDC after completing the second stage expansion stroke from the contents of cylinder 1.

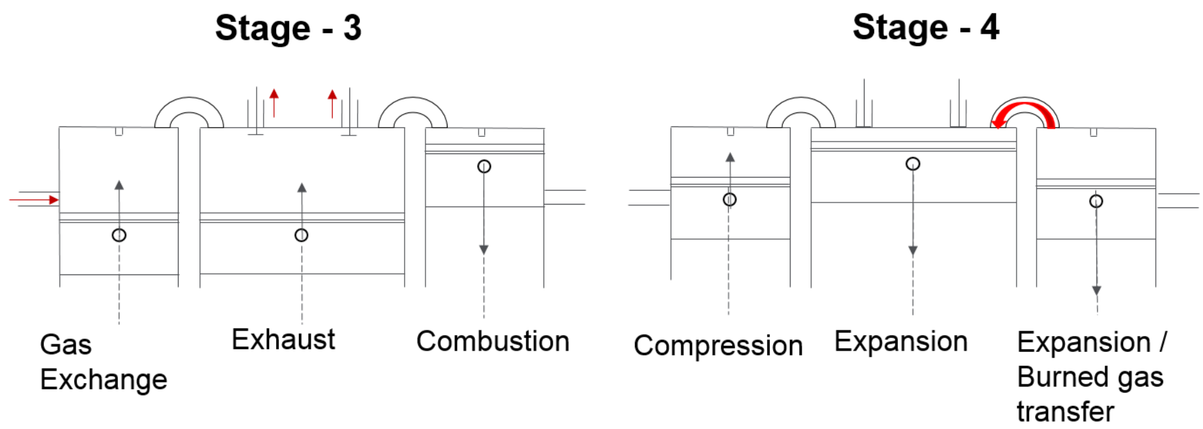
- Stage 2 : Combustion occurs in main cylinder 1 and results in expansion. When the piston is at 90 degrees from TDC, the help cylinder is at the TDC owing to the fact that it operates at twice the speed. At this point, the transfer ports are opened and the expanding gases from main cylinder 1 transfer to the help cylinder while simultaneously expanding in both the cylinders.

- Stage 3 : At around 60 degrees before BDC, the intake ports in the main cylinder 1 open and thus fresh charge begins to flow into the cylinder. The fresh charge also helps in scavenging some of the burned gases into the help cylinder before the transfer port is closed. The help cylinder performs the second stage expansion of these burned gases and reaches BDC along with main cylinder 1. Main cylinder 2 is at TDC before power stroke.

- Stage 4 : As the fuel is burnt in main cylinder 2 and the piston reaches halfway through its expansion, the transfer ports between main cylinder 2 and the help cylinder opens and the above cycle repeats.



(a) Working Cycle : Stage 1 and Stage 2



(b) Working Cycle : Stage 3 and Stage 4

Figure 4.2: Working Cycle of the SICO diesel engine concept

4.2 Two - Stroke Engines

Two stroke engines are internal combustion engines that have a power stroke for every crankshaft revolution which differs from four stroke engines where the latter produces a power cycle for every two revolutions of the crankshaft. Thus theoretically the two stroke cycle operation produces twice the power of that of four stroke engines. But in reality it is not possible due to several limitations of the operation cycle. These engines are used in light weight applications such as lawn movers, chainsaws and light motorcycles. Efficiency is one of the major drawbacks of this engine when it comes to smaller applications, although the most efficient engines in the world are two stroke marine engines.

The two stroke cycle is shown in the figure 4.3

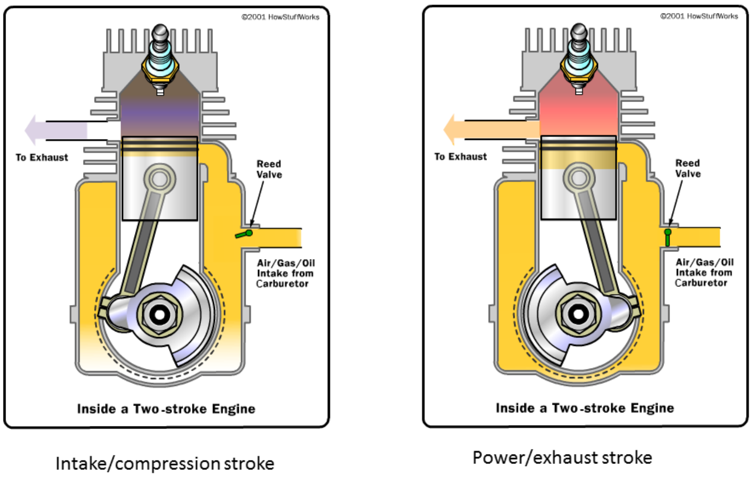


Figure 4.3: Two stroke operation cycle [6]

The cycles of the two stroke engine are compression and exhaust strokes. Since there are no separate strokes for intake and exhaust events, the timings of the gas exchange events overlaps to a large extent when compared to a four stroke cycle operation to facilitate gas exchange. Scavenging is one of the major concerns for efficiency of the two stroke engines as obtaining a low amount of residual gases while having a good amount of fresh air trapped is a challenge in a two stroke cycle operation.

4.3 Scavenging

Scavenging is the process of the removal of exhaust gases in the cylinder by the fresh air and it in turn getting trapped. To have a clearer picture of the scavenging process there are a few definitions as per Heywood literature[2]

$$Delivery\ ratio = \frac{Mass\ of\ delivered\ air\ per\ cycle}{V_d * \rho} \tag{4.1}$$

The delivery ratio is the ratio of the mass of fresh air being delivered to the product of the displacement volume and the density of the delivered air. It gives a measure of the amount of intake air.

$$Purity = \frac{Mass\ of\ fresh\ charge\ trapped\ in\ cylinder}{Mass\ of\ trapped\ cylinder\ charge} \tag{4.2}$$

The purity ratio is the measure of how effective the fresh charge is in scavenging the exhaust gases.

$$Trapping\ efficiency = \frac{Mass\ of\ delivered\ air\ (or\ mixture)}{mass\ of\ delivered\ air} \tag{4.3}$$

The challenge is to have a high purity ratio while ensuring that the trapping efficiency doesn't drop. There are various methods of scavenging, namely uni-flow, cross-flow and loop scavenging methods. Uni-Flow scavenging methods have been known to obtain a greater purity ratio when compared to the other methods but with a lower trapping efficiency. The various scavenging methods are shown in figure4.4

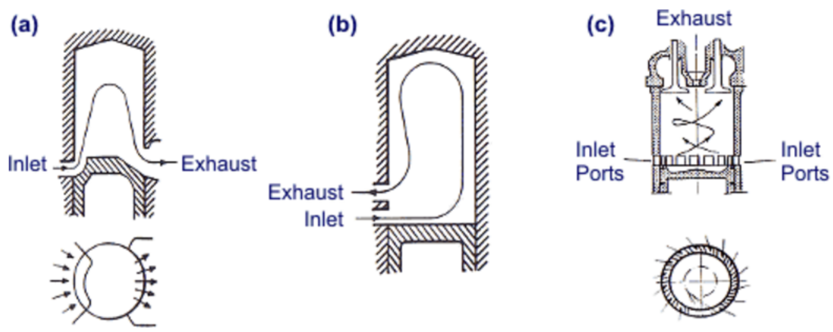


Figure 4.4: Scavenging methods- (a)Cross-Flow, (b)Loop scavenging, (c)Uni-flow[2]

The efficiency of the above methods in scavenging the exhaust gases are described by the figure 4.5

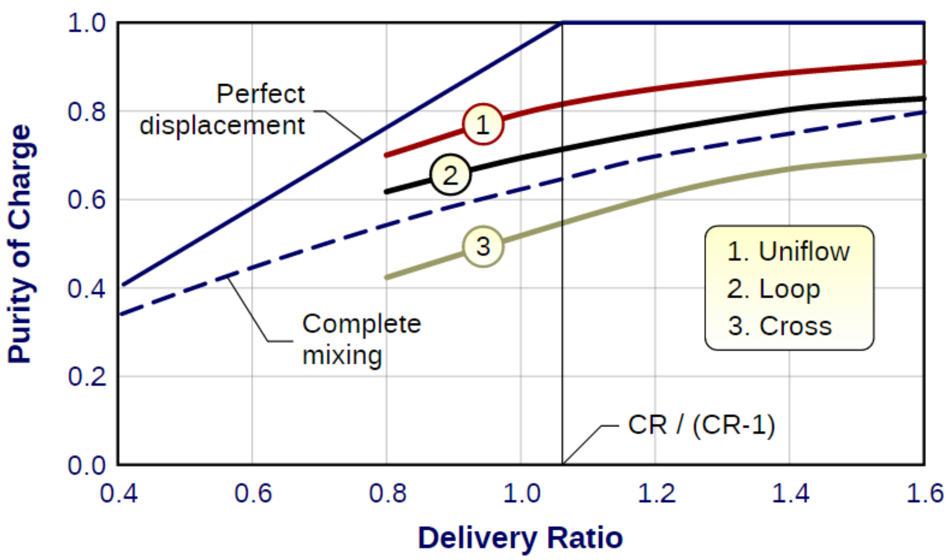


Figure 4.5: Purity ratio vs Delivery ratio for different scavenging methods[3]

As shown above the uni-flow method is more efficient in the expulsion of the burnt gases than the other methods. CFD methods can be employed to obtain the actual scavenging curve for the engines as an alternative for experimental results.

4.4 GT-Suite software

GT-Suite is a multi-physics tool developed by Gamma Technologies. It is a software tool to carry out simulations on power-train components. Such a simulation tool can be used to reduce the development time of a concept and also in investigation of an existing concept to optimize the sub components of the system and even finding the root performance affecting parameters in the system.

The major benefit of the software is that it has an extensive library that consists of many major systems such as flow, mechanical, thermal, electromagnetic and control systems with subsystems required to implement components like an after treatment device, or a detailed valve train system. The software has some extra features like the capability to implement a 3D model for a thermal or structural analysis of a few parts that require a detailed assessment such as the cylinder. The CAD tool of the software allows designing and importing a 3D geometry into the model. The use of such features are mainly that either a simple power-train model can be developed for quick results and to assess a wide range of the desired variables. Or else, a particular system can be defined in detail to assess its precision while keeping the other sub components simple to reduce the computational time. Example is a detailed gas exchange model with a simple combustion model in the cylinder. For the simulations done for this thesis, the GT-Suite Version 7.5.0 build 3 was used.

4.5 Friction

Friction is one of the significant losses in engines. Friction work as defined in Heywood[2], is the difference in the amount of work delivered to the piston while the working fluid is contained in the cylinder(during compression and exhaust strokes) and the usable work delivered to the drive shaft. The friction work is the summation of the three subsystems-

1. Pumping work by the piston

2. Resistance from relative motion of the moving parts of the engine. These include the piston assembly relative to the cylinder, the crankshaft components , valve train assembly and the gears and pulleys of the engine.

3. Accessories attached to the engine such as the oil pump, the coolant pump, cooling fan.

Although the pumping work contributes to the friction, the amount of energy utilized due to friction during pumping is defined as Pumping losses in models(Pumping Mean Effective Pressure). Friction can account to 5-10% of the energy losses in engines. The losses are higher with higher engine speeds. The friction distribution and its dependence on speed for a four/six cylinder diesel engine is shown is figure 4.6for a motored test-

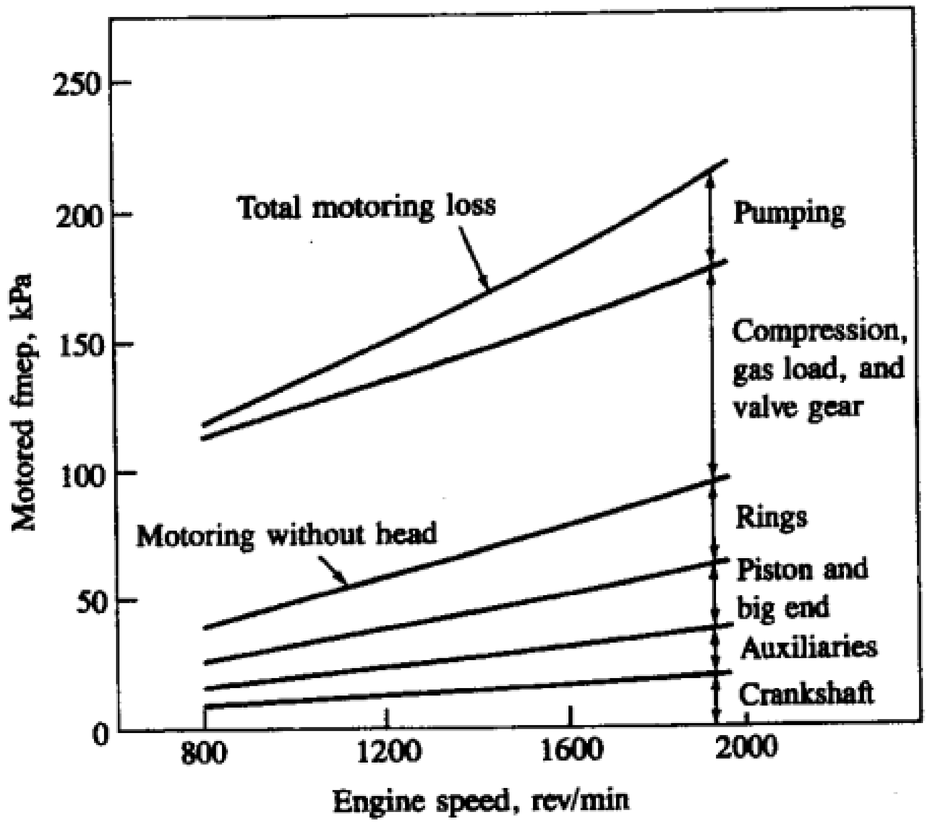


Figure 4.6: Friction distribution in a four/six cylinder diesel engine [15]

Since its effect on the total usable energy output is significant, its essential that the friction estimation is accurate. Usually the friction of the engine is experimentally determined for a configuration of engine. For simulations, an estimation of the friction is done using models such as the Chen Flynn Friction model[9] and Schwarzmeier Reulein[20] Engine Friction Model. There are also detailed friction models that consider oil temperatures, loads at each operating point to define friction for each of these points and modeling the dimensions and the material properties of each of the friction components such as the bearings,piston rings,skirt,etcetera. Due to simplicity of the model, the Chen Flynn model is used to estimate the friction for the SICO engine concept.

4.6 Compression Ratio

The Compression Ratio(CR) of a cylinder is the ratio of the maximum cylinder volume to the minimum cylinder volume. The ratio is defined by the formula 4.4

$$Compression\ Ratio = \frac{V_d + V_c}{V_c} \tag{4.4}$$

The max volume of a cylinder is the summation of the displacement volume V_d and the clearance volume V_c. The minimum volume being equal to that of the clearance volume, V_c.

This represents the geometric CR of the cylinder. The actual CR referred to as the effective CR depends on the valve closing event since the compression that occurs in the cylinder starts only after the valves are closed. Geometric compression ratio and the expansion ratio is essentially the same for a cylinder but as mentioned, the effective ratios can vary.

A greater value of compression ratio would result in an increase in the thermodynamic efficiency of the engine since there would be more volume for expansion of the gases thus increasing the work done by the gases in the form of expansion. Since when compared to a SI engine, in a CI engine there are not much concerns when it comes to engine knock, the CR of diesel engines can be high. The effect of variation in peak cylinder pressure timing is reduced by means of injection timing such that the peak pressure point occurs at the Crank Angle Degree (CAD) favourable for the performance of the engine. At greater values of CR, the engine friction increases due to greater peak cylinder pressures and hence contributes to lower brake efficiency. Due to greater CRs the losses due to heat transfer increases. In addition, greater cylinder pressures would mean a greater increase in the loads acting on the cylinder and piston and so would require a more rigid construction. Diesel engines usually run at CRs of around 14-19.

4.7 Over expanded Cycles

The exhaust gases at the end of the power stroke have energy that is allowed to dissipate in the exhaust and is thus lost. This energy can be utilized by allowing it to expand further and thus obtaining more work per cycle. The decoupling of the compression ratio and the expansion ratio contributes in realizing higher efficiencies. Thus, the additional work for the same amount of fuel input results in improved engine efficiency. The techniques used to extract this additional work is as described below. [2]

4.7.1 The Atkinson/Miller Cycle

The Atkinson cycle is a type of overexpanded cycle proposed by James Atkinson [16]. The cycle intended to improve the efficiency by making the compression stroke shorter than the expansion stroke and thereby achieving better thermal efficiency at the cost of the power obtained. It works on the principle that if the compression ratio is smaller than the expansion ratio, then an additional work can be extracted from the cylinder thus resulting in improved efficiency compared to a conventional engine [1]. Several methods have been proposed to realize the Atkinson cycles and have been analyzed [18] [19].

The Miller cycle is an extension of the Atkinson cycle and was proposed by Ralph Miller [17]. The Miller cycle also aims to achieve improved efficiency by having shorter compression stroke and longer expansion stroke but without physically varying the length of the strokes. The cycle employs valve timings to obtain smaller compression strokes by delaying the closing of the intake valves thereby reducing the actual length of compression. A supercharger in the form of a Roots blower or a screw pump is employed to provide additional air so as to negate the power loss caused due to the delayed intake valve closing.

4.7.2 Extra Expansion Cylinder

One method of separating the compression and the expansion ratio is with the employment of an extra non-firing expansion cylinder. Generally, the burned gas from the main combustion cylinder is transferred to the expansion cylinder in order to further extract work from the high pressure burned gases. This results in an increased overall expansion ratio and thereby should lead to an improved efficiency. This thesis concentrates on evaluating the effectiveness of having an extra expansion cylinder. This is the main concept that is employed in the current thesis work and its effectiveness when coupled with a two stroke diesel engine is evaluated. However, similar concepts have been studied previously and some of the different applications of the extra expansion cylinder with different modifications to the cycles have been adopted and analyzed. Some of the major studies have been presented in the following sections along with their important findings.

5

Literature Review

5.1 Double Compression Expansion Engine Concept

The Double Compression Expansion Engine (DCEE)[4] concept aims to achieve a high indicated and mechanical efficiency. The design involves two cylinders where one is referred as the High Pressure (HP) cylinder where combustion occurs whereas the second cylinder is for inducting compressed air and for second stage expansion of gases from the HP cylinder referred as the Low Pressure (LP) cylinder. The base cycle being a 4 stroke operation for a compression ignition combustion with diesel as the fuel. The assumption is since friction plays a major role on the break efficiency value, the LP cylinder is modelled similar to that of a NA, SI engine for a lesser rigid application.

Table 5.1: Dimensions of the DCEE concept

	Unit	$\lambda = 1.2$	$\lambda = 3$
Bore HP cyl	mm	95	95
Stroke HP cyl	mm	100	100
CR, HP cyl	-	11.5	11.5
Bore LP cyl	mm	317	249
Stroke LP cyl	mm	100	100
HP cyl displacement volume	dm^3	0.71	0.71
LP cyl displacement volume	dm^3	7.9	4.9

The boost pressure is due to the compression of the inlet air in the LP cylinder and the compressed air exiting the LP cylinder is around 7-16 bar and the LP cylinder is of a large displacement volume in comparison to that of the HP cylinder (6-12 times larger) to facilitate adequate mass flow. High loads can be obtained due to the high boost pressure and so the peak cylinder pressure can reach high values in their case, 300bar. In general, a higher peak cylinder pressure leads to higher friction. A lower wall surface area to cylinder volume ratio was chosen to reduce the amount of heat transferred to the walls. The main aim however was to achieve greater engine efficiency by means of higher indicated efficiency and mechanical efficiency. The indicated efficiency increase being due to a greater diluted charge available in the HP cylinder for combustion. As explained the LP cylinder is assumed to be of less rigid construction whereas the HP cylinder being rigid. Thus the strength assumption for larger LP is to reduce the friction contribution and hence the overall mechanical efficiency is higher. The simulation was performed for the engine speed of 1900rpm.

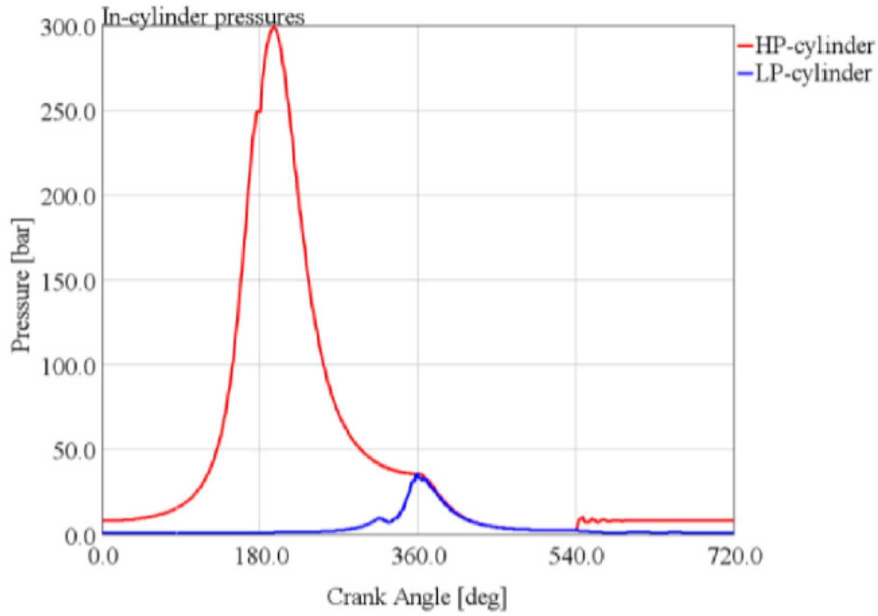


Figure 5.1: DCEE concept simulated pressure trace[4]

The operation cycle is as in the figure 5.1. At 0 CAD, the second stage compression begins in the HP cylinder to reach a pressure of 250bar at 180 CAD. During this time, the intake stroke operation occurs in the LP cylinder. After 180 CAD, combustion occurs in the HP cylinder whereas the LP cylinder performs the first stage compression. Once desired pressure is achieved in the LP cyl, it transfers this air to the Charged Air Cooler(CAC) unit and continues compression to attain similar pressure value as to that in the HP cyl when they are connected at 360 CAD. So the CAC cools the compressed air as well as stores it until 540 CAD.

The best indicated efficiency achieved by the DCEE concept was 60% and the brake efficiency of 56%. The DCEE concept utilizes a parallel expansion strategy in resemblance to the SICO concept and achieves a high engine efficiency. The mechanical efficiency is estimated to be 92%-96% based on the assumption that the construction of the large LP cylinder is lighter which is quite a high value considering the high peak cylinder pressure. According to the authors of this thesis, this calculation of the mechanical efficiency using the Chen Flynn model assuming that the terms for speed dependency and constant term could be neglected is a strategy that is debatable. This strategy of the DCEE concept and the high indicated efficiency strategy is of importance to the SICO concept study as the aim of the SICO engine is to also impre the indicated efficiency by a dual stage expansion process and additionally having a well thought out basis for the friciton calculation.

5.2 Scuderi Split Cycle Research Engine

The Scuderi Split Cycle (SSC)[5] Engine as titled, is essentially a 4 stroke engine operation split into two cycles. The configuration consists of two cylinders, namely a compressor cylinder that does the work of intake stroke and also the compression and the expander cylinder where the power stroke and exhaust strokes occur. These two cylinders are connected via a crossover port. The basis for the split of the cycle into two individual separate cycles is to optimize the cylinder specs for optimum performance of each of these strokes. Furthermore, the crossover port facilitates in better mixing due to high pressure flow across the port. The simulation model built was with the flexibility for SI/CI combustion with or without a Turbocharger.

The SSC engine's cycle split is to have a less rigid construction for the compressor cylinder and additionally the engine utilizes a Crossover port (Xover port) for the gas exchange between the cylinders. The SICO concept aims to have an expansion cylinder of a lighter construction and also utilizes transfer ports for the exchange of gases between the cylinders. The SSC is of interest because of the similarity in this strategy between them.

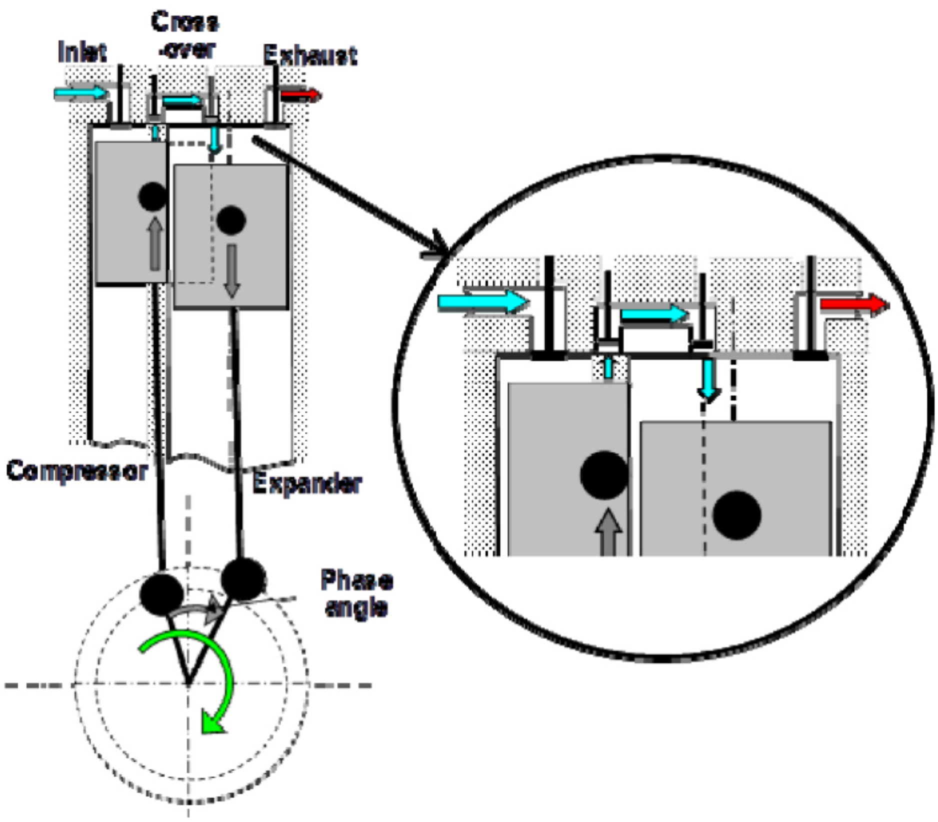


Figure 5.2: Scuderi Engine overview[5]

A hybrid arrangement was also thought about where an air tank was connected at the crossover port for storing the pressurized air for scenarios like engine breaking and also in operating points where additional air is required. The valve-train was designed as poppet valves since its easy to implement from existing production engines. These were pneumatically actuated in order to be able to have Variable Valve Timing applications and also for ease to have various test scenarios for the research engine. A test engine was built to analyze the results and this engine was operated between the ranges of 700-3500RPM. The engine was a 1.1L Naturally aspirated engine with the fuel used for the test data being Motorsport 101. Details of the engine are shown in table 5.2 -

Table 5.2: Scuderi Research Engine Test Specifications

Compressor Bore	87mm
Compressor Stroke	99mm (81mm TC)
Compressor Displacement	0.59L (0.48L TC)
CR , Compressor	96:1 (geometric)
Expander Bore	87mm
Expander Stroke	87mm
Expander Displacement	0.52L
Expander's Expansion Ratio	50:1(geometric)

The compression and expansion ratios of the cylinders that are given in table 5.2 are geometric values and the actual ratios depends on the Xover ports volumes as the effective CR is the summation of the clearance volume and respective Xover ports volume. The 1-D simulations done were for a load case of BMEP 7-10 bar in the case of NA engine whereas the TC simulations were for loads ranging from 11-17bar BMEP. The physical test performed was for a maximum load case of 10bar BMEP, although the results from the actual test data showed lower BMEP values. The tests done were using either both the crossover valves or just one. The best BSFC reached as per simulations were 267 g/kWh and 257 g/kWh for NA and TC respectively. Whereas the physical test data achieved a best BSFC of 320 g/kW.hr.

5.3 Five stroke Engine

The five stroke engine is a concept that has been studied to evaluate possible increase in the thermal efficiency with an aim of improving the expansion while having no reduction in the compression [7].

The engine consists of two combustion cylinders and an additional expansion cylinder which facilitates the extra expansion. The main combustion cylinder works on 4-stroke cycle while the additional help cylinder operates in a 2 stroke cycle. The combustion cylinder performs the intake, compression and power stroke and then transfers the exhaust gases to the help cylinder through a transfer port. The gases are further expanded in the expansion cylinder [8].

The concept has been evaluated for gasoline engines. When considered for a range extender application, the concept yielded a BSFC of 226.4 g/kWh, giving an overall efficiency of 36.1% when running at 4000 rpm and yielding a brake power of 32.8 kW [7]. A comparison between 3 cylinder 5 stroke GT Power model of the concept and a standard 2 cylinder 4 stroke model yielded a maximum increase of 0.57% in the brake thermal efficiency at lower engine speeds. This study was carried out as part of the a parametric evaluation of the concept to determine the effect of various parameters in the 5 strokes SI engines performance [8]. The significant increase in the friction due to the additional cylinder judged to counter the effects of the extra expansion work that is obtained leading to negligible efficiency improvements [8].

5.4 Application

The setup of the SICO diesel engine concept is based on a two stroke cycle operation consisting of multiple cylinders and a charging system. Initially, the base model was developed and was evaluated over a range of speeds and loads. It was initially observed from the results that the concept's performance was limited to lower speeds. Additionally, simulations showed that the concept had lesser volume utilization owing to limitations of the trapped air quantity. Iterations proved effective when the setup was modeled for a particular engine speed. Considering these limitations coupled with the size of the engine makes the system suitable for a stationary application. Hence, the work was aimed at developing and evaluating the SICO concept for a stationary application.

Diesel engines have been widely used in the power generation sector. The power generator set consists of electric machines coupled to a diesel engine. They are used for supplying power either as a back up for a main power source and are called as standby power generator or they are also used in a continuous application and are termed as prime power generators. Table 5.3 gives a listing of some of the commercially available generator sets along with a few important specifications. These engines are chosen as they are sized approximately close to the SICO diesel engine in terms of engine displacement. A comparison can be obtained with respect to the effectiveness of the SICO concept engine with respect to these existing engines under the generator set application,

Table 5.3: Specification of commercially available generator set engines[23][24][25]

Parameter	Volvo Penta TD520GE	Perkins 1104A-44TG1	Cummins QSB5-G4
Operating Speed (rpm)	1500	1500	1500
Displacement Volume (L)	4.76	4.4	4.5
Standby Power (kW)	85	65.6	103
Fuel Consumption - Full load (g/kWh)	208	208	210

Based on these specification, the concept will be evaluated to analyze its performance at 1500 rpm while aiming to achieve a brake power of around 80 kW.

6

Methodology

The following chapter explains the approach employed to conduct the thesis work. Figure 6.1 represents the work flow adopted. Initially, a brief literature review of the concept was carried out. The work was then split into two main parts. They are :

- Base Model Simulations
- Advanced Model Simulations

The results from the base model were analyzed carefully and the critical aspects and parameters of the concept were identified. These primary results were used in the advanced model simulations to further develop the model in order to obtain more optimized results. Finally, the results from the advanced model was used to evaluate the performance and feasibility of the concept along with the possible areas of applications.

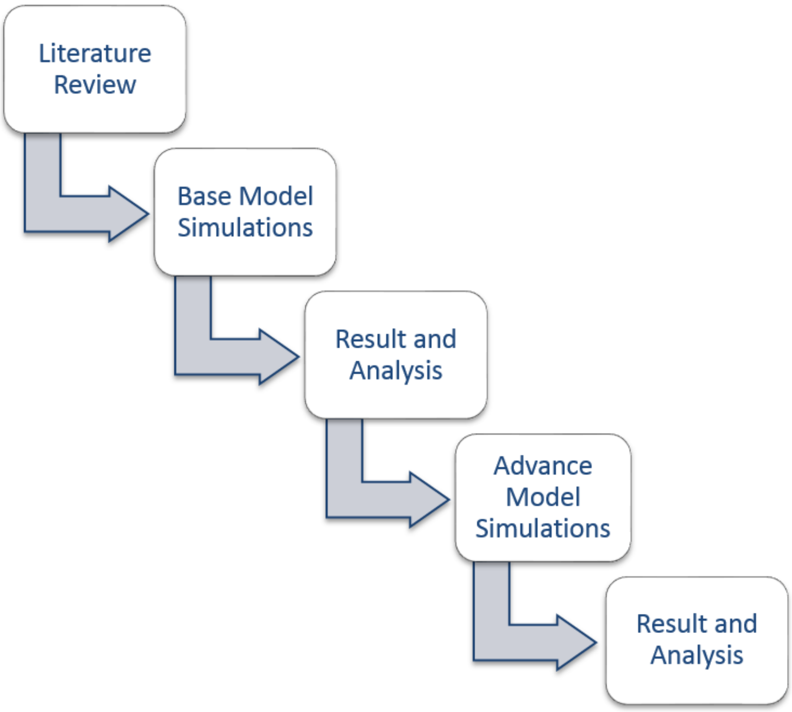


Figure 6.1: Process Chart

6.1 Base Model Simulations

The base model simulations act as the first implementation of the concept. The basic idea proposed by the inventor is modelled using GT-Power with some modifications. The various modelling approaches taken for each of the components is discussed below.

6.1.1 Main Cylinders and Help Cylinder

The dimensions of the main and help cylinders are one of primary factors that influence many different parameters of the engine. As a first step, the dimensions suggested by the inventor was taken for the base model simulations and is as given in table 6.1.

Table 6.1: Cylinder dimensions for the base model.

Cylinder	Bore (mm)	Stroke (mm)	Swept Volume (L)
Main	105	120	1.03
Help	180	140	3.56

The help cylinder is required to run at twice the speed as that of the main cylinders. In the simulation environment, this is achieved describing a dependency table defining the position of the piston with respect to crank angle degrees. Figure 6.2 gives the profile used to describe the position of the help cylinder piston over 1 cycle. The Y-axis represents the length of the stroke and X-axis gives the corresponding crank angle degrees. It should be noted that this profile only dictates the piston position with respect to global crank angle degrees with 0 mm representing TDC and -140 mm representing BDC. The Y-axis can be scaled depending on the stroke length requirements.

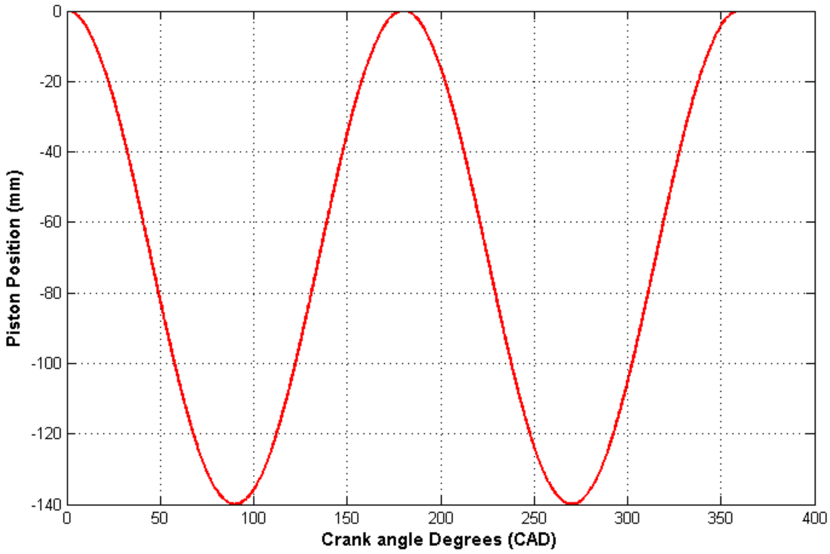


Figure 6.2: Piston position of the Help Cylinder

6.1.2 Valve and Ports

Intake ports are modelled in the main cylinder liners in order to facilitate the induction of the fresh charge. The opening and closing of the port is dependent on the piston movement. Standard values from the existing GT-Power example models with some slight modifications to fit into our model are used as a starting step. A symmetrical area profile is used to model the rate at which the port is opened and closed and is as shown in figure 6.3. Thus, the opening and closing of the port occurs at 60°on either side of the BDC. A maximum area of 1500 mm^2 is exposed before the piston reaches BDC. The motivation behind the choice of ports over poppet valves at the intake is based on the fact that the two stroke cycle working is largely impacted by scavenging. Uniflow scavenging has been known to give the most efficient scavenging when compared to other methods [2]. Thus, use of ports would facilitate in achieving this form of scavenging in practical scenarios though similar results can be obtained through both valves and ports in a simulation environment. Further, friction losses can be reduced with the elimination of valve and the corresponding valve train.

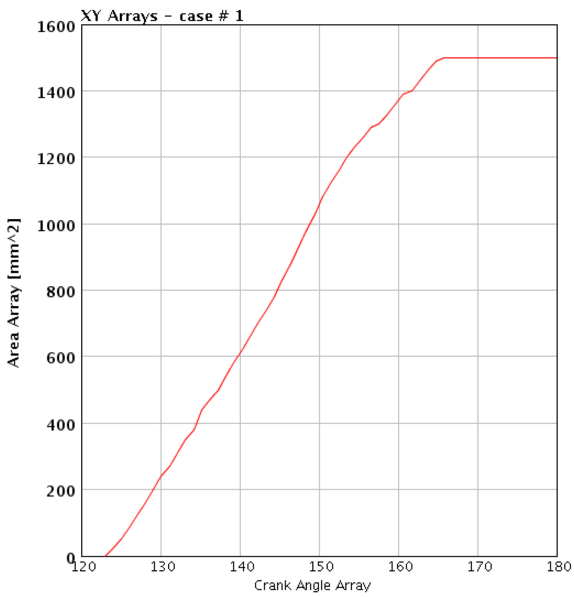
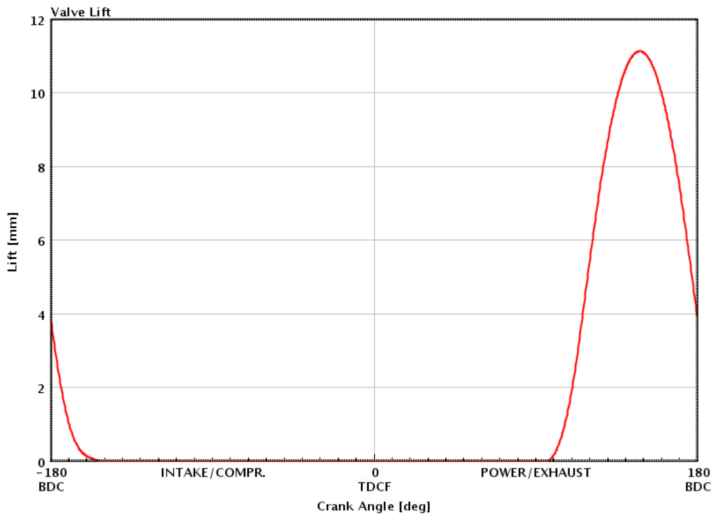
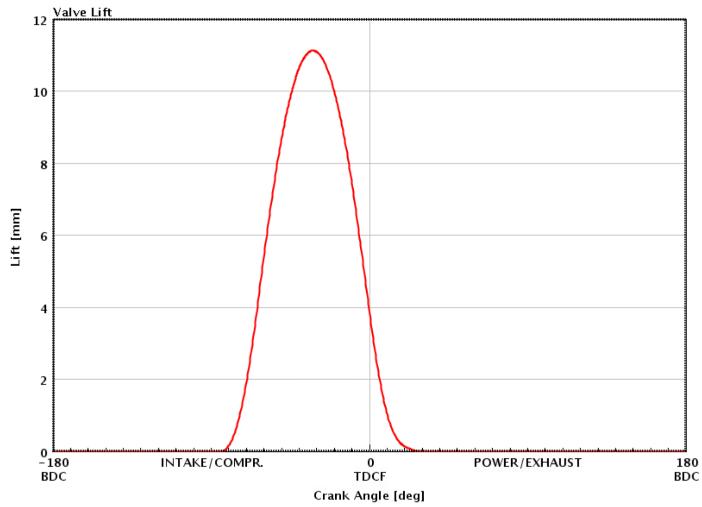


Figure 6.3: Half profile of the port area

The exhaust or the transfer of the burned gases from the main cylinder to the help cylinder occurs through a transfer channel/port which is controlled through poppet valves located on the head of the main cylinder. The final exhaust from the help cylinder after the second expansion is also through a valve situated on the head of the help cylinder. Standard values from the existing example models in GT-Power were used to model the valves and were timed as per the requirement. Figure 6.4 and 6.5 indicates the valve lifts and the timings of the three valves used. It should be noted that the exhaust valve of the help cylinder is opened twice for every 360°rotation of the main shaft in order to adjust to the fact that the help cylinder is operating at twice the speed of the main cylinders. Driver objects are used in GT-Power to trigger the opening and closing of the valves based on the speed of the main crankshaft.



(a) Main Cylinder 1



(b) Main Cylinder 2

Figure 6.4: Exhaust Valve Lifts and Duration

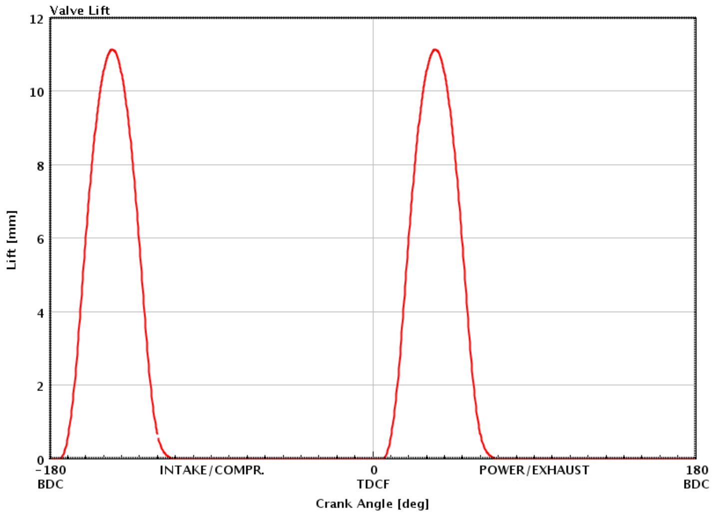


Figure 6.5: Help Cylinder exhaust Valve Lift and Duration

6.1.3 Intake and Exhaust

The intake and exhaust system was developed based on standard pipes and orifice components from the existing GT-Power example models with modifications to the diameter and the lengths as per the requirement. All the remaining factors were maintained as it is without any modifications. Further, a compressor and a inter-cooler unit was used in the intake system in order to obtain the necessary boost pressures.

The inter-cooler is also taken from the sample model and the pipe lengths and diameters are modified as per the requirements. The friction multiplier, which is used to scale the amount of friction occurring in a flow volume, is increased in order to develop realistic pressure drop across the inter-cooler. The nature of operation of the SICO concept creates a necessity for a boosted charge intake for the cycle to sustain. The help cylinder creates a back pressure which results in very high residuals in the main cylinder. In order to achieve a good scavenging, charge at higher pressures are required. Hence, a radial compressor is also modelled based on a two stroke sample model available in GT-Power [22]. The map is scaled according to the requirement using the mass, efficiency and the pressure ratio multipliers. The compressor is coupled with the engine crank shaft through a gear arrangement and thus acts as an auxiliary unit drawing power from the engine therefore reducing the efficiency. However, the increase in volumetric efficiency due to the boosted charging helps to obtain higher power and thus offsets the loss in efficiency. The nature of the cycle makes it unlikely to employ a turbocharger as the exhaust gases are almost expanded up to the atmospheric conditions and thereby giving very little energy for the turbocharger usage.

6.1.4 Heat Transfer, Friction and Scavenging

As a starting point, the standard models available in GT Power were used for the heat transfer and friction.The Woschni GT model was used to estimate the heat transfer in the cylinders. This model calculates the in-cylinder using formulas which are similar to classical Woschni heat transfer model without swirl [10]. This model differs from the original model in the way the heat transfer coefficients are taken when the valves are open. The increase in heat transfer due to the inflow velocities at the intake and the backflow velocities at the exhaust valve is accounted [22].

The Chen-Flynn friction model[9], explained in detail in section 6.2.3, was used to estimate the friction and the Vibe combustion model [13] was used. A standard scavenging curve from a GT-Power two-stroke example model, giving the relation between the amount of fresh charge in the cylinder and the quantity of burned gases exiting the cylinder is used to define the scavenging. The scavenging dictated by this curve is different from the practical scavenging that can be achieved. However, it serves as a first step for the analysis and an assumption can be made that the geometrical designing can be modified to achieve such a curve. More about the scavenging is discussed in detail in section 6.2.4. All the parameters for the heat transfer, friction, scavenging and combustion models were taken from the standard templates available in GT Power. [22].

6.2 Advanced Model Simulations

Based on the defined parameters and results of the base model simulations, more advanced models were built in order to analyse the effects of incorporating new subsystems and also variation in the strategy and/or dimensions of the components. These included the implementation of strategies like heat insulation for the help cylinder and main cylinder, higher compression ratios of the main cylinder, modifications to the dimensions of the cylinders, implementation of the bypass exhaust valve and the evaluation for the need of a valve at the help cylinder for the transfer port inlet point of the help cylinder. These concepts were individually evaluated and the results are given below. For these investigations, the results shown in this report are using the values of the variables used in the final version of the SICO model.This was to ensure that the comparison of the results are in coherence with the final engine values. The simulation results below are for a speed of 1500rpm with a load case for 90mg of injected fuel(full load condition) injected such that combustion ends before the opening of the transfer port.

6.2.1 Cylinder Dimensions

The cylinder dimensions used in the base model simulation was evaluated in order to check for improvements that can be achieved in terms of performance and gas exchange process. As explained in section 6.2.3, the friction has dependency on the mean piston speed which in turn depends on the stroke length and the speed of operation. The large stroke coupled with the higher speeds of the help cylinder causes high friction and results in reduced efficiency. An optimized valve timing further help in improving the gas exchange efficiency.

A Design of Experiments (DOE) tool was used within GT-Power in order to optimize the values of the bore and stroke for the help and main cylinders. The model was operated at 1500 rpm while running with 90mg of fuel. As mentioned in section 5.4, the normal speed of operation for the generator engines was 1500rpm. Further, a couple iterations were carried out with the DOE in order to obtain the load limit in terms of fuel by targeting a brake power of 80 kW, which was the centre point value for the generator engines in similar dimensional vicinity as that of SICO. The mean point values from the base model were used to define the range for the DOE and a Latin Hypercube sampling method was adopted. The effect of the parameter on the BSFC, brake power, brake torque, volumetric efficiency along with the impact on brake and indicated efficiencies were studied in order to determine suitable values for the cylinder dimension.

These new dimensions were held constant for all the further evaluation except when mentioned otherwise.

6.2.2 Compression Ratio variation

As described in section 4.6, the compression ratio of the combustion cylinder if increased would result in greater efficiency. The CR is defined in GT-power in the crank-train object and can be individually specified for the cylinders. The value specified would be taken as the geometric compression ratio of the cylinder. A detailed cylinder model can be defined by specifying the piston head dimensions, TDC clearance height and crevice volume. Since the value specified would be geometric CR, apart from crevice volumes all the other values would be taken into consideration before it defines the CR. Based on this, iterations were done for evaluating the variation in the performance for values of CR while not compromising on the strength requirements of the cylinder due to high peak cylinder pressures.

The effective CR of the help cylinder is the summation of the clearance volume above the piston at TDC and the transfer port volume since the configuration of the final engine model taken into consideration is such that the transfer port and help cylinder are interconnected with just an orifice. There is a need for the smooth flow of the exhaust gases from the main cylinders to the help cylinder to avoid loss of the expansion energy due to expansion in the transfer port. This is minimized by setting the CR of the help cylinder to a sufficient large value such that the pressure in the help cylinder is similar when the transfer port is opened. This pressure matching of the main and the help cylinders at the opening of the transfer port results in greater utilization of the exhaust energy and thus leads to an increase in the efficiency of the engine.

Initial iterations were done where the transfer port and the help cylinder were interconnected with a poppet valve. This was to have a lower value of CR for the help cylinder while ensuring that the pressure in the main and help cylinders could still be matched. This method had a major disadvantage which was that the transfer port would have a pressure value equal to that when the valves are closed which would be a lower value than desired. In the consecutive cycle, when the transfer port opens, there would be expansion of the burnt gases within this dead volume leading to a lower efficient expansion process. It proved to be more efficient having an orifice connection between the help cylinder and the transfer port instead due to the reasons discussed previously and in addition reducing the amount of complexity and number of components. The configuration is shown in figure 6.6

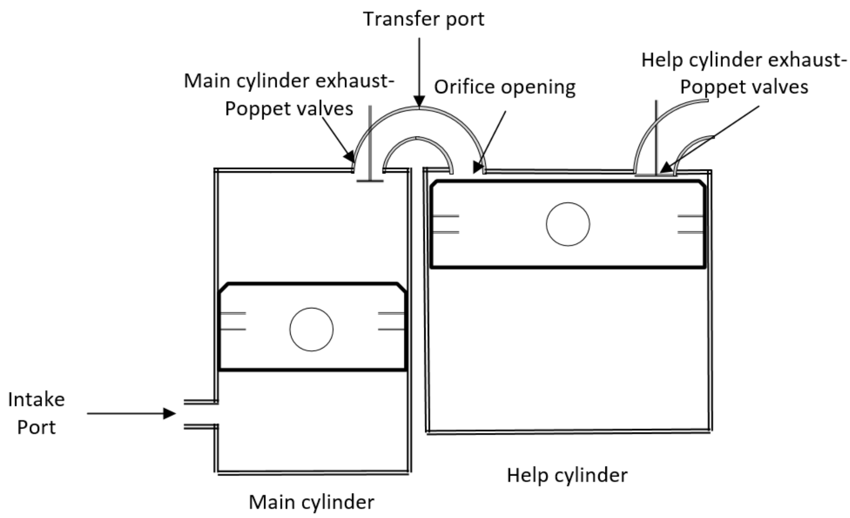


Figure 6.6: Cylinder configuration of SICO engine

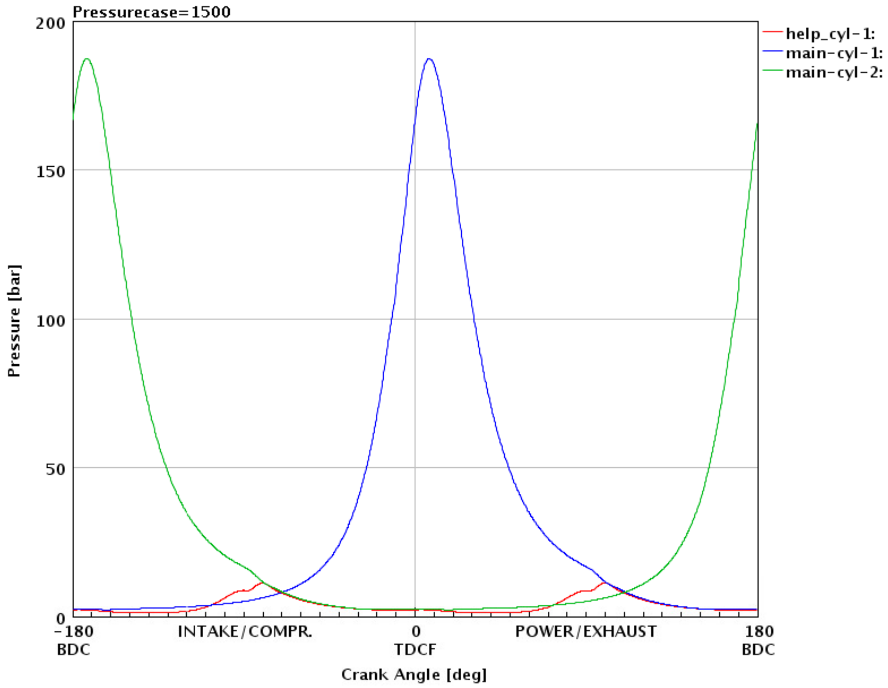


Figure 6.7: Pressure curves of the main and help cylinders

As seen in figure 6.7, the pressure in the first main cylinder (main-cyl-1) at midway of the power stroke matches that of the help cylinder. This is the event when the transfer port opens. After the opening of the transfer port, it is seen that the pressure in both the cylinder drops simultaneously due to pressure equalization in the closed system until the transfer port is closed. The CR of the help cylinder is decided in order to achieve a peak pressure same as that of the main cylinder during the opening of the transfer ports.

6.2.3 Friction modeling

The friction in GT-Power is defined using models mentioned in section 4.5. Since Chen Flynn model is relatively a simple model to define, it was decided to use the model to estimate the friction for the engine concept. The Chen Flynn friction model is defined by the equation 6.1

$$FMEP = FMEP_{constant} + A \cdot Peak\ Cyl\ Pressure + B \cdot S_p + C \cdot (S_p)^2 \tag{6.1}$$

where,
FMEP = Total friction in bar
$FMEP_{constant}$ = Constant part of equation based on load values in bar
A = Constant considering the peak cylinder pressure
B = Constant taking the speed of the piston into consideration
C = Constant considering the square of the piston speed

All the constants of the equation are determined based on experimental analysis. As seen in equation 6.1 apart from the $FMEP_{constant}$, the model takes into consideration the other parameters such as the piston speed and the peak cylinder pressure for the respective cylinder. For an engine, the FMEP of each of the cylinders is individually calculated and the summation of the individual FMEPs are later displayed. Engines using similar engine dimensions and design can use similar constants to estimate the friction to a good extent. For concept engines its quite difficult to arrive to the constants unless there are similar concepts that utilize the friction model based on experimental data. Since there is no experimental data for the SICO concept, the constants for the model cannot be found accurately. For the base engine model the constants of the Chen Flynn model were chosen from the GT-power example model for a six cylinder diesel engine.

Based on early simulation results of the friction estimation of the base engine model, it was noted that the amount of friction was notably high. This observation was made based on the way the Chen Flynn model sums up the friction of the help and main cylinders. The friction value is dependent on the constants used to define the Chen Flynn model and the constants chosen initially were not based on usage for the SICO concept rather from an example model. The constants for the model needed to be evaluated such that it captures the effect of the SICO concept's configuration. In addition to this, due to lower loads in the help cylinder, the construction of the help cylinder could be assumed to of lower strength requirements and fewer components such as fewer piston rings and smaller area subjected to rubbing to the cylinder.

In order to assess this, the friction for the cylinders were estimated using a sample model of GT-Power which is a more detailed model. This model estimates the friction due to the piston cylinder assembly and the crankshaft parts such as the bearings, flywheel, gears etc. The setup of the model is shown in figure 6.8

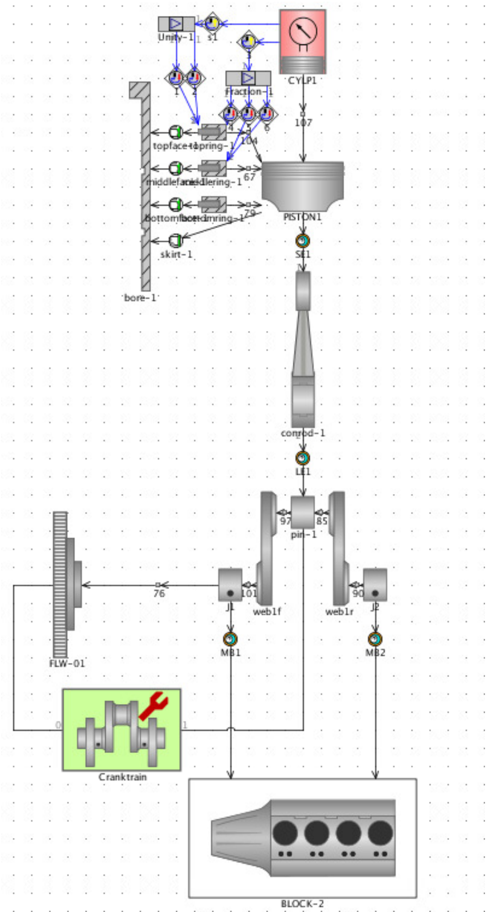


Figure 6.8: Full cylinder friction model
[20]

The variables that were defined were the bore and stroke sizes, the connecting rod length, the piston ring diameters based on the bore size, main rod bearing diameter. The other main inputs were the cylinder pressure and the speed of the engine. These pressure curves were defined based on simulation results of the SICO concept. The remaining variables were kept as the default values such as the materials, oil used, the sump temperature etc. This was done to arrive at the constants that can be used to define the Chen Flynn model as finally the Chen Flynn model was used with just variations in the constants.

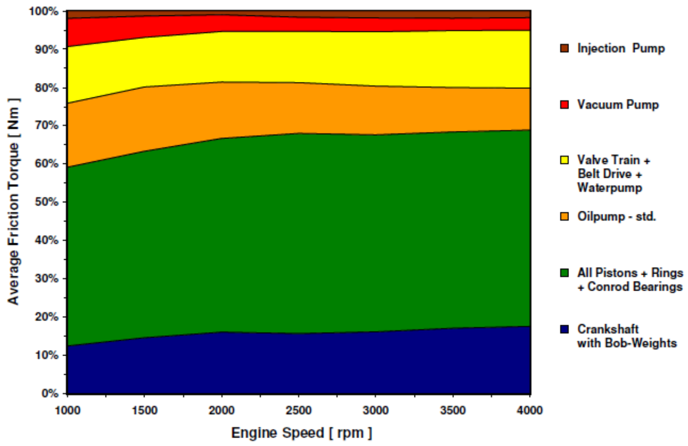


Figure 6.9: Friction teardown of DW Engine, Volvo Cars - overall distribution[21]

The friction was calculated based on these inputs. The model does not take into consideration the friction due to accessories such as the valve-train, coolant and oil pumps. Based on this, the summation of the friction of the three respective cylinders was calculated. Based on Heywood literature [2] and figure 6.9, an approximation of the friction contribution of the piston cylinder assembly and the crankshaft assembly accounts to a total of 60% of the total engine friction. The final friction value was the summation of this value and 40% of the total engine friction.

$$Friction_{total} = (2 \cdot Friction_{maincylinder}) + Friction_{helpcylinder} \tag{6.2}$$

$$Friction_{final} = \frac{Friction_{total}}{0.6} \tag{6.3}$$

The above calculations were performed for two operating points and based on the friction values obtained, the constants of the Chen Flynn model were varied in order to obtain similar friction values as that estimated based on equations 6.2 and 6.3

6.2.4 Scavenging model

GT-Power does a 1-D estimation of the gas exchange process unless a 3-D analysis is performed and variables are incorporated. It is reliable to do a CFD analysis to get a good approximation of the scavenging process in the cylinder. But due to time limitations, an alternative approach was used to define a scavenging model based on practical limitations as per Heywood [2] literature.

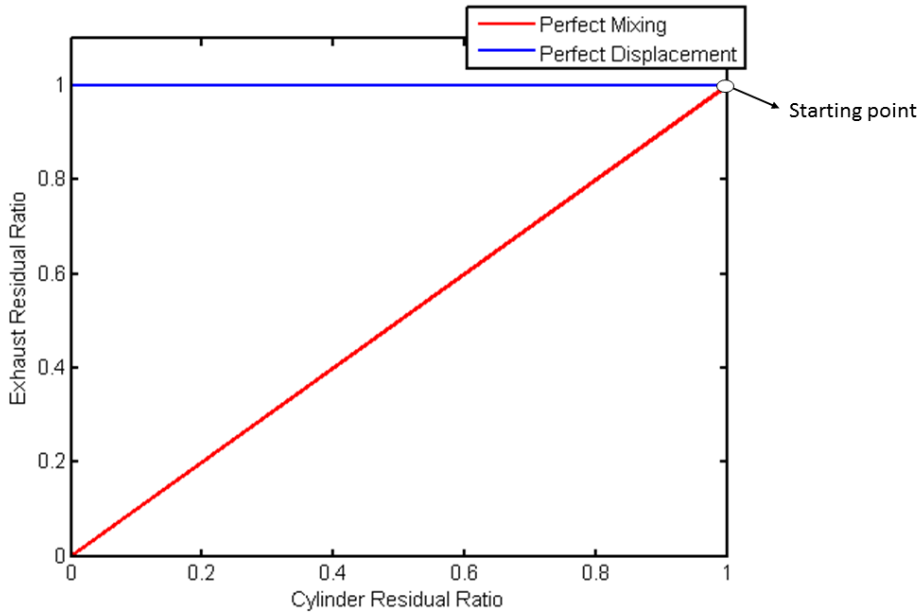


Figure 6.10: Scavenging curves for perfect displacement vs perfect mixing

The figure 6.10 compares the curve for a perfect displacement and a perfect mixing scenarios. The instantaneous gas exchange in the cylinder is governed by the scavenging curve defined. The definitions for the scavenging model is as per the GT-Suite manual. The cylinder residual ratio in the x-axis is the ratio of burned mass in the cylinder to the total mass in the cylinder. The exhaust gas residual ratio is the ratio of the burned mass exiting the cylinder to the total mass exiting the cylinder. In the figure 6.10, the point at cylinder residual ratio and exhaust residual ratios equal to 1 is the starting point, i.e, when the scavenging process begins. The blue curve representing perfect displacement shows that at all times the gases exiting the cylinder would be only the burnt gases and no fresh air. The red curve representing the perfect mixing curve shows that from the beginning of the scavenging event an equal ratio of burnt and fresh charge exits the cylinder.

It is desirable to have a perfect displacement curve since the residual gases are hot gases that cause an increase in the combustion temperature and so can reduce the thermal efficiency of the combustion process. Also higher temperatures are not favourable when it comes to NOx emissions. But in reality in a two stroke engine since the intake event is a very short event, there is a trade off when it comes to the amount of residual gases being trapped and the amount of fresh air being trapped. A setup aiming at a smaller percentage of residual gases would also lead to lesser amount of fresh charge being trapped in the cylinder.

Initial simulations showed that the scavenging of the main cylinders was affected by the back pressure due to the help cylinder. For effective removal of the residual gases, an exhaust with a lower pressure would be beneficial. To achieve this, an additional valve was introduced to the main cylinders which would act like a bypass for the exhaust. The valves are timed such that it opens for a short duration shortly after the main cylinder reaches BDC and help in the scavenging of the residual gases. The lift of this valve is kept short so as to have reasonable acceleration values. The duration is kept short so as to not have a greater loss in the amount of fresh charge being trapped. The valve and port timing events of the main cylinders are shown in figure 6.11

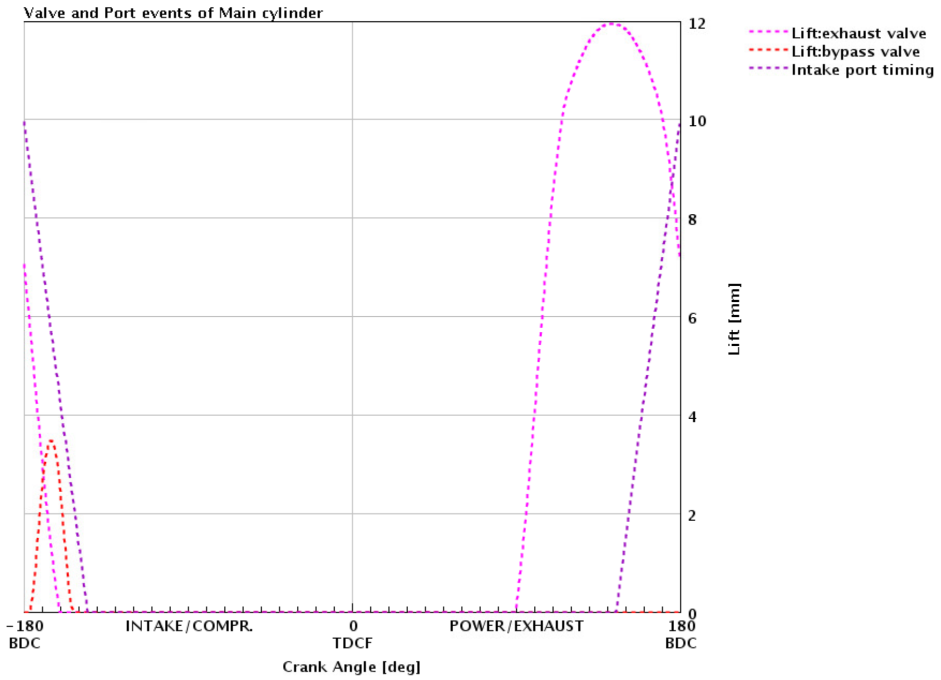


Figure 6.11: Valve and Port timings of the main cylinder

For the base engine model, to incorporate the scavenging process in the model, the scavenging curves from sample models were taken. For iterations in the advanced model, based on Heywood literature[2] and fig 4.5 the scavenging curve was plotted such that the purity ratio for respective delivery ratios were simulated in order to be similar to that for a uni-flow scavenging method. This was done by varying the delivery ratios for a single operating point and the amount of residual gases were checked based on this.

6.2.5 Main cylinder exhaust valve timing

As explained in section 4.1, the help cylinder is phased at 90 degree to the main cylinders during the event of the transfer port opening. This means that when a main cylinder is halfway down from TDC, the help cylinder would be at TDC when the transfer port opens. At this point the flow of gases into the help cylinder needs to be well planned in terms of timing such that the pressure peak would be quite close to the TDC of the help cylinder so as to get maximum benefit from the help cylinder.

Taking this into consideration, the valve timings of the exhaust valves of the main cylinders need to be designed for quick accelerations such that the pressure peak is still close to the TDC of the help cylinder. Initial iterations were based on having a dwell at the max lift point to facilitate gas exchange for small lifts without compromising the requirement for a quicker valve lift profile. Such a profile meant unrealistic decelerations at max lift point. Thus a lift profile was designed to facilitate this requirement which is shown in figure 6.11 represented by the pink curve (Lift:exhaust valve)

With this new profile, iterations were done for minute variations in valve timing and in the phasing time of the help cylinder to evaluate any improvement in the performance. However, iterations proved that the most benefit was found when the transfer port opening was timed to match the TDC timing of the help cylinder

6.2.6 Heat Transfer Effects

One of the major concerns in the development of internal combustion engines is to prevent the losses that occur through heat transfer between the working fluid and the cylinder walls. The study of heat insulation effects on the main cylinder would require in-depth study of the effect of insulation on the charging, combustion and their influence on the heat transfer and fuel consumption. Further, different modelling approaches may be required to accurately capture the insulation of the combustion cylinder. The Woschni GT [22] heat transfer model employed in the base model simulations may not completely capture the effects of insulation of the main cylinder. The heat transfer during combustion increases with high surface temperatures [11]. It has been shown that the constants in the classical Woschni heat transfer model [10] have to be altered in order to effectively capture the insulation effects [12]. However, the main cylinder is also evaluated in order to analyse these effects. Due to the time limitation involved in the thesis, the alteration required in the heat transfer model was not evaluated. The help cylinder forms an interesting case for evaluation as it does not involve any combustion. The effects of the insulation of the help cylinder were studied with the normal Woschni GT heat transfer model.

The GT-Power cylinder objects allows for description of the three wall temperatures, namely head temperature, piston temperature and the cylinder temperature. Increasing the temperature leads to reduced temperature difference between the cylinder wall and the gas and thereby reduces the heat transfer.The wall temperatures were modified to simulate the insulation effects and is as given in table 6.2.Initially, the values of the three temperature values were taken from the standard GT-Power models and used for the base model simulations. These are the average values for full load conditions and effectively represent the usual upper limit of the temperature considered for these regions [22]. Subsequently, a sweep of each of the temperatures was carried out between 400 K and 1000 K in order to evaluate the effect of each parameter on the efficiency of the system. Finally, a value of 800 K and 700K was chosen for the head and the piston temperature respectively in order to evaluate the net effect of insulation. Further, the cylinder temperature is taken as 450 K based on its effect on the efficiencies presented in section 7.5. Very high surface temperatures of about 1500 K are considered for the head and the piston for the main cylinders while the cylinder temperature is agian maintained at 450 K. This helps in understanding the effect of insulation that is predicted by GT-

Power.

Table 6.2: Help Cylinder Wall Temperatures.

Parameter	Base Model	Modified Values
Head Temperature (K)	550	800
Piston Temperature (K)	590	700
Cylinder Temperature (K)	450	450

6.3 Sub-Concepts Evaluation

Along with evaluating the subsystems that were defined in sections 6.1 and 6.2, sub-concepts mentioned in the literature review in chapter 5 were modeled and evaluated to validate the robustness of the SICO concept. The case setup for such simulations were done keeping in mind to ensure that the concepts modeled for comparison were by themselves robust to an extent to give an unbiased comparison. These were done by optimizing the sub concepts to some extent apart from performing simulations that would show the best of each the concept. Nevertheless, due to time limitations the sub-concepts weren't modeled to be in the optimum stage and would surely have a wider possibility of improvement in performance. The sub-concepts were mainly modeled as modifications to the SICO model rather than a new model. The simulations were done for a 1500 rpm engine, the load depending on the limitations of the concept.

6.3.1 Main Cylinder Only Model

The main argument of the concept is that the extra expansion in the help cylinder helps in recovering the energy that is otherwise lost in the form of exhaust. In order to evaluate the actual advantage of the help cylinder, a model without the help cylinder was simulated. The help cylinder was removed and the exhaust from the main cylinders was directed towards an end environment via a manifold. The exhaust valve timings were adjusted in order to adapt the gas exchange process for the new setup. The rest of the components were kept consistent between the two models. Both the models , i.e with and without the help cylinders, were run at 1500 rpm and at high loads.

6.3.2 Zero degree phasing (ZDP) model

As explained in section 3.4, the main and the help cylinders are phased at 90 degrees with respect to each other and the help cylinder runs at twice the speed of the main cylinders. Additionally, the main cylinders are phased at 180 degrees with respect to each other. Running at twice the speed would mean that there would be greater frictional losses and since the volume of the help cylinder is around 2.4 times the volume of the main cylinders,the friction contribution in comparison with main cylinders would be greater. Considering these shortcomings of the SICO configuration, an evaluation of a concept with the main cylinder in phase (0 degree phase) was planned referred to as the ZDP model. Furthermore, in this case the help cylinder could be designed to run at the same speed as that of the other cylinders. This would reduce the amount of friction to a good extent. But in order to achieve a comparable performance, the size of the help cylinder would have to be greater adding to the friction value.

For the ZDP model to work and have a good range of torque limits, some optimizations that are required were-

- The valve timings were varied to better suit the configuration.
- Since the help cylinder runs at the same speed of the main cylinder, the compressor map was altered to better suit the needs of the ZDP concept.
- As the burnt gases from the main cylinders would be coming simultaneously from both the main cylinders to the help cylinder, the amount of gas energy would require a larger help cylinder in order to target lower fuel consumption.

6.3.3 Compressor Cylinder (CC) model

Based on the concept initially conceived by Per Arne Sigurdsson the inventor of the SICO concept, the charging of air into the intake system to be done using a compression cylinder rather than a radial compressor. This cylinder would also be running at twice the speed of the main cylinders in order to facilitate charged induction for both the cylinders. Based on other concepts like the Double Compression Expansion Engine[4], using a compression cylinder would facilitate a large increase in the indicated efficiency of the engine by facilitating a higher boosted charge (7-16 bar in the case of DCEE) and so more mass of air being trapped. The DCEE concept have achieved an indicated efficiency of 60%. Based on these results, an attempt to use a cylinder for compression was modelled referred to as the CC model for comparison with the SICO model.

The major challenges in the CC model concept were:

- A charging cylinder would mean that the cylinder capacity needs to be large enough in order to facilitate charging and ensure a good volumetric efficiency.
- Friction would be greater since this would be a large cylinder having a contribution to the overall friction. Additionally, it would be running at twice the speed similar to the help cylinder resulting in greater friction contribution.
- The timings of the valves for this cylinder need to be adjusted to ensure a low pressure in the intake system for allowing the charge cylinder to induct air.

The layout of each of the charging systems is shown in figures 6.12 and 6.13

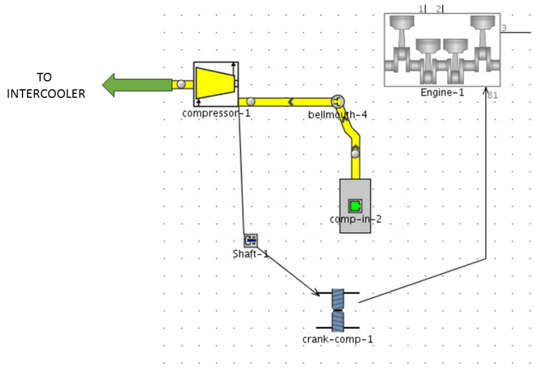


Figure 6.12: Charged induction using a radial compressor

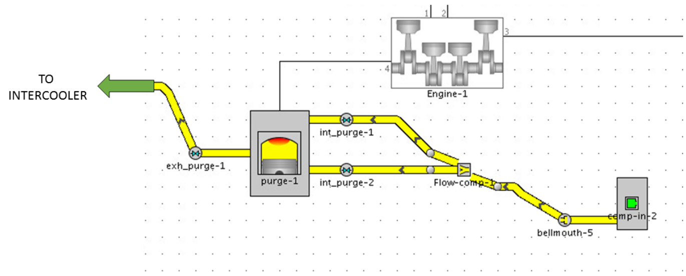


Figure 6.13: Charged induction using compression cylinder

For the compressor cylinder to work and produce energy in a similar range as that of the compressor, the energy consumed by compression and friction needs to be in the range as of the total energy consumed by the compressor for a particular mass flow rate. This can be better described by the equations below:

$$IMEP_{comp\ cyl} = \int (P \cdot dV) \tag{6.4}$$

$$Energy\ consumed\ for\ compression\ (E_{compression}) = IMEP_{comp\ cyl} \cdot V_{disp\ comp\ cyl} \tag{6.5}$$

$$Total\ energy\ consumed\ by\ comp\ cyl\ (E_{comp\ cyl}) = E_{compression} + E_{friction} \tag{6.6}$$

Iterations were done so as to reduce the value of $E_{comp\ cyl}$ without compromising the mass flow rate while still having a good pressure ratio to help the scavenging process.

6.3.4 Five stroke model (5S)

Based on the paper on the five stroke engine concept[8], a simulation model was built that was similar in terms of cylinder phasing as that for the five stroke engine. In other terms, the SICO concept was modified for the main cylinders to work on a four-stroke cycle operation with an extra expansion cylinder. The main cylinders were phased at 360 degrees with respect to each other while the help cylinder and the main cylinder phased at 180 degrees to each other. Several advantages and disadvantages were considered while making this model. The concept involves a four-stroke operating cycle for the main cylinders and the help cylinder working as a two stroke cycle operation. This was seen to produce better volumetric efficiency since there are distinct intake and exhaust strokes avoiding the drawbacks of scavenging such as higher residual gases. AFR ratios could be higher than that employed for the SICO concept. Friction behaviour would be different too since there would be distinct pumping cycles as compared to that of the SICO concept.

Simulations were done to check the max load capacity taking into account the max amount of fresh charge trapped in the cylinder. Since in a two stroke cycle the power stroke is once every revolution, the total power output would be twice considering same input conditions. As the limit for higher loads is mainly governed by the AFR in a diesel engine, this shortcoming of the five stroke concept could be overcome as compared to the SICO concept.

7

Results

7.1 Base Model Results

The base model was simulated for 1500 rpm and at high load in order to match the necessities of the application, as mentioned in section 5.4. At these conditions, the engine almost runs at its air fuel ratio (AFR) limit of about $\lambda = 1.1$.The important results are listed in table 7.1.

Table 7.1: Base Model Results

Parameter	Base Model Values
Indicated Efficiency (%)	50.8
Brake Torque (N-m)	533
Brake Efficiency (%)	35.45
BMEP (bar)	5.94
Brake Power (kW)	83.8

It can be observed that the engine operates at a low BMEP value even at the higher loads.The overall displacement of the engine is about 5.64L and is poorly utilized when compared to the amount of power developed. A deeper investigation on the dimension and the corresponding variation of the different valve timings is necessary to evaluate the possible improvements in the BMEP and the efficiencies. Further, table 7.1 gives an indication of the fuel energy break up for the above mentioned. The two competing parameters for the given case are the indicated power and the friction power. It can be observed that an high indicated efficiency is obtained and this may be over-estimated due to fact that an ideal scavenging situation is considered in this model as mentioned in section 6.1. In reality, the scavenging is influenced by several factors such as the cylinder dimensions, the cylinder geometry, the type of scavenging employed etc. Further, the friction is also another parameter that has to be evaluated with more focus. Around 11% of the available power is lost as friction in the base model. However, the unique nature of operation where the help cylinder is running at twice the speed as that of the main cylinder, may need to be considered with respect to the friction model in order to thoroughly validate the above results.

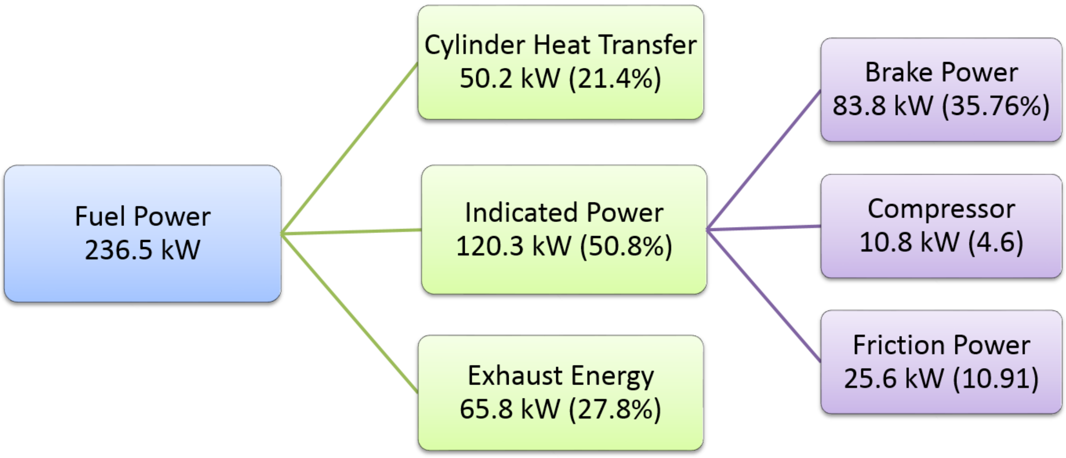


Figure 7.1: Fuel Energy Break up

The results of the above mentioned parameters along with further evaluation of the heat transfer and the concepts are presented n the following sections. A final SICO model is obtained at the end of the advance model evaluation and the results are presented.

7.2 Cylinder Dimension

The cylinder dimension values were optimized by conducting a Design of Experiments for the bore and stroke of the main and help cylinder under the conditions mentioned in section 6.2.1. The engine speed was kept at 1500 rpm with a fuel mass of 90 mg. The base model was used in order to determine appropriate values for the cylinder dimension and their effects on the performance of the engine was studied. Based on the study, a new set of dimensions for the help and the main cylinders were chosen and are as shown in table 7.2.

Table 7.2: Cylinder Dimensions

Parameter	Base Model Dimensions (mm)	Modified Dimensions (mm)
Main Cylinder Bore	105	85
Main Cylinder Stroke	120	115
Help Cylinder Bore	180	163
Help Cylinder Stroke	140	88

The stroke length of the help cylinder is reduced which results in a reduced frictional losses in the help cylinder. Also, the bore is slightly reduced compared to the original value. Though this results in an increased FMEP due to higher peak cylinder pressures, the net displacement volume is reduced. Thus the calculated friction reduces resulting in better brake efficiencies while having a smaller cylinder size.

Further, the combined displacement volume of the help and the main cylinders is reduced from 5.64L to 3.14 L and thus a significant increase in the BMEP can be observed as shown in figure 7.2. A better brake efficiency is obtained owing to reduction in the losses through friction due to the reduced dimensions. The decrease in the indicated efficiency can be attributed to the increased in-cylinder pressure and temperature leading to higher heat transfer. However, the absolute values of efficiencies are still dependent on the friction,scavenging, compression ratio and several other factors. Thus these graphs should only be considered in terms of comparison for the new dimensions over the previous values.

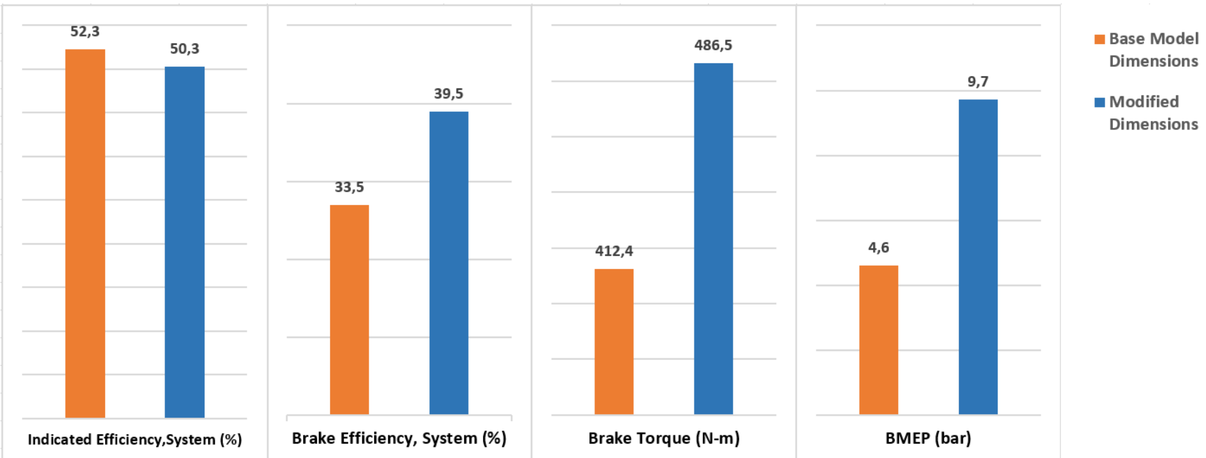
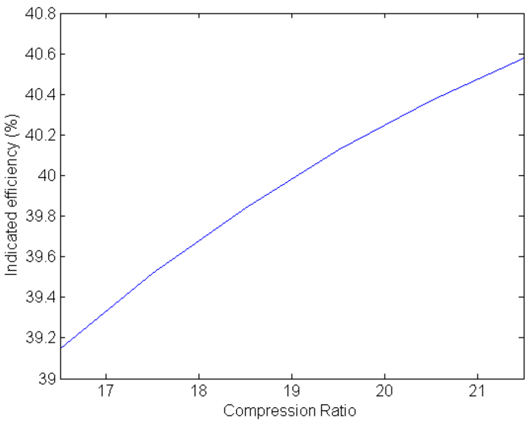


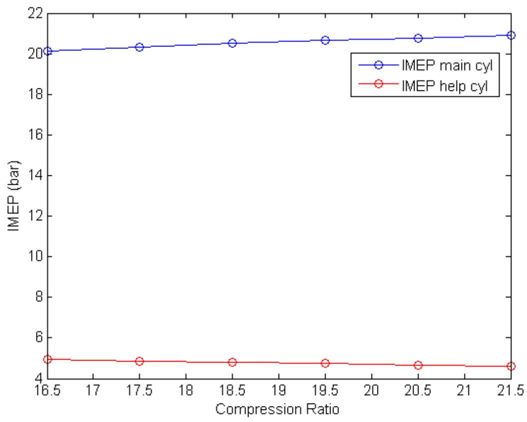
Figure 7.2: Effect of main and help cylinder dimensional changes

7.3 Compression Ratio

Based on the assumptions in sections 4.6 and 6.2.2, the compression ratio of the main cylinders were varied within the range of 16.5 to 21.5. The results of the IMEP and the indicated efficiency of the main cylinders and the brake efficiency of the engine against the variation of the compression ratio are shown in figure:



(a) Indicated efficiency vs CR - Main cylinders



(b) IMEP vs CR for Main and Help cylinders

Figure 7.3: Main cylinder compression ratio variation effect

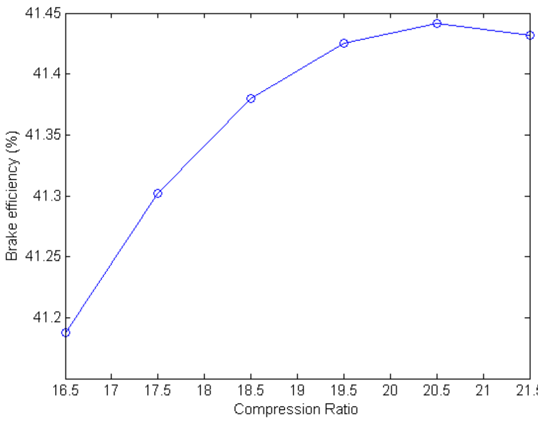
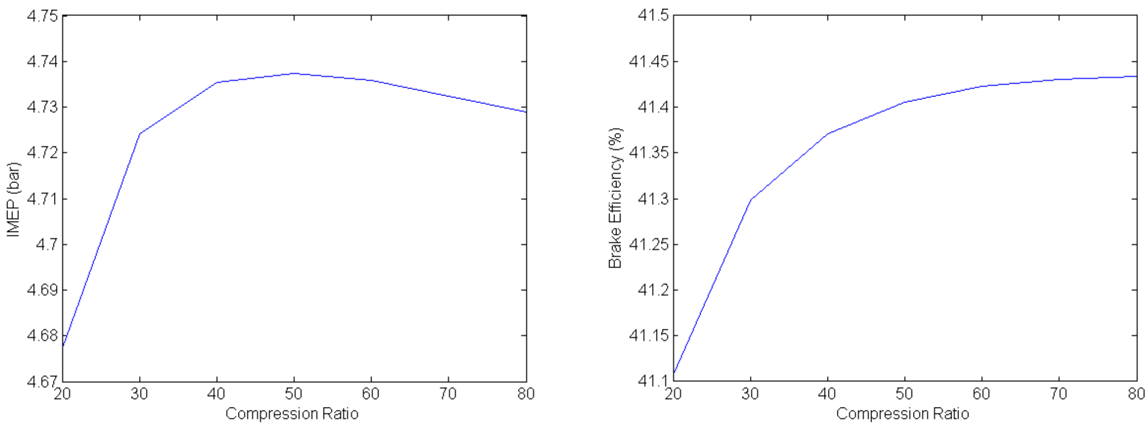


Figure 7.4: Engine brake efficiency vs CR

As seen an increase in the compression ratio results in an increase in the indicated efficiency and IMEP of the combustion cylinders. This trend was seen for CR values as high as 24. But after a CR value of 20.5 the IMEP of the help cylinder dropped. This was a result of the balance between the summation of the compression work and expansion work of both the cylinders was greater for the lower CR than the higher CR values. The overall increase in engine efficiency due to the increase in CR was very less significant after CR=19.5. In addition to this, at higher CR the peak cylinder pressure of the cylinder is also quite high(around 185 bar at max load) which would mean that the cylinder head would have higher strength requirements. Based on these results, the CR of the main cylinder was set to 19.5.

The CR of the help cylinder was also iterated based on the section 6.2.2. It was observed that the smaller the volume of the transfer port, the higher was the performance. Since this volume is dependent on the diameter of the valve connecting the transfer port to the help cylinder, the valve diameter was reduced until the the opening was wide enough to prevent any large resistances to the flow. Based on this, the transfer port was modeled to be a U shaped round bar with a constant cross section of 34mm diameter. The volume of the transfer port was calculated to be $69.88mm^3$. The transfer port was defined to be externally cooled to 600K. This resulted in an increase of the heat transfer losses but was negligible when compared to the total engine efficiency. Additionally owing to the U shape of the transfer port, GT does a 1-D estimation of the flow rather than a 3D estimation. Nevertheless, it does estimate the pressure drop between the ends which is observed to be 2.75 kPa for a high load case.

Based on this, the geometric CR of the help cylinder was varied for a range of 20 to 80 to check its effect on the performance of the engine. The effective CR of the help cylinder considering the effect of the transfer port would then be ranging from 12.02-20.72.



(a) IMEP vs CR - Help cylinder **(b)** η_{brake} vs CR

Figure 7.5: Help cylinder compression ratio variation effect

As seen from the above figures, an increase in the compression ratio of the help cylinder increases the IMEP of the cylinder until a value of CR=50. After this the IMEP of the cylinder starts to drop. The increase in the IMEP due to increase in CR was due to the greater pressure peaks achieved by the greater CR and thus more work is extracted. But after CR=50, the amount of work obtained from expansion of the gases in the cylinder is out balanced amount of work done to compress the gases during the compression stroke. This includes heat transfer losses that increase with increase in temperature at higher CRs.

The figure 7.5b shows an increase in engine brake efficiency even at CR greater than 50. Though the help cylinder extracts less work from the expanding gases at CR > 50, with increase in the CR the in-cylinder pressure in the help cylinder reaches a lower value after expansion/exhaust and so during the intake stroke of the help cylinder, the back pressure generated is lower and so offers less resistance for the exhaust gases from the main cylinder to flow into the help cylinder. With more burnt gases being transferred from the main cylinders to the help cylinders, the scavenging of the main cylinders is also improved. Thus the overall indicated power of the engine improves hence increasing the brake efficiency.

Although an increase in the CR of the help cylinder results in an increase in the brake efficiency, the increase after CR=60 is very small. Moreover, a CR value of 60 meant that the clearance above the piston at TDC would be 1.45mm which would be required to accommodate the valve. Thus for the final engine model, the CR of the help cylinder was defined as 60.

7.4 Friction

As stated in section 6.2.3, the Chen Flynn model was used to define the friction of the engine. For the base engine model, the constants for the Chen Flynn model were taken from the sample model as a basis for friction estimation considering lack of initial data and that the values were based on experimental analysis. These constants did not have a large variation from the constants used in other sample models of GT-Power such as a four litre gasoline engine and diesel engine. It was assumed as a start of approximation.

Using the full friction example model of GT as explained in section 6.2.3, the friction for the main cylinder and the help cylinder was individually found. The data as input for the full friction model was for an operating point at 1000rpm and 1600rpm for 90mg fuel injection(load case). The pressure curve obtained for the respective cylinders were the inputs to the full friction model and based on these the results for the friction obtained are given below-

Table 7.3: Friction calculation

	1000rpm	1600rpm
$InitialFriction_{maincylinder}$	0.65kW	1.36kW
$InitialFriction_{helpcylinder}$	1.06kW	2.90kW

From the values obtained above, the friction was calculated for the 1000 rpm case using equations and method described in section 6.2.3, as shown below-

$$Friction_{total} = (2 \cdot Friction_{maincylinder}) + Friction_{helpcylinder} \tag{7.1}$$

$$Friction_{total} = (2 \cdot 0.65) + 1.06 = 2.36kW \tag{7.2}$$

$$Friction_{final} = \frac{Friction_{total}}{0.6} \tag{7.3}$$

The final friction value accounts the 40% friction contribution from the accessories as explained in section 6.2.3

$$Friction_{final} = \frac{2.36}{0.6} = 3.93kW \tag{7.4}$$

Similarly, for the 1600rpm case the $Friction_{final}$ was calculated to be as 9.37kW. The final friction values obtained using the Chen Flynn model for the constants used initially and the friction values obtained based on these calculations are listed below

Table 7.4: Final friction values

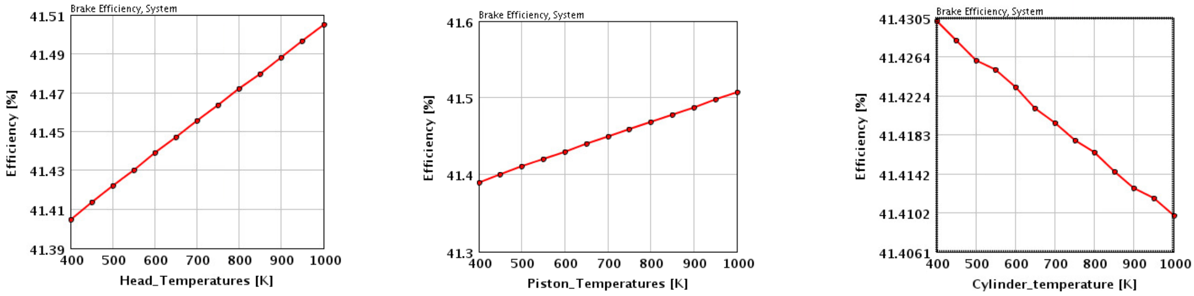
	1000rpm	1600rpm
Initial total friction	7.09kW	14.169kW
$Friction_{final}$ based on full friction model	3.93kW	9.36kW

The difference of the friction calculated using the initial constants for the base model and that using the full friciton model was approximately 1.8 times for the 1000 rpm case and 1.5 times for the 1600 rpm case. Since the final model was optimized for 1500rpm case, the friction difference taken into account was 1.5 times. Thus, the base constants of the Chen Flynn model were divided by 1.5. Using the new constants, the friction power obtained for 1500rpm case for the same load case is 9.03kW. Based on a linear interpolation of the curve joining the points of the calculated friction value for 1000rpm and 1600rpm, the friction at 1500rpm would be 8.5kW. As the factor for division of the constants was chosen as 1.5, the final friction estimation by the simulation would be higher than that calculated. This is a safety margin to ensure that the friction estimation at higher speeds could still be captured.

The friction at 1500rpm accounts to 4.6% loss of engine efficiency which is excluding the compressor work.

7.5 Heat Transfer Effects

The changes in the brake thermal efficiency with variations in the head , piston and cylinder temperature of the help cylinder is as presented in figure 7.6. Increase in the head and the piston temperature relates to a very small increase in the brake efficiency and is as represented by figures 7.6a and 7.6b. However, an increase in the cylinder temperature results in an opposing trend and the brake efficiency decreases with increase in the temperatures and is represented in figure . This is caused due to the heating of the inflowing air resulting in reduced volumetric efficiency of the help cylinder which offsets the gains obtained due to reduced heat transfer.



(a) Head Temperatures

(b) Piston Temperatures

(c) Cylinder Temperatures

Figure 7.6: Help Cylinder wall temperature variation effect on brake efficiency

A simulation to evaluate the combined insulation effect of all the three wall temperatures of the help cylinder on the brake efficiency was carried and the results obtained are as shown in table 7.5. The head temperature was maintained at 800 K, while the piston and cylinder temperature were maintained at 700K and 450 K respectively as explained in section 6.2.6. It can be observed that only a small improvement can be seen in the brake efficiency. The heat transfer from the help cylinder reduces by about 23% when compared to the model without insulation. However, this energy is mostly used up to increase the exhaust temperatures and thus is lost as the sensible energy from the exhaust. Figure 7.7 represents the distribution of energy for both the cases. The net energy lost through the exhaust and the heat transfer in both the cases almost remains the same. The exhaust energy correspondingly increases with a variation in the heat transfer and indicated that the actual energy available for conversion into work in the help cylinder is very small. Thus the overall brake efficiency remains the same.

Table 7.5: Effect of Insulation of help cylinder

Parameter	Without Insulation	With Insulation
BSFC (g/kWh)	202.08	201.78
Cylinder Heat Transfer(kW)	5.52	4.26
Exhaust Temperature (K)	797	813
Brake Efficiency (%)	41.42	41.48

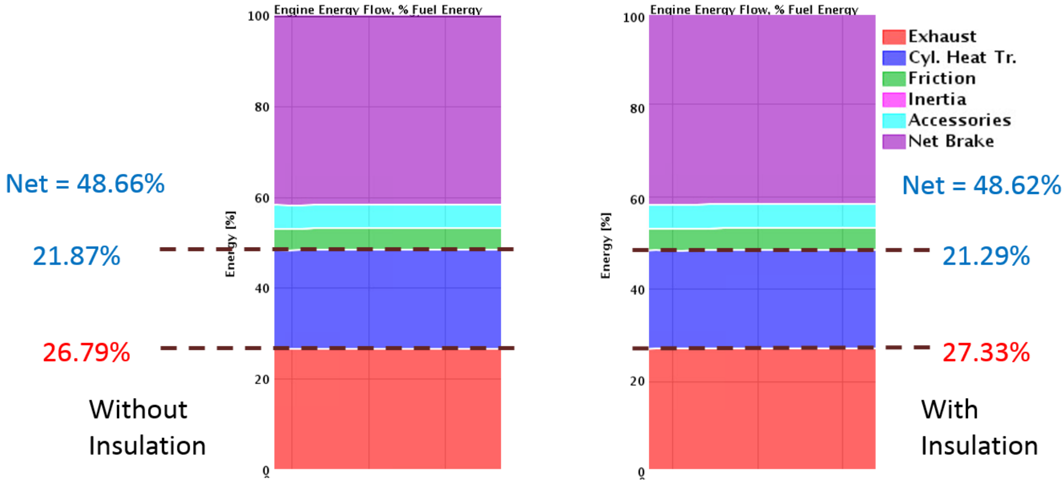


Figure 7.7: Energy distribution with and without insulation

In order to assess the results with respect to the main cylinder, the wall temperatures of the main cylinder were also modified to the temperatures as mentioned above. Further, the wall temperatures are raised to very high values of 1500 K and the effects on the engine performance is evaluated. The significant parameters are as given in table 7.6. It can be observed that the brake efficiency improves by about 3% when the head and piston temperatures are maintained at 1500 K. However, experimental results have shown that the fuel consumption increases when the surface temperatures are high [11] [12].

Table 7.6: Effect of Insulation of Main cylinders

Parameter	Without Insulation	With Insulation
BSFC (g/kWh)	202.93	189.46
Cylinder Heat Transfer(kW)	32.52	17.8
Exhaust Temperature (K)	839	929
Brake Efficiecny (%)	41.25	44.08

7.6 Scavenging

The scavenging curve chosen for the base engine model is shown below -

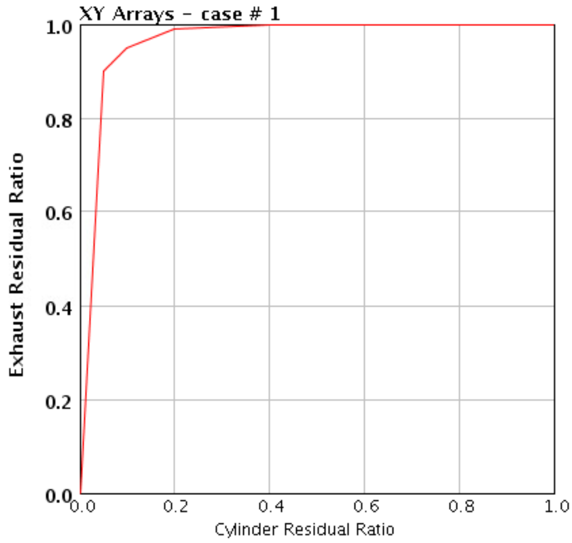


Figure 7.8: Scavenging curve used for base model taken from GT-Suite Examples [20]

As seen above and based on the section 6.2.4, the curve in figure 7.8 is more idealistic where the scavenging results show high purity and high trapping. This is not the case in a real engine and thus a new curve was plotted in order to obtain values close to Heywood literature as explained in sections 4.3 and 6.2.4. The curve chosen is shown below -

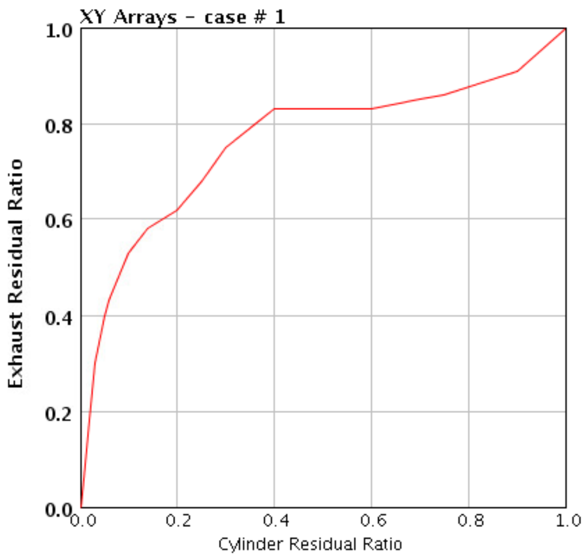


Figure 7.9: New scavenging curve defined

The curve defined above was based on checking the residual gases left in the cylinder by varying the delivery ratios of the air intake. This is compared to the residual gas percentage obtained using the scavenging curve shown in figure 7.8.

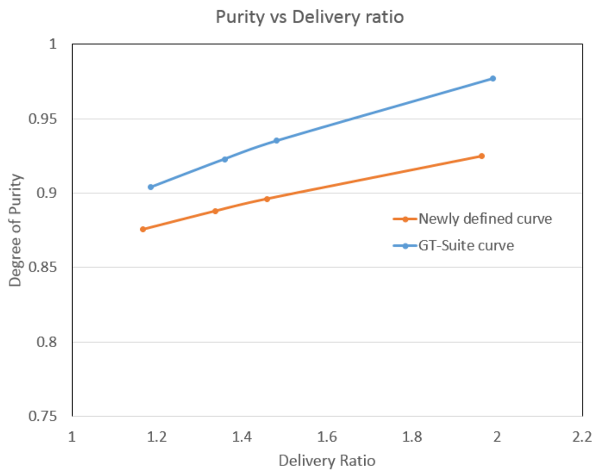


Figure 7.10: Purity vs Delivery ratio for the GT scavenging curve and for the new curve

From the figure above it can be seen that the newly defined scavenging curve has a more realistic estimation of the scavenging and this was used in the final advanced model to define the scavenging process. The new curve when in comparison with that estimated in literature as seen in figure 4.5 for a uni-flow scavenging mechanism, is more in coherence than the GT-Suite scavenging curve.

7.7 Main Cylinder Only Model

The effect of the help cylinder is analyzed as mentioned in section 6.3.1. The model was created and a few iterations were carried out in order to adjust the valve timings as per the requirements of the model. A comparison of some of the important parameters between the model with the help cylinder and the Main Cylinder Only (MCO) model are as shown in figure 7.11.

The MCO model is evaluated at an fuel level of 85 mg beyond which the engine runs below a $\lambda = 1.1$. Thus, the SICO diesel model is also simulated at a fuel level of 85 mg , which results in $\lambda = 1.1$, and also at it its peak point where $\lambda = 1.02$. Figure 7.11 gives a comparison of the MCO model and the SICO concept for the above mentioned lambda values.

It can be observed that the help cylinder causes a significant increase in the indicated and brake efficiencies. When running at $\lambda = 1.1$, the help cylinder was found to increase the indicated power by about 30 % leading to an improvement in the indicated efficiency by about 10.5% . However, the big size of the help cylinder coupled with its higher speed increases the friction to twice its original value and thus leads to an effective increase of the brake efficiency by about 8%.

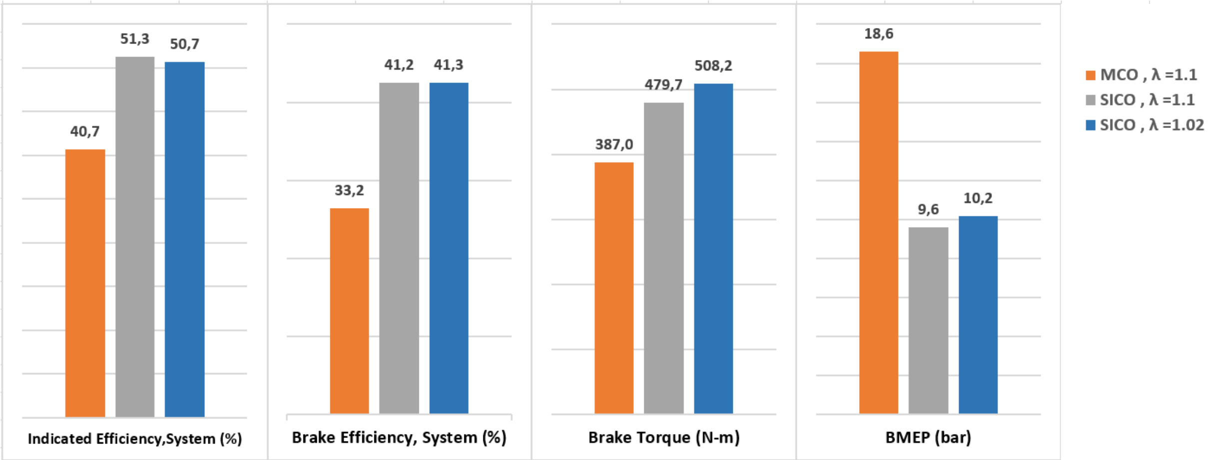


Figure 7.11: Performance comparison : Main Cylinder Only Model vs SICO

7.8 Zero degree phasing model

For evaluating the zero degree phasing of the main cylinders, the model was built as per section 6.3.2. As stated, the requirements for the model need to be met for it to work and be comparable to the SICO concept. The 'zero degree phasing'(ZDP) model had a few small changes such as the timing of the transfer port valve, the help cylinder dimensions and the compressor map. Few iterations were done in order to have a slightly higher average pressure ratio of the compressor for the ZDP model in comparison to the SICO by around 10-20% while moderating the increase of the power requirement of the compressor. The valve timings were varied keeping in mind that the intake valve timings of the main cylinders could not be varied owing to the two stroke cycle gas exchange requirements. Thus it was found difficult to phase the main and help cylinders to 180 degree which could allow better utilization of the expansion process. Rather the phasing was maintained at 90 degrees which showed the best efficiency when compared to the rest of the cases owing to the scavenging process. The cylinder bore of the help cylinder as mentioned was increased to improve performance of the engine. The comparison between the help cylinders of the ZDP model and the SICO model are shown in table7.7

Table 7.7: Help cylinder dimensions used in the SICO and ZDP model

	SICO	ZDP
Bore	163mm	230mm
Stroke	88mm	88mm
Displacement Volume	1.84L	3.63L
Compression Ratio	60	24

As seen the dimensions of the cylinder size for the ZDP model was chosen to be 230mm.

A lower value would reduce the fuel consumption to a small extent until it is outbalanced by the reduction in the indicated efficiency of the whole engine. A greater value reduces brake efficiency since friction increases but a slight increase in indicated efficiency can be observed from the results. The CR value was chosen so as to ensure a pressure matching a high loads.

Owing to the problems of lower trapped air in the ZDP model, the amount of fuel injected had to be limited in the ZDP model to 60mg of fuel. Thus for unbiased comparison, the SICO model was simulated for 60mg fuel($\lambda = 1.76$) as well as 90mg($\lambda = 1.1$) fuel cases. The results comparison are shown below-

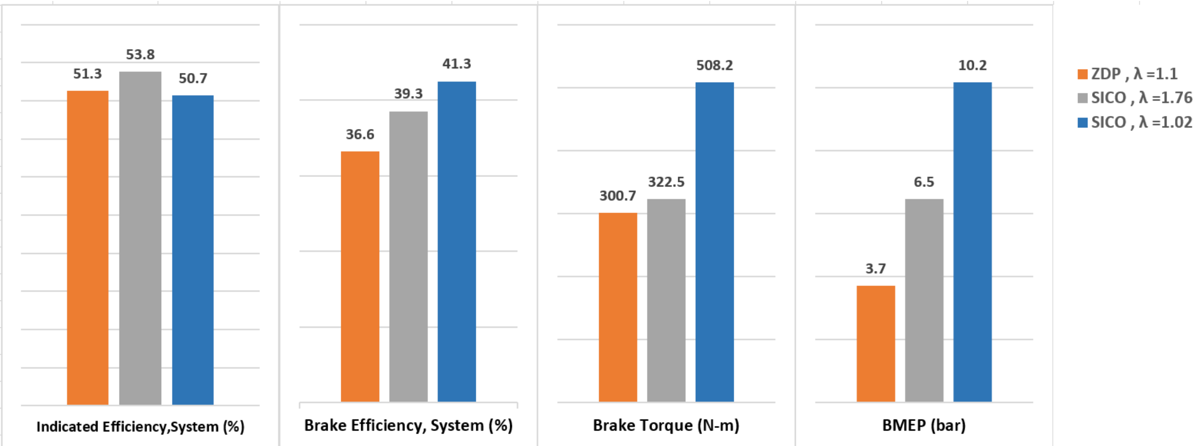


Figure 7.12: Performance comparison : Zero Degree Phase model vs SICO

From the figure 7.12 it can be noted that for the case with equal fuel energy as input to the system, the indicated efficiency of the SICO model was found to be greater than that of the ZDP model. This is due to the better utilization of the exhaust energy in the case of the SICO model as compared to the ZDP model. Though the friction value in terms of FMEP was 40% lower in the case of the ZDP model compared to the SICO model, the value of the friction in Kilowatt was found to be not differing a lot owing to the larger dimensions of the ZDP model.

Though the difference in brake efficiency among the two concepts is significantly low, in terms of size utilization, the SICO concept has the upper-hand which can be inferred from the BMEP values. The SICO model has a volume that is 35% smaller than the ZDP model. Additionally due to higher air trapping capabilities of the SICO model, the concept when simulated at its AFR limit showed a peak load brake efficiency of 41.3% which is much more than the ZDP model. Hence the max load capabilities of the SICO model are greater for the same charged induction of air.

7.9 Compressor Cylinder model

The compressor cylinder for the 'Compressor Cylinder model' (CC) was designed based on the constraints and equations mentioned in the section 6.3.3. The major criteria that influenced the performance of the CC model was the size of the compressor cylinder. Based on simulation results, these are the values of the compressor cylinder.

Table 7.8: Compressor cylinder dimensions used in the CC model

Bore	275mm
Stroke	100mm
Displacement Volume	5.94L
Compression Ratio	2.6
Cylinder speed	3000rpm (twice of main cylinders)

The volume of this cylinder is large in order to allow enough air intake into the system. In spite of this, the volumetric efficiency of the cylinder was around 50% and it is difficult to achieve a higher number as there is a very small window when the pressure in the compressor cylinder is below atmosphere to allow air intake. If the intake event of the compressor cylinder overlaps with the exhaust of the compression cylinder, a back flow is observed due to the atmosphere being 1bar and for the intake of fresh air into the main cylinders, a pressure of above 2 bars is required to facilitate scavenging. Thus the trapped air in the compressor cylinder is lesser than required. Owing to the lower trapped mass, the AFR restricts the maximum amount of fuel injected to 60mg so as to have a lean burn(λ>1) for higher indicated efficiency. With an increase in compression ratio the amount of air intake could be increased but then overall brake efficiency of the engine drops. Thus the CR was kept as 2.6.

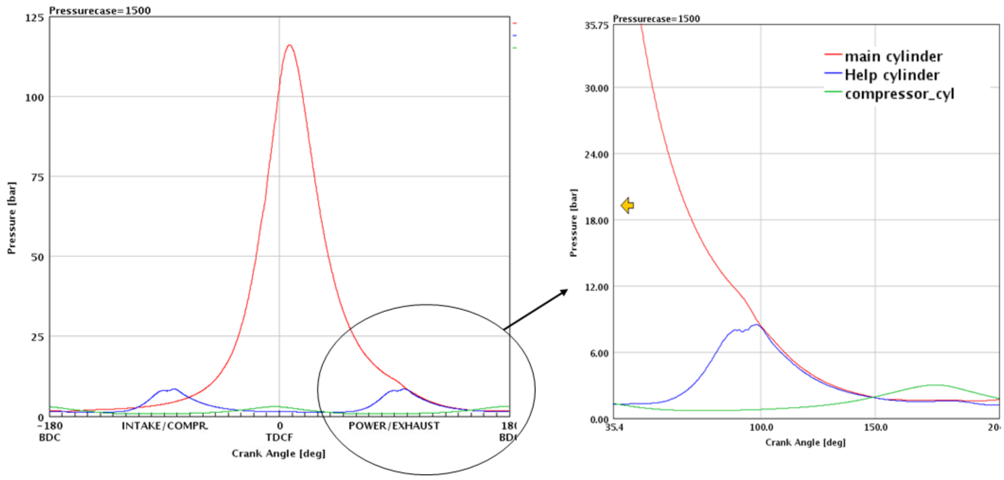


Figure 7.13: Pressure curves of the compressor, main, and help cylinders

Figure 7.13 shows the pressure curves in the compressor, main and help cylinders for a single revolution of the crank train. The pressure equalisation that happens in the main and help cylinders are similar to that what happens in the SICO concept. The green line in the figure 7.13 represents the addition of the compressor cylinder. It can be seen that when the intake port of the main cylinder opens (145 CAD after TDC), the pressure of the compressor cylinder and the respective main cylinders do not equalize quickly. Rather it equalizes at around 208 CAD after TDC which is a few degrees before the intake port of the main cylinders close. This delay is attributed to the volume of the inter cooler and the volume of the intake ports before the main cylinders, to which the boosted air from the compressor cylinder expands into.

The comparison of the results of the CC model and the SICO model are shown in figure 7.14.

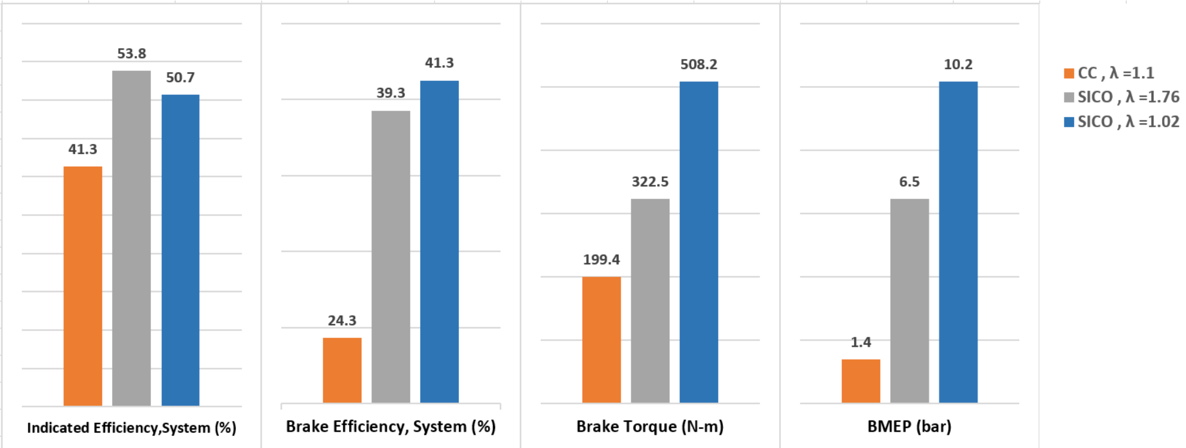


Figure 7.14: Performance comparison : Compressor Cylinder model vs SICO

The CC model has a total displacement volume of 9.3L which is almost 3 times larger than the SICO engine apart from the compressor. From the results, we can infer that firstly the indicated efficiency is much lower in the case of the CC model for the same amount of fuel energy as input (orange and grey bars). This is because in the CC model the IMEP contribution is negative as it functions is just as a compressor and no work is obtained from it and is represented in the indicated efficiency. In the case of the radial compressor in the SICO model, it is represented as auxiliary power and reflected as losses in brake efficiency. Additionally, it can be seen that the brake efficiency difference is much larger in comparison to the indicated efficiency owing to the large compression cylinder that contributes to the friction more than the power consumed by the radial compressor as in the SICO concept. Load capabilities are also lower in the CC model due to the AFR limit as mentioned before resulting in lesser scavenging than that in the SICO model.

7.10 Five stroke model

The intake port timing of the main cylinders were adjusted to favour the gas exchange for the 5 stroke (5S) model and so was the exhaust timing of the help cylinder. It was observed that the trapped air was greater in this concept due to the reasons explained in section 6.3.4. Thus the AFR limit is in favour of this concept and so the mass of fuel injected could be higher. Comparisons of the results were done based on same AFR for both concepts as well as for the highest possible for the 5S model which is shown in figure 7.15

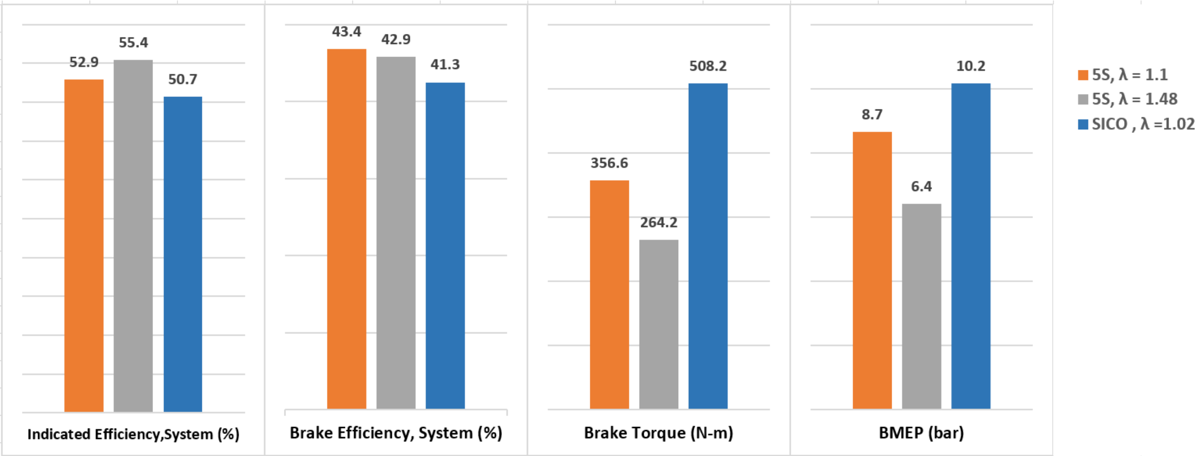


Figure 7.15: Performance comparison : Five-Stroke model vs SICO

The overall brake efficiency of the 5S model is greater than the SICO model due to the capability of the engine to achieve greater in-cylinder efficiencies from lesser residual gas fractions and due to the possibility to have a much more lean burning due to greater air availability from better trapping of fresh charge. The orange bar depicts the max load capabilities of the 5S model where it was simulated for a fuel input of 120mg corresponding to $\lambda = 1.1$ for the concept.

Since the SICO concept is based on a two-stroke operation, it can be seen that the max torque of the engine is 519.4Nm and hence can produce a greater power at that load point considering the swept volume size of the concept. Thus the greater BMEP of the engine as compared to the 5S model.

7.11 Final SICO model results

Based on all the evaluations explained above, the SICO engine concept was modified to achieve a high brake efficiency. The final engine model comprised of the following configuration shown in table 7.9

Table 7.9: Final SICO model parameters

Main cylinder bore	85mm
Main cylinder stroke	115mm
Main cylinder displacement volume	0.65L
Compression ratio main cylinder	16.5
Help cylinder bore	163mm
Help cylinder stroke	88mm
Help cylinder displacement volume	1.84L
Geometric Compression ratio help cylinder	60
Total displacement volume	3.14L
Intake port area at BDC	2550 mm^2
Intake port timing	35 CAD to BDC
Main cylinder exhaust valve diameter	34mm
EVO of Main cylinder	90 CAD after TDC (local CA)
EVC of Main cylinder	200 CAD after TDC (local CA)
EVO of Help cylinder	13 CAD before BDC (local CA)
EVC of Help cylinder	37 CAD before TDC (local CA)
VO of bypass valve	3 CAD after BDC (local CA)
VC of bypass valve	27.5 CAD after BDC (local CA)
Compressor gear ratio	62:1

Some critical results pertaining to the cylinder results of the concept at 1500 rpm max load are shown in table 7.10 and that of the whole engine is shown in table 7.11

Table 7.10: Final SICO model results-Cylinder

Main cylinder IMEP	22.55 bar
Main cylinder indicated efficiency	38%
Main cylinder heat transfer	17.88kW
Main cylinder trapping ratio	83.1%
Main cylinder residual ratio	7.35%
Main cylinder injection timing	14CAD before TDC
Help cylinder IMEP	5.34bar
Help cylinder heat transfer	6.23kW

Table 7.11: Final SICO model results-Engine

Brake torque	508.21 Nm
BSFC prime power	202.9g/kWh
BSFC 75% power	207g/kWh
BSFC 50% power	224g/kWh
IMEP	12.5bar
BMEP	10.2bar
FMEP	1 bar
Brake efficiency	41.3%
Indicated power	98.1kW
Total Heat transfer	42kW
Friction power	7.9kW
Compressor power	10.3kW
Brake Power	79.8KW

As the SICO concept's performance is attributed to the extraction of energy from the help cylinder, the pressure equalization in the main and help cylinder during the event of opening of the transfer port is one of the main characteristics of the concept. Figure 7.16 shows the pressure in the main and help cylinders for a single revolution of the engine.

The Inlet port of the main cylinder closes at 35 CAD after BDC (-145 CAD in figure) and the fresh charge starts to be compressed. The fuel is injected at -14 CAD before

TDC and combustion starts in the main cylinder. As the Help cylinder runs at twice the speed of the main cylinder, when the main cylinder is at TDC, the help cylinder is at its BDC and starts to compress the remaining residual air in the cylinder. Meanwhile at 90 CAD after TDC of the main cylinder, the transfer port is open and the help cylinder at this point is at its TDC. The burnt gases start to flow from the main cylinder to the help cylinder and continue to expand until both reach their BDC.

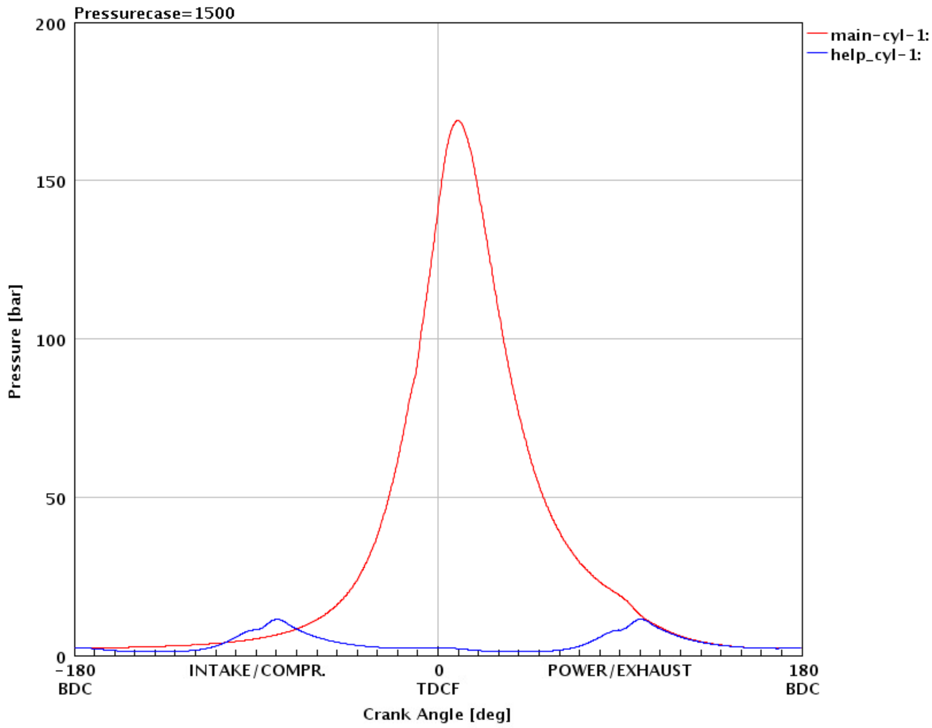


Figure 7.16: Pressure curves of one main cylinder and the help cylinder

A speed sweep was done using the configuration in table 7.9 to show the maximum load capabilities of the engine at different speeds. It must be noted that the results of the sweep are based on the compressor running at a speed based on the gear ratio set. The sweep results for an engine speed range of 900-2000 rpm were studied and the results are shown in figures 7.17 and 7.18.

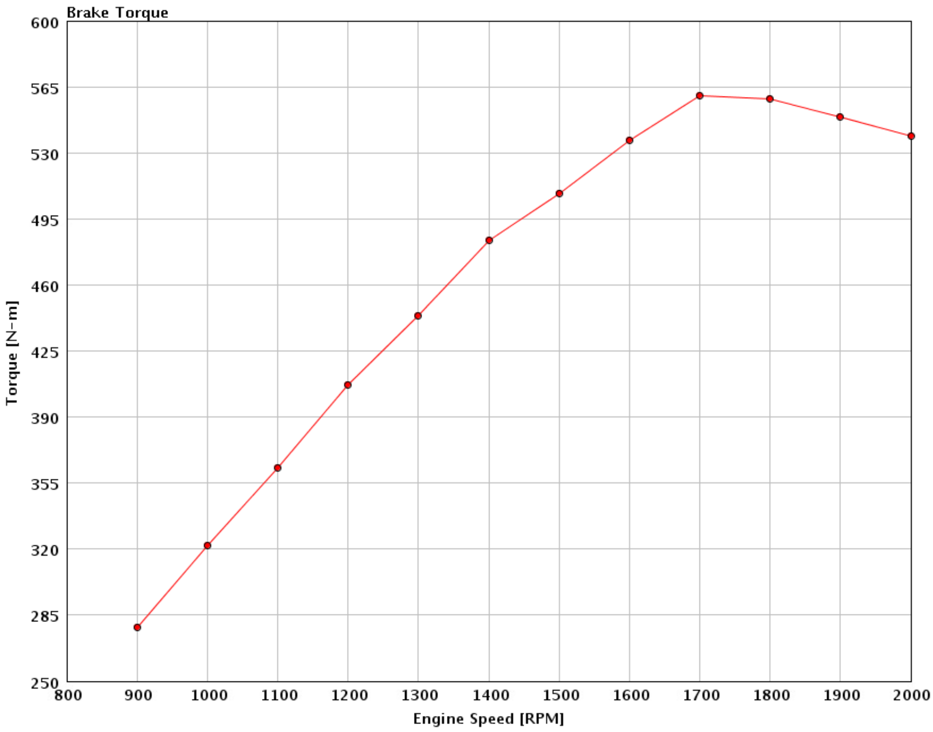


Figure 7.17: SICO concept - Brake torque vs Engine speed

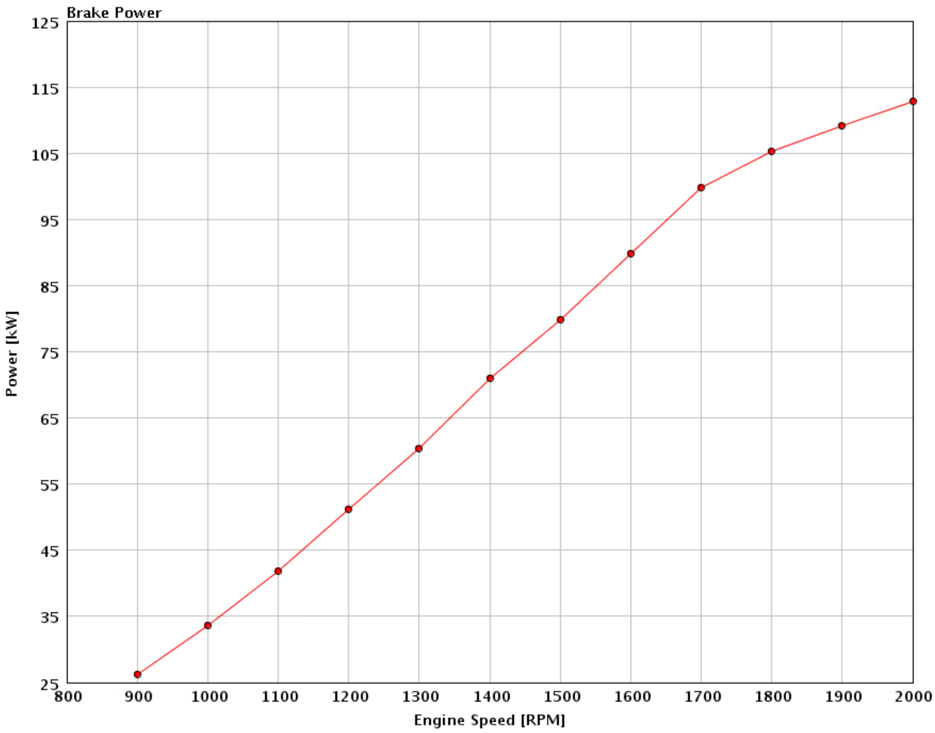


Figure 7.18: SICO concept - Brake power vs Engine speed

The SICO engine concept though was designed for a generator setup based on initial feasibility limitations owing to the compressor power consumption and hence the AFR limit being reached earlier. Since the engine concept has evolved to a much more efficient engine since the initial base model, the performance of the engine at higher speeds and to an extent at lower speeds are still quite significant. This can be seen from the figures 7.17 and 7.18. As the SICO concept aimed at higher efficiency, the performance of the concept in terms of brake specific fuel consumption(g/kWh) is shown in figure 7.19.

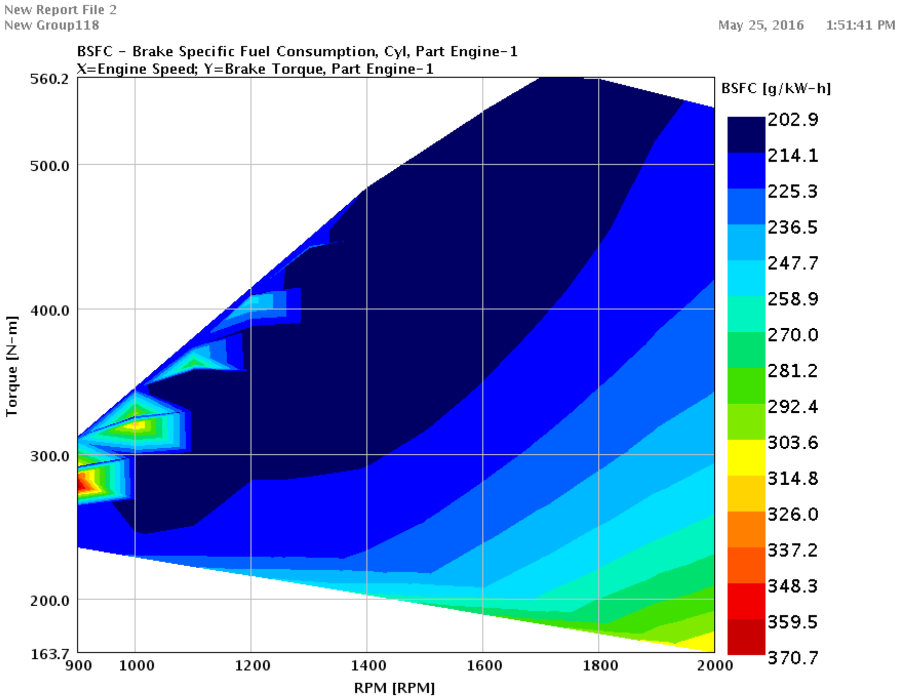


Figure 7.19: SICO concept - BSFC map

7.12 SICO concept vs Existing Engines

The evaluation of the SICO concept was based on its application as a power generator set. Existing generator sets were used for as a basis for comparison which are explained in section 5.4. The comparison of the simulation results of the SICO engine concept with the performance of the existing production generator engines are shown in figure 7.20.

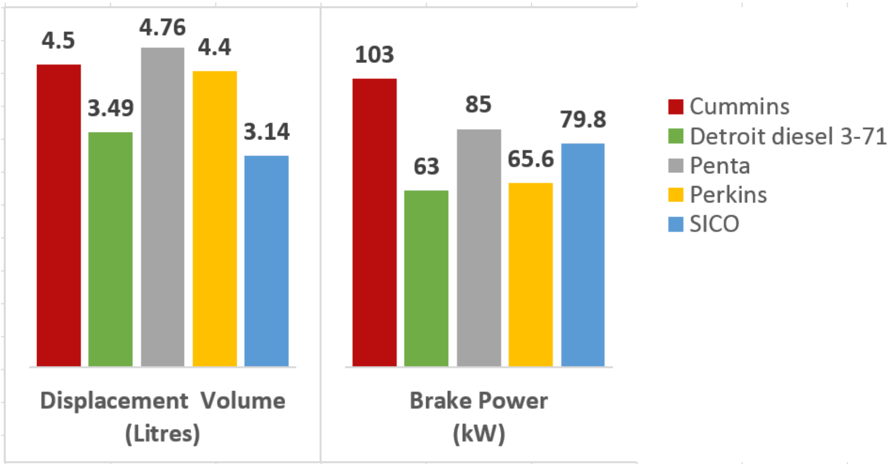


Figure 7.20: Performance comparison- Production engines vs SICO concept[23] [24] [25] [26]

The figure 7.21 shows the comparison of the existing engines and the SICO concept's performance at full load 75% load and 50% load.

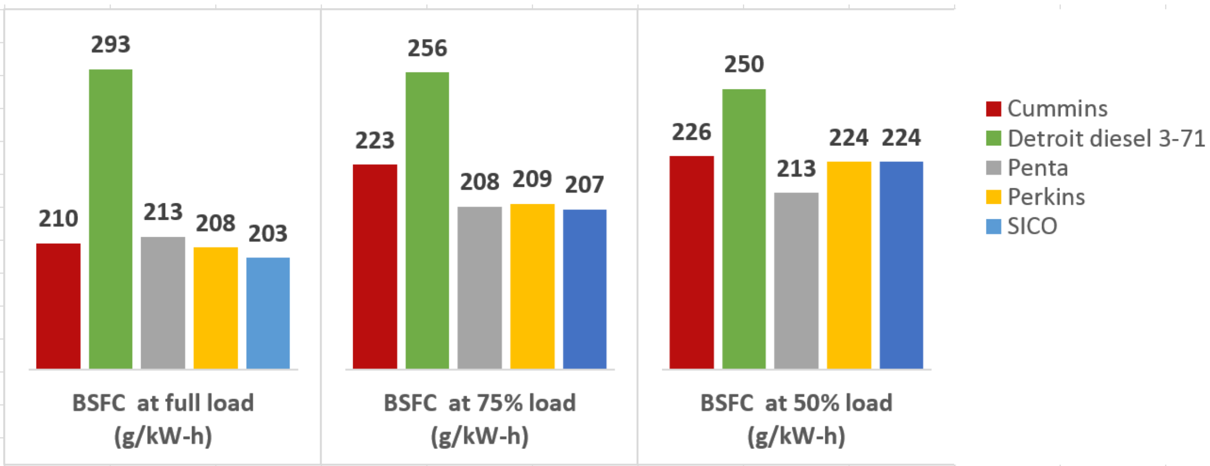


Figure 7.21: Efficiency comparison- Production engines vs SICO concept[23] [24] [25] [26]

From the figure 7.20 and 7.21 it can be inferred that the SICO concept is the smallest in terms of displacement volume in comparison to the other engines. Simulation results show that the peak power produced by the SICO engine is around 80kW making it a candidate for comparison among the existing engines. The fuel consumption value of the SICO concept shows the least value in comparison to the rest of the engines. The power to size advantage of the SICO concept is mainly owed to its two-stroke cycle operation among the other advantages of the concept. Although the results of the concept are based on simulations and not on actual physical data, the SICO engine is seen to be a possible concept to further investigate.

8

CONCLUSIONS

The SICO diesel engine conceptualized by Per Arne Sigurdsson was modelled and evaluated aiming for a high engine efficiency using GT-Suite software. Initial iteration termed as the Base model was modelled using the dimensions and working principle of the concept as per the inventor, with the exception of the radial compressor for charging rather than the compressor cylinder . Observing the shortcomings and possible avenues for improvement of the concept, the Advanced model of the concept was modelled and analyzed considering factors such as the scavenging, heat transfer and friction. The model was evaluated for the application as a power generator set and was developed for the same. Based on other concepts that implement a dual stage/post expansion strategy, concept models were built and compared with the SICO model to validate the SICO engine as a concept. A comparison of the simulation results of the SICO concept with existing engines in similar area of application was done to further affirm the concept's viability.

The certain key highlights of the concept evaluation are-

- The help cylinder being the main performance addition to the concept contributes a brake efficiency increase of 7%.
- Scavenging constraints of the concept due to the 1-D nature of the simulations, was evaluated to better suit the concept while keeping in check the realistic limitations. Around 7% of residual gases were found in the combustion cylinders which was observed to be a good estimation considering the high delivery rate of the fresh charge.
- As the concept's design is unconventional, the friction model was evaluated and augmented such that the friction values are in relation to the concept. Final iterations showed that the friction contributed to around 4-6% loss of engine efficiency.
- To understand the merits of the SICO engine as a concept, models implementing strategies based on literature were additionally built and compared with the SICO concept. The evaluation showed that though the SICO concept has a few limitations, it performed better in some categories. Comparing the SICO concept with the five-stroke model concept, it was observed that the five stroke model performed better at full load achieving a 2% higher brake efficiency than the SICO concept. The SICO concept however has higher torque capabilities in comparison to that achieved by the five-stroke model owing to the two-stroke cycle operation of the SICO concept.

- The effect of insulating the cylinders was evaluated and it was found that insulation of the help cylinder does not result in any significant increase in the performance of the engine. Further, the main cylinder insulation reduces the heat transfer and improves the efficiency of the engine by about 2% at high surface temperatures. However, the heat transfer model employed for the simulation needs to evaluated in detail as it might require changes in the values of the constants taken for the calculation in order to capture the complete effects of insulation at high surface temperatures [12].

- Cylinder dimensions plays a significant role in the performance of the engine. Smaller stroke and a larger bore for the help cylinder is found to give good results while the main cylinder was found to improve the engine performance when the stroke to bore ratio was around 1.2 - 1.4 when optimized for load conditions considered in this thesis.

Further work could be carried out in order to address some of the inherent limitations of 1-D modelling. A more thorough scavenging curve could be defined based on a CFD analysis of the engine setup. Also, the complete effects of the transfer port are difficult to capture through 1-D analysis only and more work with respect to the design and flow through 3D simulations can be helpful. The friction model can be evaluated in detail and the values can be experimentally validated in order to get a definitive understanding of the friction. The structural and design challenges involved in the concept forms a major area that can be evaluated. Considering practicality, the crankshaft setup needs to be analyzed to realize the different operating speeds of the help and the main cylinders.

The current 1-D simulations have been made with some appropriate assumptions and the results indicate that the concept may be viable. However, the SICO concept has to be evaluated more in detail in order to reach a decisive conclusion on its effectiveness.

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Appendix

1.Valve acceleration of the exhaust valve of the main cylinders

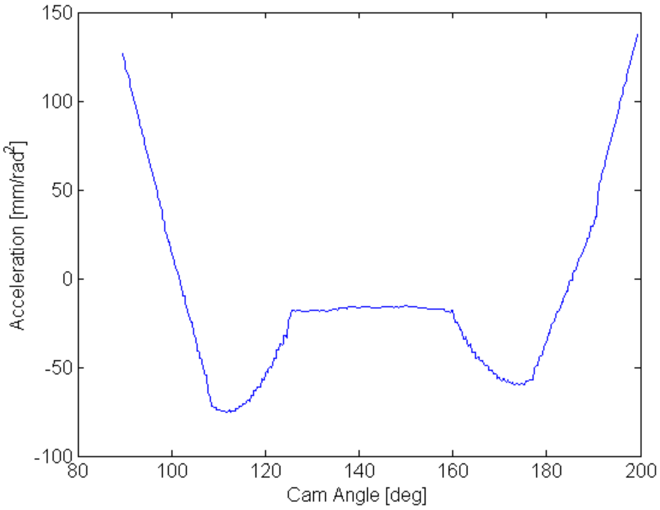

Figure A.1: Valve acceleration of the exhaust valves of the main cylinders

2. Model setup in GT-Suite with labels

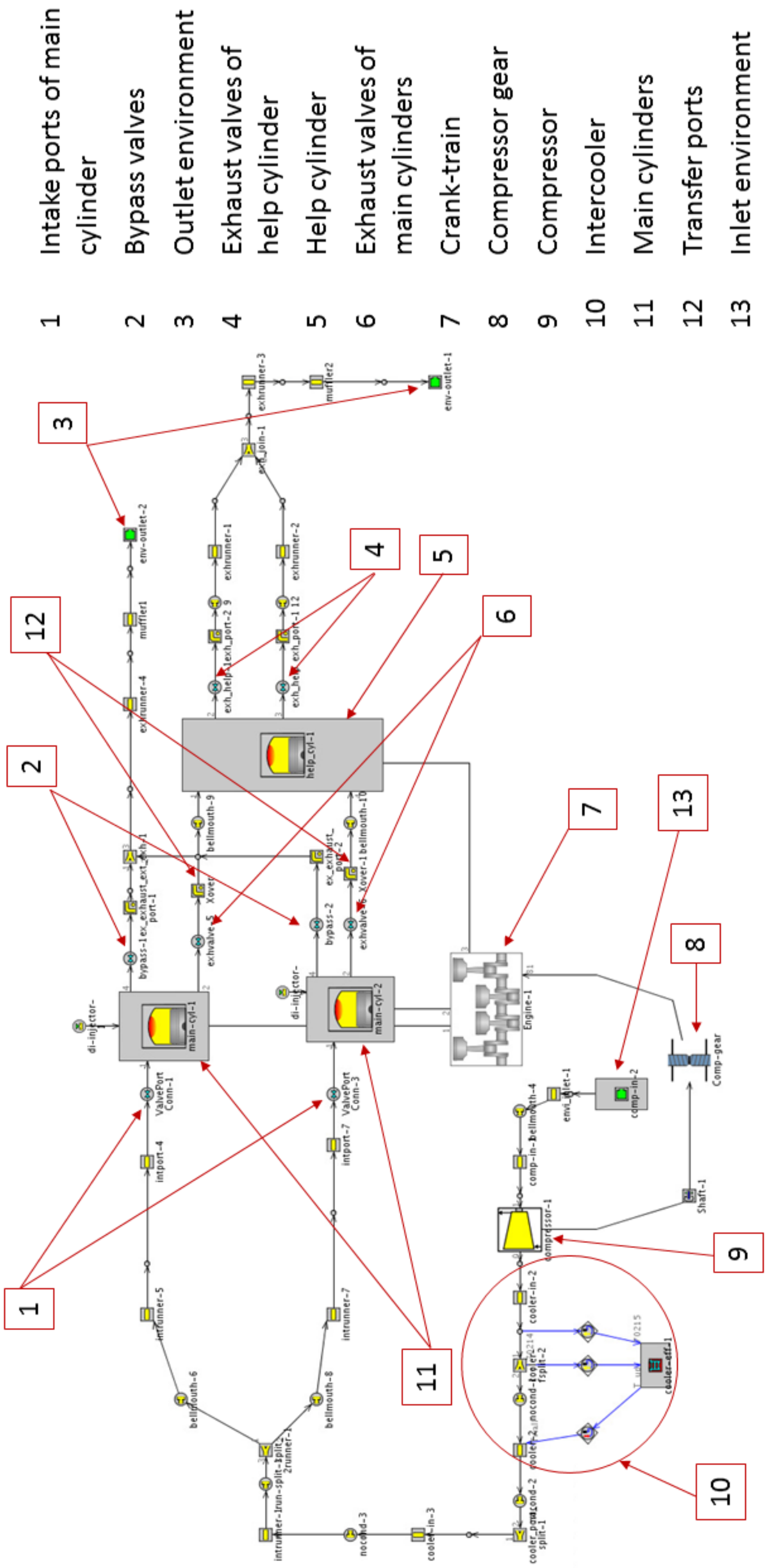

Figure A.2: Model setup in GT-Suite with labels

3. Speed map of the compressor

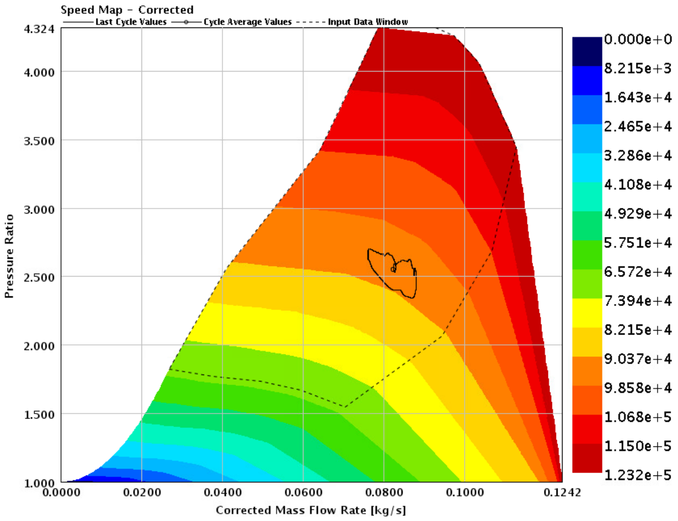

Figure A.3: Speed map of the compressor

4. Efficiency map of the compressor

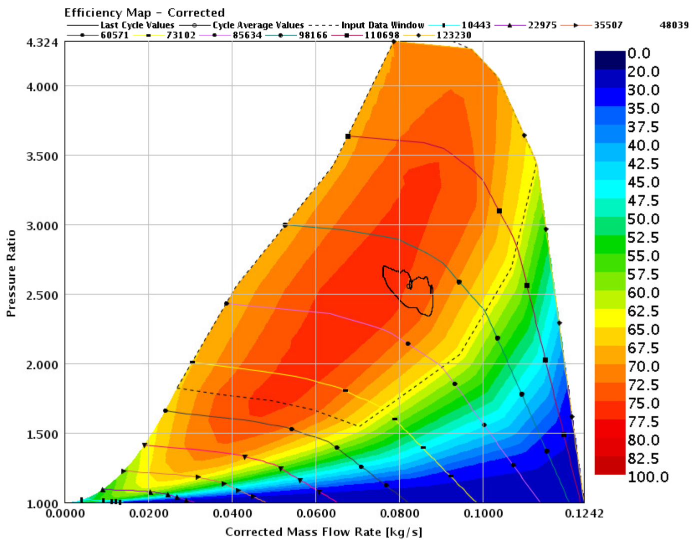

Figure A.4: Efficiency map of the compressor