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Divided Exhaust Period on Heavy-Duty Diesel Engines

Stefan Gundmalm

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Department of Machine Design
Royal Institute of Technology
SE-100 44 Stockholm

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Abstract

Due to growing concerns regarding global energy security and environmental sustainability it is becoming increasingly important to increase the energy efficiency of the transport sector. The internal combustion engine will probably continue to be the main propulsion system for road transportation for many years to come. Hence, much effort must be put in reducing the fuel consumption of the internal combustion engine to prolong a future decline in fossil fuel production and to reduce greenhouse gas emissions.

Turbocharging and variable valve actuation applied to any engine has shown great benefits to engine efficiency and performance. However, using a turbocharger on an engine gives some drawbacks. In an attempt to solve some of these issues and increase engine efficiency further this thesis deals with the investigation of a novel gas exchange concept called divided exhaust period (DEP). The core idea of the DEP concept is to utilize variable valve timing technology on the exhaust side in combination with turbocharging. The principle of the concept is to let the initial high energy blow-down pulse feed the turbocharger, but bypass the turbine during the latter part of the exhaust stroke when back pressure dominates the pumping work. The exhaust flow from the cylinder is divided between two exhaust manifolds of which one is connected to the turbine, and one bypasses the turbine. The flow split between the manifolds is controlled with a variable valve train system.

The DEP concept has been studied through simulations on three heavy-duty diesel engines; one without exhaust gas recirculation (EGR), one with short route EGR and one with long route EGR. Simulations show a potential improvement to pumping work, due to reduced backpressure, with increased overall engine efficiency as a result. Although, the efficiency improvement is highly dependent on exhaust valve size and configuration due to issues with choked flow in the exhaust valves. The EGR system of choice also proves to have a high impact on the working principle of the DEP application. Furthermore, the DEP concept allows better control of the boost pressure and allows the turbine to operate at higher efficiency across the whole load and speed range. The option of discarding both wastegate and variable geometry turbine is apparent, and there is little need for a twin-entry type turbine since pulse interference between cylinders is less of an issue.

List of Publications

Paper I

Divided Exhaust Period on Heavy-Duty Diesel Engines
S. Gundmalm, A. Cronhjort, H.E. Ångström
Proceedings of THIESEL 2012 Conference on Thermo- and Fluid Dynamic Processes in Direct Injection Engines

Presented at the THIESEL 2012 Conference.
11th-14th of September 2012, Valencia, Spain.

Paper II

Divided Exhaust Period: Effects of Changing the Relation between Intake, Blow-Down and Scavenging Valve Area
S. Gundmalm, A. Cronhjort, H.E. Ångström
SAE Technical Paper 2013

To be presented at the SAE 2013 World Congress & Exhibition.
16th-18th of April 2013, Detroit, Michigan USA.

Preface

This PhD project is funded by the Competence Center for Gas Exchange, CCGEx. Main partners of CCGEx are the Swedish vehicle OEM's, the Swedish Energy Agency and KTH. The aim of CCGEx is to perform demand driven research within the field of internal combustion engines, with focus on gas exchange and charging systems/processes as well as strategic engine components. CCGEx and its partners are greatly acknowledged for the financial and practical support.

I would like to especially thank my main supervisor Hans-Erik Ångström and my co-supervisors Andreas Cronhjort and Nils Tillmark for the guidance and the fruitful discussions. Many thanks also to Martin Örtengren and Daniel Norling for the practical help on behalf of Scania CV AB. To all my present and former colleagues at the department of internal combustion engines, thank you for creating a nice working atmosphere.

To my family, for your support and patience.

January 2013, Stockholm
Stefan Gundmalm

Abbreviations

BDC	bottom dead center
BDEVC	blow-down exhaust valve closing
BDEVO	blow-down exhaust valve opening
BMEP	break mean effective pressure
BSFC	break specific fuel consumption
CAD	crank angle degrees
CI	compression ignited
DEP	divided exhaust period
EGR	exhaust gas recirculation
ESC	European stationary cycle
FVVT	fully variable valve train
HC	hydrocarbons
HCCI	homogenous charge compression ignition
ICE	internal combustion engine
$IMEP_G$	gross indicated mean effective pressure
$IMEP_N$	net indicated mean effective pressure
IVC	intake valve closing
IVO	intake valve opening
LR	long route
PMEP	pumping mean effective pressure
RPM	revolutions per minute
SEVC	scavenging exhaust valve closing
SEVO	scavenging exhaust valve opening
SI	spark ignited
SR	short route
TD	turbo-discharging
TDC(F)	top dead center (firing)
VEMB	valve-event modulated boost
VGT	variable geometry turbine
VVT	variable valve timing

Contents

CHAPTER 1

Introduction

The price of conventional fuels will probably continue to be volatile during the coming years due to growing concerns on global oil scarcity [1], and a CO_2 limit for heavy-duty vehicles is under discussion. Thus it is becoming increasingly important to reduce the fuel consumption of the internal combustion engine (ICE) both from an environmental and energy security perspective, as well as for the sake of the end consumer's fuel economy. The reduction of fuel consumption should preferably be achieved while simultaneously keeping or even exceeding the present performance levels. This creates, together with increasingly stringent emission legislations, a very tough challenge for engine developers as these factors usually contradict each other. Engine development is carried out in many different research areas which when combined creates small but steady performance improvements as have been seen over the now around hundred year life span of the ICE. While big technology leaps have been performed in the past, increasing engine efficiency by several percent each year, the steps taken today are smaller and more costly. To further move closer to the theoretical or ideal efficiency of the ICE, improvements need to be made in all areas such as combustion efficiency, thermal efficiency, friction and pumping work. Two of the research areas that have become more and more prominent during the last years are turbocharging and variable valve train technology. The divided exhaust period (DEP) concept aims at combining the positive effects of using a turbocharger while removing some of the negative aspects, with the use of a variable valve timing (VVT) system on the exhaust side. In this thesis the feasibility of the DEP concept, when applied to heavy-duty diesel engines, is investigated. This chapter will present an introduction to turbocharging and variable valve timing as well as a brief motivation and main objectives of the thesis.

1.1 Turbocharging

Turbocharging of the ICE is a means of utilizing the otherwise wasted exhaust gas energy to increase the inlet air density and is one of the oldest waste heat

recovery measures found for the ICE [2]. This is done with a turbine driven compressor, where the exhaust gas expanded through the turbine will power the compressor, which in turn will compress the inlet air, increasing its density. Increased inlet air density brings the possibility to combust a proportionally larger amount of fuel and the power output of the engine is increased. For a schematic drawing of a turbocharging system see Figure 1.1.

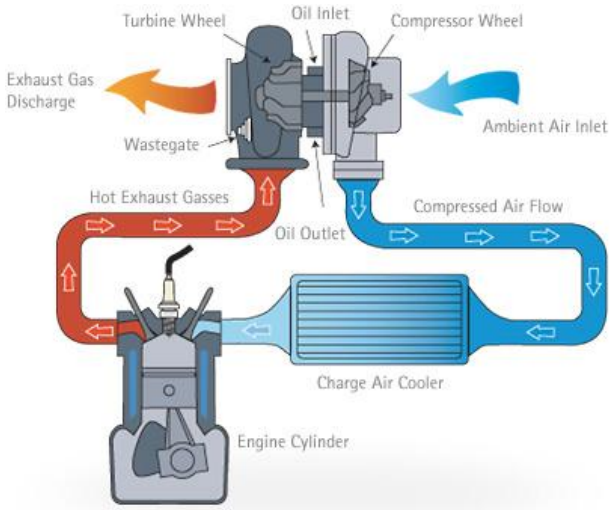

FIGURE 1.1. Schematic of a turbocharger system. *Image from BTN Turbo (www.btnturbo.com).*

The working principle of the turbocharger system is to use the turbine to expand the high pressure and high temperature exhaust gas that is expelled from the engine cylinder after combustion. Roughly 30-40 % of the energy released from combustion appears as energy in the exhaust gas [3]. The power extracted from the exhaust gas by the turbine wheel, is used to drive the compressor wheel which is connected via the turbocharger shaft. The compressor wheel will compress the incoming air above ambient pressure, increasing its density and oxygen content. The air temperature will also increase during this compression phase and it is common to cool it through a charge air cooler, further increasing the density, before inducing it into the cylinder. With a larger charge mass containing more oxygen, more fuel can be combusted which will increase the power output of the engine. Inversely, with constant power output the cylinder size can be reduced for increased engine efficiency which is referred to as down-sizing. This is especially true for spark-ignited (SI) engines which suffer from throttle losses at part load [4]. However, there are limitations to how much the inlet air mass can be increased. Firstly, the mechanical loading of the engine is limited by the peak cylinder pressure. By increasing the inlet air pressure and

amount of fuel burnt, the cylinder pressure during combustion will also increase and the structural design of the engine will set a limit for the peak cylinder pressure. Secondly, the thermal loading limit is set by combustion temperature which increases with increased levels of boosting. Too high in-cylinder and exhaust gas temperature can damage engine components, such as the turbine wheel. Furthermore, in pre-mixed combustion systems where the air and fuel is mixed before it is ignited there is a risk of pre-ignitions or onset of knock due to high in-cylinder gas temperature or component (injector, valves, spark plugs etc.) temperatures. The boost level limits can be adjusted by changing the compression ratio. For a certain compression ratio and turbocharger setup, exceeding the thermal and mechanical limits during engine operation can be avoided by using a wastegate or retarding the combustion process. The wastegate is basically a pressure valve that lets exhaust gas bypass the turbine, thus reducing the amount of work extracted by the turbine and reducing the charge pressure.

1.1.1 Utilization of exhaust gas energy

As mentioned above the exhaust gas energy expelled from the cylinder can contain 30-40 % of the total fuel energy released during combustion. The ability to extract a part of this energy in an efficient way in the turbine can have a large effect on total engine performance. The principle on which this otherwise wasted energy is utilized in a turbocharged engine is explained with the help of Figure 1.2.

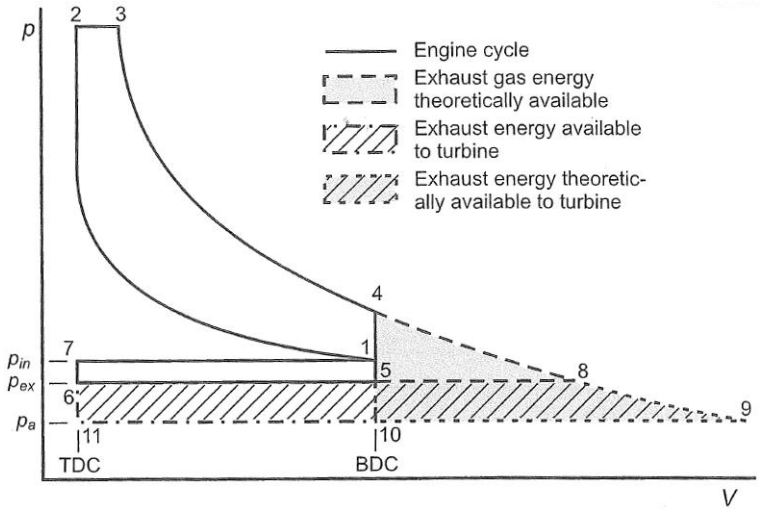

FIGURE 1.2. Ideal cylinder pressure vs. volume diagram for a turbocharged engine [3].

After the expansion phase (3-4) when the exhaust valves open at point 4 the cylinder content still has high pressure and temperature which in a naturally aspirated engine is wasted to the atmosphere. If the gas could be expanded in the cylinder further down to point 9 the energy represented by the area 4-9-10, commonly referred to as the blow-down energy, could theoretically be recovered. Instead of expanding it in the cylinder, the exhaust gas can be expanded in a turbine. The exhaust energy can be extracted with a turbine in mainly two ways, with a constant pressure or pulse pressure system.

A constant pressure system is achieved if the manifold volume is large enough to damp out any pressure pulsations from the exhaust valve events. When the exhaust valves open at point 4 the gas expands through the valve to point 5. During blow-down phase and the remaining exhaust stroke (5-6) the manifold and turbine inlet pressure remains constant and equal to p_{ex}. Hence, the kinetic energy and the high pressure of the blow-down pulse are not utilized for turbine work. The advantage with a constant pressure system is that the turbine will work under steady state conditions for which it can be optimized and a higher efficiency is therefore achieved.

As a way to recover a part of the blow-down energy, a pulse pressure system can be used instead. This is achieved if the manifold volume is kept to a minimum that will cause no damping of pressure pulsations. When the exhaust valves open at point 4, the pressure pulse caused by the cylinder pressure falling from 4-5 will be utilized by the turbine. The turbine inlet pressure will then fall from 4 and ideally down to 9 recovering the whole of the blow-down energy represented by the area 4-9-10. This is the ideal case where the turbine inlet pressure would need to reach cylinder pressure instantaneously at point 4 and then the manifold pressure would need to expand completely down to ambient. In reality this is not possible since there are flow losses in the valves, and with practical turbines there will always be a pressure drop over the turbine. Hence, for the real application only a part of the theoretically available blow-down energy (area 4-9-10) can be utilized due to these two reasons. The down-side to this system is that the turbine will only operate momentarily at its design point and the efficiency will suffer. The energy available for conversion is larger, but the efficiency of conversion will be lower. However, in most automotive application the pulse pressure system have shown larger overall benefits and is most commonly used.

1.1.2 A note on pumping work

To clarify, this section will briefly define what is being refereed to when discussing pumping work. Throughout the thesis the pumping mean effective pressure (PMEP) will be used as defined by Equation 1.1.

$$PMEP = \int_{180}^{540} \frac{pdV}{V_d} \tag{1.1}$$

Where p is cylinder pressure, V is cylinder volume and V_d is the displacement volume. PMEP is then calculated as an average of all cylinders. *Pumping work* is proportional to PMEP but with a different unit. PMEP multiplied by V_d gives pumping work in joules per cycle. Hence, PMEP is a measure of pumping work where the work has been normalized by the displacement volume so that engines of different size can be compared to each other. In short, PMEP is the work added to the crankshaft during the exhaust and intake stroke and is linked to the work it takes to exchange the exhaust gas with the fresh charge. This means that with positive PMEP, work is added to the crankshaft during the gas exchange phase. With negative PMEP, work is subtracted. Consequently, a higher positive PMEP is better for engine efficiency than a negative PMEP.

Pumping work exists in all engine types, naturally aspirated as well as turbocharged. For turbocharged applications, pumping work can be explained with the help of Figure 1.2. During the intake stroke (7-1) the higher than ambient pressure p_{in} will push the piston down, adding work to the crankshaft. During the exhaust stroke (5-6) the piston needs to push against the pressure p_{ex} which subtracts work from the crankshaft. The total pumping work or PMEP is then represented by the area 1-5-6-7. Since the intake pressure is higher than the exhaust pressure, PMEP will be positive and consequently add work to the crankshaft. Let's assume that p_{ex} is higher than p_{in}, then work is still added to the crankshaft during the intake stroke. However, the work subtracted during the exhaust stroke is larger, resulting in a total PMEP that is negative and consequently subtracting work from the crankshaft when looking at the whole gas exchange cycle. Total engine efficiency is then reduced.

1.1.3 Issues with turbocharging

Even though the overall effect of turbocharging gives a benefit in terms of both performance and efficiency some issues arise when applying a turbo machine to the internal combustion engine. The DEP concept aims at resolving some of

these issues described below. More on how they can be resolved will be discussed in Chapter 5.6.

(i)

Due to the reciprocating nature of the ICE, and the fact that the turbocharger is a rotary device, makes it difficult to match the two machines together. The ICE is basically a displacement pump where the flow out of the cylinders is highly pulsating and the total mass flow through it varies within a large span depending on engine speed. The characteristics of a turbine instead are that of a nozzle where the pressure ratio across the turbine is proportional to the density and the square of the flow velocity. This means that at low engine speeds and low mass flow rates, the expansion ratio is low leading to low power output. For high engine speeds and mass flow rates the turbine eventually becomes choked or gives too much power to the compressor. The problem of matching the turbocharger to the engine which operates at a large span of engine speeds and mass flow rates then becomes apparent.

(ii)

Since the turbo machine is a rotary device, its optimal and highest efficiency operating condition is during steady state flow with a certain mass flow and pressure ratio. This becomes a problem since the exhaust flow of the ICE is highly pulsating due to the blow-down and exhaust stroke events described previously. This in turn will result in a situation where the turbine only operates with its highest efficiency during a part of the whole duration of the engine cycle [3]. Hence, for the on engine application, it is only during small range of engine speed and only during a part of the whole exhaust period that the turbine will operate at its highest efficiency.

(iii)

Introducing a turbocharger on the engine will cause a backpressure that the piston has to work against as it expels the exhaust gas from the cylinder during the exhaust stroke. This leads to reduced pumping work. As explained above, some of this lost work is recovered in the turbine.

(iv)

The backpressure that the turbine creates also causes a larger amount of hot residuals to be trapped in the cylinder since it will be more difficult to evacuate all the burnt gases against this backpressure. High backpressure and more hot residuals will affect engine performance in a variety of ways. Higher backpressure will cause a backflow of the exhaust gas into the intake manifold. Together with a larger amount of hot residuals (with higher density) this will

decrease the volumetric efficiency, or the amount of fresh charge that can be induced. More residuals can lead to abnormal combustion phenomena such as knock in SI engines (since the in-cylinder temperature is increased) or it can affect *NOx* and soot emissions for diffusion type combustion [4].

(v)

The turbocharger will affect the transient response of the engine. Since the power output of the engine is directly related to the induced mass flow, a rapid increase in mass flow is required to achieve a fast transient response when increasing engine torque. However, the exhaust gas energy delivered to the turbine must first be used to increase the turbocharger speed before it can deliver sufficient power to the compressor to increase the air flow to the engine. The time it takes for the turbocharger to spin up is referred to as turbocharger lag, and it results in an engine torque increase that is delayed.

(vi)

Due to the difficulties matching the turbocharger to the ICE explained above, if a large turbocharger is chosen it will operate with high efficiency at high load and engine speed but the efficiency will be poor for low load and speed. The turbocharger lag will also be more severe with a larger turbocharger. If a small turbocharger is chosen, it will operate with high efficiency at low load and speed and the turbocharger lag will be reduced. However, at high load and speed the efficiency will be poor and due to choking of the turbine and the need to limit boost pressure at high load, a wastegate might be required. By introducing a wastegate a part of the energy available from the exhaust gas will not be utilized since a part of the mass flow bypasses the turbine wheel. With opened wastegate the pressure ratio and mass flow is reduced and consequently the total turbocharger efficiency is also reduced. This in turn leads to decreased PMEP.

1.2 Variable valve actuation

In this section a brief introduction to variable valve actuation will be given. The first part will present some common variable valve train systems and the second part will present ways in which variable valve timing strategies can be applied to internal combustion engines.

1.2.1 Variable valve train systems

The development from the conventional fixed camshaft driven valve train systems to the camless fully variable valve train (FVVT) has gone through many phases. The conventional fixed camshaft driven system works on the principle that the cam lobe situated on the camshaft (which rotates with half the engine

speed) will push the valve down against a valve spring. The valve spring will force the valve to a closed position on the way back. This kind of system is cheap to manufacture and operate but gives a fixed valve lift event. Since optimal valve timing is different at low and high speeds these fixed system will always represent a compromise.

The first step to move away from this compromise was to incorporate variable phasing systems of the camshaft. With this system it is possible to change the opening and closure time of the valve, though not independently of each other. This is done through a mechanism that allows the camshaft to be rotated in respect to the crankshaft. Note that with this system the duration of the valve event is still fixed. The first manufacturer to incorporate this system in production engines was Alfa Romeo based on a patent from 1980 [5].

Following that, was the implementation of cam profile switching system which can be done in many different ways but follows the same principle. By switching between, usually two, different cam lobes different valve lift heights and durations are possible. This kind of system was presented by Honda under the name VTEC [6]. Furthermore, by combining this type of cam switching system with a phasing system it is possible to vary both the opening time and duration of the valve event. This type of system was for example introduced on the Porsche 911 Turbo engine under the name VarioCam Plus [7].

The change to fully variable cam shaft based systems followed shortly after which made it possible to change the valve lift height continuously instead of just discrete steps. One of these systems is Valvetronic manufactured by BMW [8]. This system combines a phasing system with a mechanism that allows continuous adjustments to valve lift height. This is done through a conventional primary cam shaft with a variable phasing mechanism. However, a secondary eccentric shaft in connection with a series of levers and rollers acting on the valves makes it possible to regulate the impact that the primary cam lobe has on the valve itself. The levers are regulated continuously with an electric motor, resulting in a continuous variation of the valve lift and duration.

A lost-motion system is based on a conventional camshaft but incorporates a hydraulic link between camshaft and valve. The pressure in this hydraulic link is controlled by a solenoid. The oil in this link acts as a rigid body, making the valve follow the lift profile of the cam lobe. If the pressure in the hydraulic link is released by the solenoid the valve closes independently of the cam lobe position. This makes it possible to vary the duration of the valve lift or use multiple valve lift during one cycle. Currently the only mass produced valve

system of this kind is the UniAir [9] system developed by Schaeffler Group and Fiat Group. This system also goes under the name of MultiAir [10] when incorporated into Fiat Group products.

During the last few years the development of camless valve actuation systems has progressed rapidly. Some examples of these are an electromagnetic system from GM [11], an electro-hydraulic system from Lotus [12] and an electro-pneumatic system from Cargine Engineering [13]. The advantage of these systems is the total control of the valve opening event considering opening/closure time and lift height. The disadvantage is that they are still quite complex and expensive and there are still stability issues when running on the engine especially when opening against high pressures.

1.2.2 Application of variable valve timing

VVT systems are used for many different applications and engine types and there are of course numerous publications on this topic and just a few examples will be given in this section, with focus on compression ignited (CI) engines. For SI engines the main reason for using variable valve trains is to optimize the gas exchange process and cylinder charge filling for the whole engine speed range. Lately with the introduction of fully variable camshaft based valve systems it is also possible to remove the intake throttle and therefore removing the negative pumping work that comes with it. Throttle-less SI engines utilizing VVT are presented in [10,14]. In these engines the focus is on the intake side whereas in [15] for example more focus is put on the variation of the exhaust valve timing to optimize it together with a variable geometry turbine.

For CI engines the use of variable valve trains are less common in production engines. This is mainly because of the much smaller engine speed range of the CI engine, and the need to optimize the gas exchange process for a wide speed range is not as critical as for the SI engine. There is also no intake throttle in conventional CI applications. Even so, studies of VVT on diesel engines have been performed in several publications from which some examples will follow.

In [16], the potential of increase in output torque was shown by optimizing the intake and exhaust valve timing together with the turbocharger. In [17] the conclusion from using a VVT system on a single-cylinder CI engine was that there was no significant improvement to pumping work as compared to SI engines, but it showed to be useful for emission reduction by the use of internal exhaust gas recirculation (EGR). Improvement in low-end torque and fuel consumption by optimizing the cylinder charging was shown in [18] where the

authors also concludes that the VVT system offers fast and accurate control of internal EGR. Improvements in warm-up abilities and control of exhaust gas temperature was proved with better catalyst efficiency and particle filter regeneration as an effect. Improvement of fuel consumption of up to 6 % by improved cylinder charging with the use of VVT was presented in [19].

A common research area for FVVT systems is for homogenous charge compression ignition (HCCI) engines where the full variability is essential to control the phasing of the complex HCCI combustion process. For example, by means of varying the valve events, the effective compression ratio can be changed, thus controlling at which point the fuel and air mixture will auto ignite. Studies on HCCI combustion by the use of VVT have been presented in publications such as [20,21] to mention two examples.

A more non-conventional field where the use of some kind of VVT is required is the pneumatic- or air-hybrid engine. The main concept is that the engine's cylinders are connected to a pressurized tank which is charged by the ICE with the use of VVT strategies. The energy stored in this tank can then be used as complementary energy source for vehicle propulsion. Studies on pneumatic hybridization have been done in publications such as [22,23].

VVT also enables the possibility for Miller and Atkinson cycling which both lowers the effective compression ratio compared to the expansion ratio. This is done by either early intake valve closing (IVC) which is the Miller cycle or late IVC which is defined as the Atkinson cycle. Both cycles aim at decreasing the in-cylinder combustion process temperature and as a consequence the *NOx* formation is reduced. Substantial emission reduction, in particular *NOx*, by the use of late IVC was shown in both [24] and [25]. The conclusion is the same in [26] but the authors see no advantage of the Miller/Atkinson cycle compared to conventional EGR systems.

1.3 Motivation

As the road towards increasing engine efficiency continues, incremental improvements in all aspects of the complete engine system are needed as well as the introduction of new disruptive technologies. The introduction of turbocharging and variable valve actuation are two examples of how new groundbreaking technologies in the past have allowed big leaps in engine efficiency. As was shown in the previous section, turbocharging has some inherent drawbacks even though the overall effect it has on engine performance is positive. The aim of this thesis is to show the potential of the DEP concept to

improve some aspects of turbocharging, such as pumping work, by means of variable valve actuation. The DEP concept has previously shown positive results to engine efficiency on light-duty SI engines but investigations on heavy-duty CI engines are much scarcer. Therefore, this thesis will present an in-depth study to the advantages and disadvantages of introducing the DEP concept on heavy-duty diesel engines.

1.4 Objectives

The aim of this thesis is to present how the DEP concept affects engine performance and efficiency when introduced on a heavy-duty diesel engine. This will be performed through engine simulation on three different engines; without EGR, with short route (SR) and with long route (LR) configuration of the EGR systems. To be more specific, when applying the DEP concept, the objective is to investigate if it is possible to:

- Increase pumping work and thereby reduce fuel consumption
- Reduce residual gas content for increased volumetric efficiency
- Improve the use of the turbocharger's operating range
- Improve the boost control by using one exhaust valve as a wastegate
- Improve transient response

In addition to this, the effect different valve size configuration has to the DEP concept will be studied. This will be done by:

- Using standard exhaust valve area
- Increasing total exhaust valve area
- Increasing exhaust valve area, by decreasing intake area and thereby maintaining total valve area
- Changing the valve area relation between the two exhaust valves

CHAPTER 2

Divided Exhaust Period

This chapter will describe how some aspects of turbocharging are improved by means of variable valve timing when introducing the DEP concept. The basic idea behind the concept will be explained and a brief presentation of DEP from literature will be given. Lastly, the contribution from this thesis to the research field will be discussed.

2.1 Technical concept

Introducing a turbocharger on an engine creates an exhaust back pressure for the piston to work against during the exhaust stroke as shown in the previous chapter. As a way to improve the pumping work a second exhaust manifold is introduced, that leads the exhaust gases past the turbine. The standard two exhaust valves are each assigned to one of the two manifolds, as can be seen in Figure 2.1. The valves and manifold connected to the turbine are denoted as the blow-down system, while the valves and manifold that are bypassing the turbine are denoted as the scavenging system. The strategy of the DEP concept is to let the initial high energy blow-down pulse run the turbine, but lead the exhaust gases through the scavenging system during the rest of the exhaust stroke. Since the piston then can expel the exhaust gas out of the cylinder against ambient pressure instead of the turbine back pressure, an increase of pumping work is achieved. This in turn leads to a potential reduction in break specific fuel consumption (BSFC).

To be able to control the quantity of the exhaust flow distribution (which depends on the operating condition) to the different manifolds, the DEP concept requires a variable valve train system for the exhaust side, see Figure 2.2. Since the blow-down valve only needs a variable closure time but requires stability because it opens at high cylinder pressure, a lost-motion valve system similar to the UniAir system [9] may be considered. The scavenging valve on the other hand requires more degrees of freedom in terms of opening- and closure time, but it opens at low pressure. Therefore a FVVT system such as the one

developed by Cargine Engineering [27] may be considered. It should be noted that no study has been made as to which valve system is best suited, but these systems have been acting as templates in this thesis for a possible real implementation.

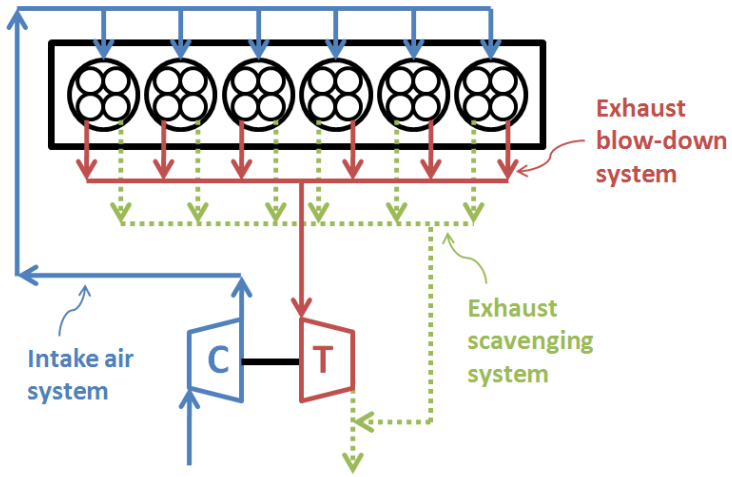

FIGURE 2.1. Sketch of manifold layout for the DEP concept.

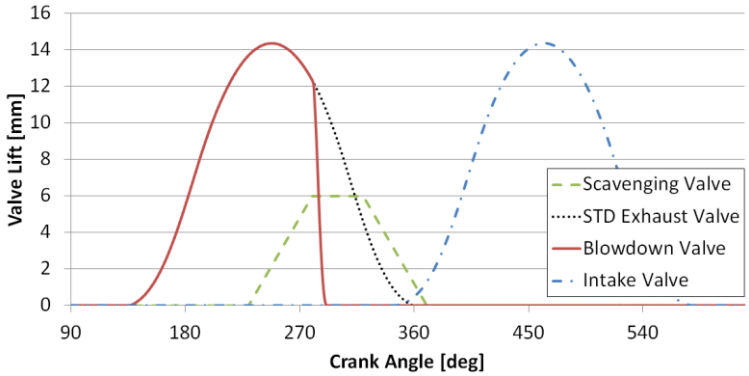

FIGURE 2.2. Conceptual valve lift curves for the DEP concept.

As can be seen in Figure 2.2, the strategy is to let the blow-down valve follow the lift profile of the original cam lobe until the blow-down phase is over, at which point it will be closed. The scavenging valve will then open to reduce the cylinder pressure to ambient level. This allows easier expulsion of the exhaust gases during the last phase of the exhaust stroke. A certain overlap between the blow-down and the scavenging valve is necessary to avoid choked flow. The closure time of the scavenging valve can be adjusted accordingly depending on what is being optimized, fuel consumption or residual gas content. The term *early*

scavenging will be used to refer to early closure of the blow-down valve and early opening of the scavenging valve. The term *late scavenging* will be used to refer to late closure of the blow-down valve and late opening of the scavenging valve.

The theory behind the concept can also be explained with a cylinder pressure vs. volume (p-V) diagram, see Figure 2.3. Note that this theoretical p-V diagram is far from the real process in an engine when flow losses and non-instantaneous combustion and valve lift events are considered. It serves only as a simplification to explain the principle. The area enclosed by the blue line is the gross indicated mean effective pressure ($IMEP_G$), meaning work done to the piston during the compression, combustion and expansion phase. At the end of the expansion stroke when the exhaust valves opens at (A), the cylinder gas temperature and pressure is still high which means a loss in potential work. By introducing a turbocharger the exhaust gases can be expanded further in a turbine and a part of this potential work (grey area) is recovered to drive the compressor as described in the previous chapter.

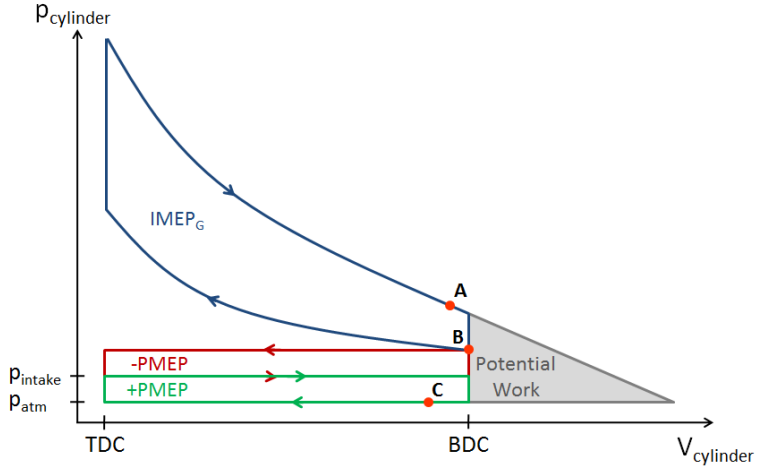

FIGURE 2.3. Theoretical cylinder pressure vs. volume with a positive and a negative pumping loop.

As mentioned previously the turbine introduces a drawback in terms of an exhaust back pressure that the piston has to work against. If this back pressure is higher than the intake charge pressure, a negative PMEP is achieved. As the piston travels upwards from bottom dead center (BDC) to top dead center (TDC) the cylinder pressure is higher than when it travels downwards, this means that it takes more work from the piston on the exhaust stroke than the piston gains on the intake stroke. This is seen as the red pump loop in Figure 2.3, starting at point B and going in an anti-clockwise direction.

With the DEP concept, the idea is to open the blow-down valve at point A to let the high energy blow-down pulse drive the turbine. When the pressure is equalized between the cylinder and exhaust manifold the blow-down valve is closed and the scavenging valve is opened instead (C). Since the pressure in the scavenging manifold is close to ambient level the piston will work with less resistance, compared to the usual turbine back pressure, as it travels from BDC to TDC. With an equal intake charge pressure, a positive PMEP is achieved. This is seen as the green pump loop in Figure 2.3, starting at point C and going in a clock-wise direction.

The net indicated mean effective pressure ($IMEP_N$) equals $IMEP_G$ plus PMEP, which makes it clear that the total indicated work done during the whole cycle will theoretically become larger with the DEP concept. This is also true for the cases where the standard turbocharged engine has a positive PMEP, since the positive PMEP for a DEP engine will theoretically be larger for the same intake pressure. A larger $IMEP_N$ for the same $IMEP_G$ (same amount of fuel injected) can also be translated into a reduction in fuel consumption if $IMEP_N$ is kept constant.

Aside from improved pumping work, the DEP concept offers other benefits as well and resolves some of the issues described in section 1.1.3. These benefits will be explained further in section 5.6.

2.2 DEP in literature

The base of this concept was first mentioned in a British patent from 1924 [28] and further patent claims have been made since then by several companies, e.g. Deutz AG [29], Fleming Thermodynamics Ltd. [30] and Saab Automobile AB [31]. In connection to the Saab patents the DEP concept was investigated by means of both simulation and engine tests [32]. The investigation was carried out on a passenger car SI engine with camshafts that gave fixed blow-down valve duration and three different scavenging valve durations. The exhaust valve area was increased by 30 %. Results from this study show improved full load torque and efficiency due to improved pumping work and reduced residual gas content (increases cylinder filling and knock resistance). Catalyst light-off time was reduced since the exhaust mass flow can bypass the turbine (which acts as a heat sink) through the scavenging manifold. Some negative aspects such as high turbine inlet temperature, choking of valves at high speeds and the need for a VVT system were pointed out. Due to the limitation of the fixed valve timing there was a need to implement a trapping valve located in the scavenging manifold. At low load for example the trapping valve needed to be closed to get

sufficient mass flow over the turbine. Furthermore, the trapping valve needed to be closed to avoid blow through of the fresh charge to the scavenging manifold. The trapping valve causes a backpressure in the scavenging manifold which contradicts the purpose of the DEP concept. Both of these situations could be avoided if it was possible to change the valve duration of both exhaust valves, and the trapping valve could be removed. Lastly, it was pointed out that a standard turbocharger was used which made it difficult to reach target boost levels at low engine speeds since a part of the exhaust flow is bypassed the turbine. A new turbocharger matching using a smaller turbine could have addressed this issue.

A numerical study of this concept was performed at TU Dresden with the help of VVT [33]. The difference in this study is that the two exhaust manifolds are both connected to one turbocharger each to offer the possibility of sequential or two-stage turbo charging. This setup allows the two turbines to be adapted for the different characteristics of the mass flow in the beginning and the end of the exhaust stroke. At low engine speeds valve timing is set so that a large part of the exhaust flow is directed through the blow-down valve to the primary turbocharger (sequential setup) or the high-pressure turbine (two-stage setup) and a large scavenging and intake valve overlap is used to reduce residuals. The reduced residuals and the use of a smaller turbine than baseline, gives improvement in both stationary and transient conditions at low engine speeds. When engine speed is increased the scavenging valve duration is increased to direct more flow to the secondary turbine (sequential setup) or the low-pressure turbine (two-stage setup) to limit further torque increase. This has the same function of a wastegate but the exhaust energy is still utilized in the second turbocharger stage. As engine speed increases further the scavenging valve opening time will be identical to the blow-down valve resulting in parallel operation of both turbines (sequential setup) and phasing out of the high-pressure stage (two-stage setup). The conclusion from this study was clear improvement in the scavenging behavior at low speeds, reduction of fuel consumption at rated power and improved torque build up. This has been patented for SI engines in a German patent [34]. It should be pointed out that this concept does not improve pumping work in the same manner explained previously since a turbine, that creates backpressure, is used for both manifolds.

BorgWarner have explored the DEP concept further with both simulations and engine tests for a turbocharged SI engine with different EGR routing [35,36]. Their concept, called valve-event modulated boost (VEMB) system, incorporates a concentric cam system powered by cam phasers which allow the opening and closure time of both blow-down and scavenging valve to be adjusted

individually. Note that the duration of both valve events are fixed with this valve train. The EGR flow is deducted from the scavenging manifold to preserve energy to the turbine, gain PMEP benefits and re-burn hydrocarbons. The EGR can be delivered to either the compressor inlet (at high load), via an EGR cooler for maximum knock relief, or un-cooled to the intake manifold (at low load) where the pressure difference is enhanced by the intake throttle. A thorough turbocharger matching was performed resulted in a single entry turbine smaller than baseline. As mentioned above, no wastegate is needed since the scavenging valve event has the same function. Exhaust valve area was increased by 6.5 % and port flow characteristics were improved by 13-15 %. Results from both simulations and engine tests showed improvements in both boost control and boost levels. Improved fuel efficiency by 1.5-5.5 % due to improved pumping work was shown. Furthermore, by reducing the backpressure at valve overlap the residual gas fraction was reduced, leading to a spark advance by up to 17° which in turn improved efficiency further for a total 12.3 % fuel consumption reduction. The reduced residuals would also make it possible to increase the compression ratio without risk of knock. The authors suggest that the compression ratio could be increased by 2-3 whole ratios for further efficiency improvements across the whole operating range. By taking the EGR flow from the scavenging manifold the 3-5 times higher HC content in this manifold could be rebreathed. For 15 % EGR rate, 40-70 % of the total hydrocarbon (HC) emissions were burnt in the cylinder instead of the catalyst. Exhaust valve choking at high engine speeds were apparent in this study as well.

At Loughborough University, DEP has been studied in a turbo-discharging (TD) concept by means of both simulations and engine tests [37,38]. TD can be applied to both naturally aspirated and turbocharged engines. For naturally aspirated engines the principle is to use a turbine on the blow-down manifold while the scavenging manifold is bypassed the turbine, similar to Figure 2.1. However, the difference is that the outlet of blow-down and scavenging manifold are both connected to the compressor driven by the turbine. The compressor outlet goes to the atmosphere. Hence, the compressor is here used to depressurize the scavenging manifold and post turbine side instead of increasing intake manifold pressure. What this depressurization does is that the scavenging manifold will have sub-atmospheric pressure (<0.5 bar) causing significantly improved pumping work during the scavenging phase. The depressurization of the post turbine side leads to an increase pressure ratio over the turbine for the same blow-down pulse pressure. The principle is the same when implementing TD on a turbocharged engine. In this case, both the scavenging manifold and primary turbine outlet is connected to a secondary turbine before the primary compressor. The secondary compressor is then used

to compress the intake air. This has the same effect of increasing pumping work and increase the pressure ratio over the turbines for the same blow-down pulse pressure. Result from simulations show that TD can increase the total pumping work, and even add work during the exhaust stroke, which in turn leads to fuel economy improvements of ~5 % or torque increase by 7 %. Similarly to the studies above, a reduction of hot residuals are proven due to the decreased exhaust pressure during valve overlap, which in turn reduces risk of knock and spark advance can improve efficiency further. Initial experimental tests seem to agree with the simulations. The TD concept was run with fixed exhaust valve timing where both exhaust valves had equal duration and a large overlap between them. The authors claim that the system performance was insensitive to exhaust valve timing. However, only blow-down valve opening and scavenging valve closure was varied in the sensitivity test. As will be seen in this thesis, the blow-down valve closing and scavenging valve opening has a large effect on the DEP concept. Hence, the TD concept would most likely benefit with a more extensive valve timing investigation. Further need for turbocharger matching was also pointed out.

The only study of DEP on heavy-duty diesel engines found to date is an engineering thesis from KTH performed in collaboration with Scania [39]. This study covers DEP both with and without EGR. The EGR cases are divided in one with short route EGR and one with scavenge sourced EGR. Note that the scavenging sourced EGR is directed to the post-compressor side, causing a backpressure for the piston, opposing the basic idea of DEP. Sweeps of different blow-down and scavenging valve duration were performed together with three different turbine sizes. One set of sweeps included the two exhaust valves to be open in series with a small overlap between them. Another set of sweeps were performed were the exhaust valves opened in parallel, with the blow-down valve always fully open, and the scavenging valve had varied duration. None of the cases studied here showed any improvement to engine performance. The reason is probably that in the first set of sweeps, the overlap between the valves was too small and the turbine size was not decreased enough to reach target boost level. The second set of sweeps, with both exhaust valves opened in parallel will cause the pressure ratio over the turbine to be decreased, again leading to problems reaching target boost level. Larger overlap for the series sweep and a smaller turbine could have resolved this issue.

2.3 Thesis contribution

From the previous section it is clear that the DEP concept displays a large potential to increase engine efficiency. However, most studies concerns DEP on

passenger car SI engines. This is probably due to their inherent problems with throttle losses, large amount of residuals, spark timing and compression ratio limitations, which makes a DEP implementation on SI engines more alluring. In spite of this the DEP concept in its basic form is in no way restricted to SI applications since the potential of pumping work improvements applies to all turbocharged engines. The study on heavy-duty diesel engine discussed in the previous section proved no benefit of using DEP, but due to the time-frame and scope of that study more in depth analysis is needed to understand the reason for this. Hence, this thesis will contribute with a more elaborate investigation of the DEP concept on heavy-duty diesel engines to complement the already positive potential of DEP shown on SI engines. The thesis will include a more extensive study of which the effects of DEP are when valve timing, valve size, turbocharger size and manifold geometry is allowed to be varied in a more unrestricted way.

CHAPTER 3

Engine Simulation Tool

The DEP concept has been studied through simulations with GT-Power, which is a 1D fluid dynamic simulation tool for engine applications. GT-Power solves the Navier-Stokes equations in one dimension for the flow in the piping system of the engine and uses simplified models, maps and lookup tables for other critical parts such as valves, turbocharger, cylinders, friction and combustion. These simplified models usually require extensive calibration which is performed based on measurement data from real engine tests. This chapter will give a brief description of how the fluid flow and some of these critical components are modeled. For more detailed information regarding GT-Power see [40].

3.1 Flow modeling

An engine model in GT-Power is built up by a large number of objects representing different components of the engine. A large part of these objects consists of the engine's piping system; intake pipe, inter-cooler, intake manifold and ports, exhaust manifolds and ports, exhaust pipe, EGR routing, EGR cooler etc. These different pipe objects represent the geometry of the real engine and describe straight pipes, pipe bends, flow splits and junctions. The whole piping system is discretized with a *staggered grid* approach, see Figure 3.1. This means that each pipe object is discretized into many sub volumes, and each sub volume is connected to its neighbor via a boundary. Scalar variables such as pressure, temperature, density and enthalpy are assumed to be uniform across each sub volume and are calculated at its centroid. Vector variables such as mass flux and velocity are calculated at each boundary of the sub volumes.

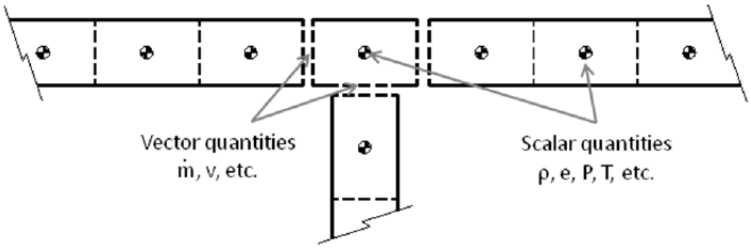

FIGURE 3.1. Schematic of staggered grid approach: scalars calculated at centroid, vector quantities at boundaries [40].

The fluid flow model then involves the solution of the Navier-Stokes equations, namely conservation of mass, momentum and energy. These equations are solved in one dimension, which means that all quantities are averaged across the flow direction. Mass conservation states that the rate of change of mass within a sub volume equals the sum of mass flux in and out of the sub volume:

$$\frac{dm}{dt} = \sum_b \dot{m} \tag{3.1}$$

Where m is the mass of the volume, subscript *b* denotes the boundaries and $\dot{m}$ is the boundary mass flux.

Momentum conservation states that the rate of change of momentum in the sub volume is equal to the net pressure forces and wall shear forces acting in a system plus the net flow of momentum in and out of the sub volume:

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_b(\dot{m}u) - 4C_f \frac{\rho u|u|}{2}\frac{dxA}{D} - C_p\left(\frac{1}{2}\rho u|u|\right)A}{dx} \tag{3.2}$$

Where p is the pressure, A is the flow area, u is the velocity at the boundary, D is the equivalent diameter, ρ is the density and dx is the discretization length. C_f is the skin friction coefficient and accounts to flow losses due to wall friction which is a function of both wall surface roughness and Reynolds number. C_p is the pressure loss coefficient and accounts for other flow losses than friction. This could for example be due to changing cross-sections, tapered pipes or pipe bends. The pressure loss coefficient can be set by the user or calculated by the code.

Energy conservation states that the rate of change of energy in a sub volume is equal to the sum of energy transfer in and out of the sub volume. Energy

transfer in the volume involves energy connected to work, mass flow and heat transfer:

$$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum_b (\dot{m}H) - hA_s\left(T_{fluid} - T_{wall}\right) \tag{3.3}$$

Where e is total internal energy, V is the volume, H is total enthalpy, h is the heat transfer coefficient, A_s is the heat transfer surface area and T_{fluid} and T_{wall} are the fluid and wall temperatures respectively. The heat transfer coefficient is calculated at every time step and is a function of the fluid velocity, the thermo-physical properties of the fluid and the wall surface roughness.

The conservation equations are then solved for each time step and sub volume. The preferred time integration method for most engine simulations is the explicit method since it more accurately captures pressure pulsations but requires small time steps. The implicit method allows larger time steps and therefore has faster execution time but does not capture pulsations as accurately, and is there for more suited for long duration simulations, e.g. exhaust system warm-up or cooling systems. In the explicit method the primary solution variables (mass flux, density and internal energy) are calculated based on the conservation equations by using values from the previous time step of the current sub volume and its neighbors. This gives the derivative of the primary variables and allows the value at the new time step to be calculated. At each time step the pressure and temperature are calculated by first solving the continuity and energy equations which will give the mass and energy in the volume. With mass known, density and energy can be calculated. The solver will then iterate on pressure and temperature (through the equation of state) until they satisfy the density and energy calculated for the time step. The time step is in turn restricted to satisfy the Courant condition which in turn depends on the discretization length. A small discretization length will result in more accurate predictions but a longer execution time.

When the flow in the engine's piping system is solved, simplified models, maps and look-up tables are used to represent the remaining components of the engine. The treatment of three key components will be discussed in the following sections.

3.2 Combustion modeling

When the in-cylinder mass has been predicted by the flow solver, a model is required to simulate the amount of chemical energy released as heat in the

cylinder during the combustion phase. One common approach is to express this in the heat release rate, which is the energy release as a function of time. The heat release rate can be calculated by the following equation, which states that the energy released by combustion (dQ) of the fuel equals the heat transfer to the cylinder walls (dQ_{HT}), energy lost into the crevices (dQ_{Crev}), the change in internal energy (dU) and the amount of work done by the system (dW):

$$dQ = dQ_{HT} + dQ_{Crev} + dU + dW \tag{3.4}$$

The heat release rate can then be expressed as a function of cylinder pressure and crank angle:

$$\frac{dQ}{d\theta} = Ah_c(T - T_w) + dQ_{Crev} + \frac{\gamma}{\gamma - 1}p\frac{dV}{d\theta} + \frac{\gamma}{\gamma - 1}V\frac{dp}{d\theta} \tag{3.5}$$

Where A is the combustion chamber surface area, h_c is the heat transfer coefficient, $T$and T_w are the gas and wall temperatures, γ is the specific heat ratio, p is the cylinder pressure, V is the cylinder volume and θ is the crankshaft angle. The heat transfer coefficient is estimated by the Woschni correlation [4]. The loss due to crevices can be treated in different ways, but since these are usually small the term can be omitted. Especially for diesel combustion since there will be no unburnt hydrocarbons stored in the crevices. With measured cylinder pressure, the heat release rate can then be calculated and inserted in GT-Power to model the combustion process.

A simplified way of describing the heat release rate is through the Wiebe function [4]. This equation describes the percentage of burnt fuel mass as a function of angle of start of combustion, combustion duration and two tunable parameters. By derivation, this will result in a bell shaped curved that with tuning of the parameters should resemble the real heat release rate.

3.3 Valve modeling

To correctly predict the induced mass into and out of the cylinder, flow losses across intake and exhaust valves need to be modeled. All valves (intake/exhaust valves, ball valves, throttles etc.) are in GT-Power handled in the same way but this section will focus on the intake and exhaust valves. The valve object itself does not contain any volume, instead it will describe the flow losses that will occur when the fluid travels from the volume upstream to the volume downstream of the valve object. This is done through a *discharge coefficient* which is defined as the ratio between the effective flow area and a reference flow area, usually the geometric area of the connecting pipe. It includes friction losses and

errors in assumptions of velocity profiles in the orifice equations. For simple flow restrictions the code can calculate the discharge coefficient but for more complex geometries like valves, the discharge coefficient needs to be entered by the user. For intake and exhaust valves the discharge coefficient is usually entered as a function of valve lift. The discharge coefficient is calculated with the following equations (isentropic flow through an orifice), based on measurement data performed on the cylinder head:

$$\dot{m} = A_{eff}\rho_{is}U_{is} = C_D A_R \rho_{is} U_{is} \tag{3.6}$$

$$\rho_{is} = \rho_0 (P_r)^{\frac{1}{\gamma}} \tag{3.7}$$

$$U_{is} = \sqrt{RT_0}\left\{\frac{2\gamma}{\gamma - 1}\left[1 - P_r^{\frac{\gamma-1}{\gamma}}\right]\right\}^{1/2} \tag{3.8}$$

Where A_{eff} is the effective flow area, ρ_{is} is the density at the throat, U_{is} is the isentropic velocity at the throat, C_D is the discharge coefficient, A_R is the reference flow area (on which the measurements are based on), ρ_0 is the upstream stagnation density, P_r is the absolute pressure ratio (static outlet pressure/total inlet pressure), R is the gas constant and T_0 is the upstream stagnation temperature. For choked flow Equation 3.7 and 3.8 is modified to be only a function of the upstream density, temperature, R and γ.

A common practice within industry is to measure C_D across intake and exhaust valve for different valve lift heights but for only one pressure ratio. This pressure ratio is usually very small (~0.95). Compare this to the relatively large pressure ratios (~0.53) that can occur during exhaust valve opening causing critical flow during the initial stage of the exhaust phase. This becomes especially important for the DEP concept since the two exhaust valves opens at completely different pressure ratios but use the same C_D values. It is suggested by Blair that to properly account for the real flow losses across valves in internal combustion engines, C_D needs to be a function of both valve lift and a large range of pressure ratios. Furthermore it should be measured for both intake and exhaust valve separately and in both directions [41].

3.4 Turbocharger modeling

In analogy with the valve objects, the turbine and compressor object in GT-Power is just a more sophisticated look-up table. The turbine and compressor object does not contain any volume itself for which the flow is solved, instead

they contain performance maps. These performance maps are used to calculate the upstream and downstream condition of the turbocharger which is then used by the flow solver for the rest of the piping system. The performance maps contain information about turbine and compressor efficiency as a function of mass flow, pressure ratio and turbocharger speed. The input data to these performance maps come from steady state measurements performed in a gas stand. These measured values are then inter- and extrapolated to cover the whole map. Since instantaneous turbine performance is required for the simulation the turbine is modeled with a quasi-steady approach. This means that it is assumed that the turbine performs under non-steady flow in the same manner as it would if those instantaneous conditions were steady [2]. An example of one of these performance maps with the measured values included can be seen in Figure 3.2.

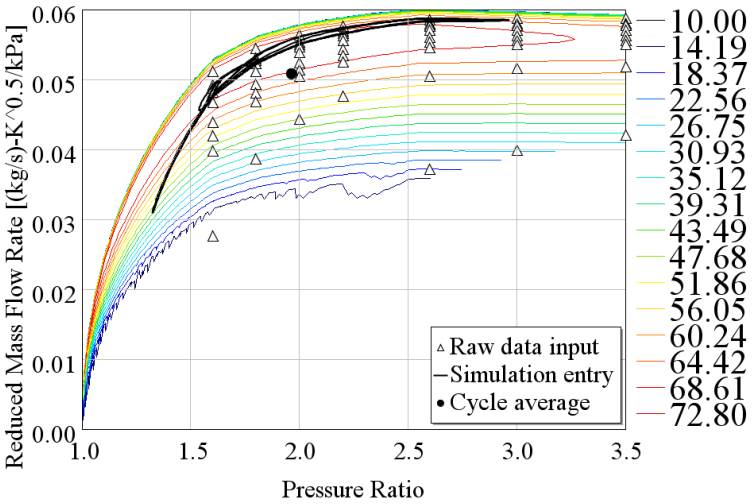

FIGURE 3.2. Example of turbine performance map generated by GT-Power. Contour lines are the turbine efficiency, white triangles the raw data input on which the map is built. The full cycle and cycle average values from the simulation is included as well. Speed lines are omitted for sake of visibility.

The turbine is then modeled in the following iterative manner (the process is similar for compressor and is not included here). The pressure in the adjacent sub volumes upstream and downstream the turbine is predicted by the flow solver and the turbine speed is known from the previous time step. The performance map is then used to look up the mass flow rate and efficiency that is then imposed in the solution. The isentropic enthalpy change across the turbine is then calculated based on the upstream conditions with the following equations:

$$T_{total,in} = T_{in} + \frac{u_{in}^2}{2c_p} \tag{3.9}$$

$$\Delta h_s = c_p T_{total,in}\left(1 - P_R^{\frac{1-\gamma}{\gamma}}\right) \tag{3.10}$$

Where $T_{total,in}$ is the total inlet temperature, T_{in} is the inlet temperature, u_{in} is the inlet velocity, c_p is the specific heat, Δh_s is the isentropic enthalpy change and P_R is the pressure ratio (total or static depending on data in performance map). The enthalpy change will then together with the turbine efficiency give the imposed outlet temperature or outlet enthalpy which in turn is used to calculate the power produced:

$$h_{out} = h_{in} - \Delta h_s \eta_s \tag{3.11}$$

$$P = \dot{m}(h_{in} - h_{out}) \tag{3.12}$$

Where h_{out} is the outlet enthalpy, h_{in} is the inlet enthalpy (known from the upstream state conditions), η_s is the turbine efficiency (from the map) and P is the turbine power. With the turbine power, the turbine torque can be calculated. Together with the consumed compressor torque and friction the change in turbocharger speed can be calculated:

$$\Delta\omega = \frac{\Delta t\left(\tau_{turbine} - \tau_{compressor} - \tau_{friction}\right)}{I} \tag{3.13}$$

Where $\Delta\omega$ is the change in shaft speed, Δt is the simulation time step, I is the shaft inertia and τ is the turbine, compressor and friction torque respectively. This means that any imbalance between the turbine and compressor torque will result in a new turbocharger speed. These new values of turbocharger speed and mass flow is then imposed on the solution for the next time step and the iterative process is repeated. This iteration process will continue until the mass flow and turbocharger speed reaches a set steady-state criteria.

Two issues arise from this kind of treatment of turbocharger modeling in GT-Power. Firstly, the measured points used to create the performance map are usually sparse and only covers a part of the whole operating range which makes it necessary to interpolate and extrapolate to cover the whole map. For on engine application, due to the highly pulsatile flow through the turbine, this will result in errors when predicting turbine efficiency since the operating conditions are often outside the measured values where the real efficiency is unknown. This can be

seen in Figure 3.2 where it is clear that a large part of the cycle is outside the measured values. For low load points that have lower pressure ratio and mass flow, this issue is even more pronounced. The implication this has to the engine simulation has for example been studied by Westin [42]. Secondly, the turbine object in GT-Power does not account for the internal volume of the turbine. Due to the highly unsteady flow in the turbine, the effects of mass accumulation in the volute are not considered. The implication of this has been studied in [43] where the exhaust pulse peak pressure is shown to be overestimated by ~12 % when the volute volume is not integrated into the engine model. This in turn led to an overestimation of the low-end torque by 1.8 bar of break mean effective pressure (BMEP). They suggest a solution where the proper volute volume is added as a pipe object directly upstream the turbine object. The discrepancy in both time resolved pressure pulsation amplitude and engine torque was eliminated with this solution.

CHAPTER 4

Methodology

This chapter describes the three engines and their baseline models for which the investigation has been carried out and the implementation of the DEP concept on these models. A motivation to the selected operating conditions will be given and lastly a short note on optimization procedure is discussed.

4.1 Baseline engine models

The DEP concept has been studied with GT-Power on three different heavy-duty diesel engines, one without EGR, one with a SR-EGR system and one with a LR-EGR system. The models for these three cases are in turn based on two Scania production engines, see Table 4.1.

TABLE 4.1. Engine specification for the two engines for which the models are based on.

Engine label	**Scania DC1201**	**Scania DC1306**
Emission class	Euro 3	Euro 5
Max. power [hp]	380	360
Max. torque [Nm]	1900	1900
Displacement [dm^3]	11.7	12.7
Bore [mm]	127	130
Stroke [mm]	154	160
Conn. rod length [mm]	255	255
Compression ratio	18	17.3
IVO [°ATDC]	346	344
IVC [°ATDC]	-142	-136
EVO [°ATDC]	136	118
EVC [°ATDC]	359	378
Intake dia. [mm]	41	44
Exhaust dia. [mm]	39	41
Turbocharger type	Twin-entry	VGT
EGR System	-	Short route

The case without EGR is based on the Scania DC1201 which is an 11.7 liter, six cylinder, in-line heavy-duty diesel engine. This engine incorporates a twin-entry

turbine and no EGR system. The baseline model for this engine was calibrated for full load in a previous work [44] and it was used as the base for the simulations. From now on this engine will be referred to as the non-EGR case.

The cases with SR-EGR and LR-EGR are based on the Scania DC1306 which is a 12.7 liter six cylinder, in-line heavy-duty diesel engine. This baseline engine has a variable geometry turbine (VGT) turbocharger and a SR-EGR system leading the EGR flow from the pre-turbine side to post-compressor/intercooler side, via a two-stage EGR cooling system. The VGT is used to regulate the backpressure to drive the EGR flow to the intake side, via an EGR-valve. The GT-Power model for this engine was then modified to incorporate a LR-EGR system, leading EGR flow from post-turbine side to pre-compressor side. An EGR throttle on the exhaust pipe is used to drive the correct EGR flow to the intake side. Here the VGT was replaced by a twin-entry turbine. The EGR rates are the same as in the SR-EGR engine and λ was kept identical as far as the fix geometry turbine allowed. From now on these two configurations will be referred to as the SR-EGR and LR-EGR case.

The baseline GT-Power model for the DC1306 engine was calibrated and provided by the engine manufacturer and was verified to measurements performed on the engine at the KTH engine test facility. The verification was performed based on measurements for the European stationary cycle (ESC) points. Key parameters were measured, such as:

- Cylinder pressure
- Intake pressure
- Exhaust backpressure
- Turbocharger speed
- Air and fuel mass flow
- Temperature before and after turbine
- Temperature before and after compressor

Measured pressures, mass flows and turbocharger speed deviated less than ± 5 % of simulation results, and temperatures less than ± 10 % of simulation results (in °C). The larger temperature discrepancies may be due to a modified intercooler and engine cooling system of the KTH engine test facility. Since the current study aims at comparing a baseline model to a modified version of the model, any discrepancy from measurements is assumed to equally affect both models. Therefore, no further calibration of the provided model was performed for the purpose of this study.

4.2 DEP model implementation

The three baseline engine models described above were then modified to incorporate the DEP concept. A second manifold connected to one valve per cylinder was introduced that leads the exhaust gas to the post-turbine side according to Figure 2.1. A single-entry, fixed geometry, turbine was used with the DEP concept for all three engine configurations. Since only the scavenging valve is open at the end of the exhaust stroke, there will be no expected blow-down pulse interference between the cylinders which is usually the case in an in-line six cylinder engine. Note that the baseline non-EGR and LR-EGR engines use a twin-entry turbine due to this pulse interference. In the SR-EGR case the VGT turbine can be omitted for the DEP configuration since it is possible to change the exhaust valve timing to control the backpressure instead. The backpressure is adjusted with proper valve timing strategies to get the necessary pressure difference to drive the EGR flow to the intake side.

The valve lift profiles for the scavenging valves were based on an idealization of the valve dynamics of the Cargine valve and data for opening/closure velocity were used accordingly [13,45]. The blow-down valve lift profile was modeled with a lost motion system in mind and valve closure velocity was set to the original valve system closing velocity.

Turbocharger matching was performed by scaling the mass flow axis (using *mass multiplier* in GT-Power) of the standard engines turbocharger maps. In the non-EGR and SR-EGR case, when a good match was achieved for the selected valve timings, new performance maps were implemented. These new maps correspond to the scaled maps, but represent a turbocharger in production. In this way a large number of turbochargers could be tested without handling a large number of different performance maps. For the LR-EGR case both the baseline and DEP engine use the same performance maps, and only mass multiplier has been used to differentiate the size of the turbocharger. This was done for a more fair comparison, so that any difference between baseline and DEP models is due to the concept itself and not to specific turbocharger maps. For both the SR-EGR and LR-EGR case, the DEP models were run with the same EGR rates as baseline and λ could vary within ± 4 %. By keeping identical EGR rates and λ, the same combustion model could be used and emission formation was assumed to be identical to baseline.

4.3 Operating conditions

The strategy for the simulations was to optimize the engine with the DEP concept for one load point considered important from a fuel economy

perspective. This means that the hardware like exhaust valve size, exhaust manifold geometry, turbocharger etc. were chosen based on optimization around this one load point. The optimization for the rest of the engine operating map was then based solely on valve timing strategies of the two exhaust valves.

Parallel to this study a thesis was performed investigating engine operating time statistics [46]. The aim of the thesis was to investigate at which torque and speed a truck engine operates most of its run time in the vehicle. At the time of the GT-Power simulations of the non-EGR case all the data for this thesis was not complete and therefore the load point was chosen somewhat arbitrary but still reasonable. The operating condition to optimize the hardware around was chosen in a medium load/speed point, which is denoted A7 in Figure 4.1. The rest of the load points (A1-A14) were chosen to cover the extremes of the engine's operating range.

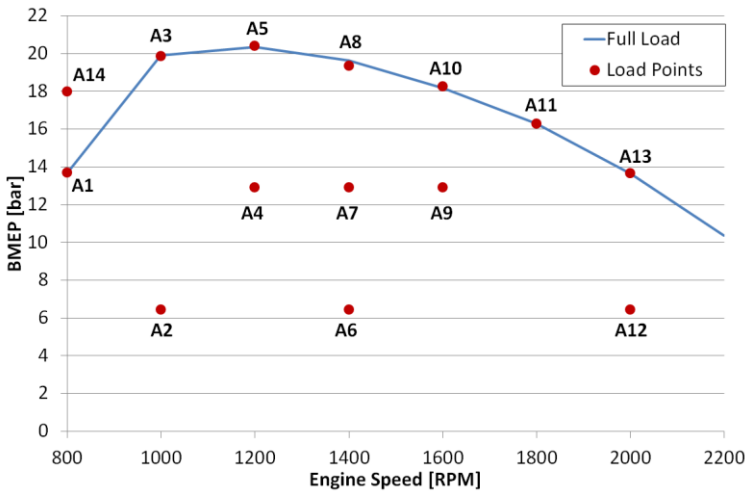

FIGURE 4.1. Map of chosen load points for the non-EGR case.

For the SR- and LR-EGR case, the results from [46] was used when the optimization load point was chosen. Depending on vehicle load and type of road (highway, city, hillside) the trends on which load point had most operating hours differed. A trend that can be seen though is that the automatic gearbox controls the engine speed to stay within a quite narrow range around 1400 RPM for most cases. For engine torque it is harder to find a single point that is more prominent than others. For city and hillside driving operating hours are more spread out in terms of torque, but always close to 1400 RPM. For highway driving the most prominent operating point lays around 100 kW and 1400 RPM. Since the comparisons between the baseline and the DEP engines are carried out on the ESC cycle points, it was decided to optimize the hardware for an ESC load point

close to this condition. The chosen load point is denoted B9 in Figure 4.2. Aside from the 12 ESC cycle points (idle is omitted) four 100 kW points were added to the study to represent highway driving at different gears.

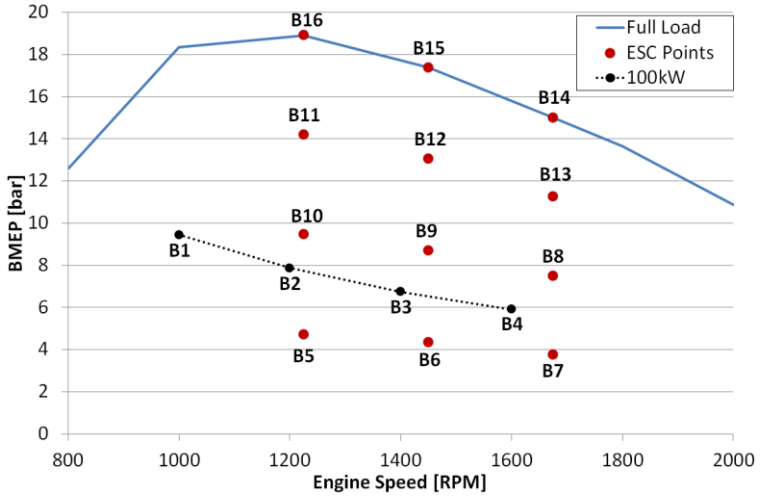

FIGURE 4.2. Map of chosen load points for the SR-EGR and LR-EGR cases.

4.4 Optimization procedure

The optimization procedure of all three cases where performed in an iterative procedure. It was not possible to optimize one parameter at a time since it was apparent already at an early stage that there were strong interaction effects between turbine size, valve size, valve timing, manifold geometry etc. It was decided to not test all parameters against each other, to limit the test matrix. Instead the optimization went through an iterative process, for which an example will be given here. Beginning with some initial valve and manifold size the exhaust valve timing was swept together with turbocharger size. The exhaust valve timing sweep consists of changing blow-down exhaust valve closing (BDEVC) and scavenging exhaust valve opening (SEVO). Some limitations were put to how early BDEVC timing was and how large the overlap between BDEVC and SEVO could be. Blow-down exhaust valve opening (BDEVO) and scavenging exhaust valve closing (SEVC) timing was identical to the baseline case in the first iteration. When an optimal match was found between turbocharger size and valve timing, this setting was run with different manifold geometry and valve size. The manifold geometry was changed in terms of pipe diameter only, since the length is restricted by the distance between the cylinders. In effect it is the manifold volume that is changed. For the scavenging manifold the length of the pipe after the junction (which connects all cylinders) was changed since in some cases there was a pulsatile flow also in this manifold. The

length was changed to get low pressure at the scavenging port during the scavenging phase. At this stage SEVC was optimized as well. The valve timing and turbocharger sweeps were then re-run for this new geometry setting. When the same settings were achieved in the new iteration step the optimization was considered finalized. This procedure was only performed for the optimized load points A7 and B9. For the remaining load points it was only valve timing that was optimized (BDEVC, SEVO and SEVC). Tests with changing BDEVO were performed as well but without improving the total efficiency. There are obviously many more parameters that could need optimization when switching to the DEP concept, e.g. injection strategies, boost requirements or the combustion system, but further optimization of the complete engine system was deemed to be outside the scope of this thesis.

CHAPTER 5

Results

In this chapter a brief summary of the result from the appended papers will be given. The three studied cases will first be discussed individually, followed by some complementary results not presented in the papers. The chapter is finalized with a discussion around the similarities between the three cases. For more detailed results the reader is referred to the appended papers.

5.1 Studied cases

The DEP concept has been studied on three different cases and a brief motivation to why these cases were chosen will be presented in this section. As mentioned in Chapter 2, the reason to focus on heavy-duty diesel engines was due to the lack of publications of the DEP concept applied to this segment of engines. The investigation was initialized on an engine without any EGR system. The reason for this was to keep the complexity of the engine system at a minimum so that the effects of the DEP concept could be studied in isolation. With this as a base, the investigation was moved to an engine with a SR-EGR system to understand the interaction between the EGR system and the DEP concept. This added to the complexity since the DEP concept in this case strongly interacts with the control of EGR rate. In an attempt to separate the DEP concept from the EGR system, allowing more freedom for valve timing strategies and turbocharger matching, a third case study was initialized. In this third study, the DEP concept was introduced on an engine with a LR-EGR system, which is a well-established and simple system to introduce cooled EGR [47]. From now on these three cases will be referred to as the non-EGR case, the SR-EGR case and the LR-EGR case. More information on these engines can be found in section 4.1.

5.2 The non-EGR case

This section presents the main results from the non-EGR case study. Results presented here concerns the pumping loop for the optimized load point,

turbocharger performance and fuel consumption for all investigated operating conditions. For more detailed information regarding this case, the reader is referred to **Paper 1**.

5.2.1 Gas exchange process

As mentioned in Chapter 4, the hardware was chosen for load point A7 in Figure 4.1, and the optimal valve time strategy was sought while matching the turbocharger to give the lowest BSFC. The intake pressure was set to be the same as the baseline engine to not affect the combustion process or emission formation. With similar in-cylinder conditions the emission formation is assumed to be unaffected. Note that the exhaust valve size is 30 % larger than baseline. The resulting pumping loop with optimal valve timing and turbocharger size is presented in Figure 5.1.

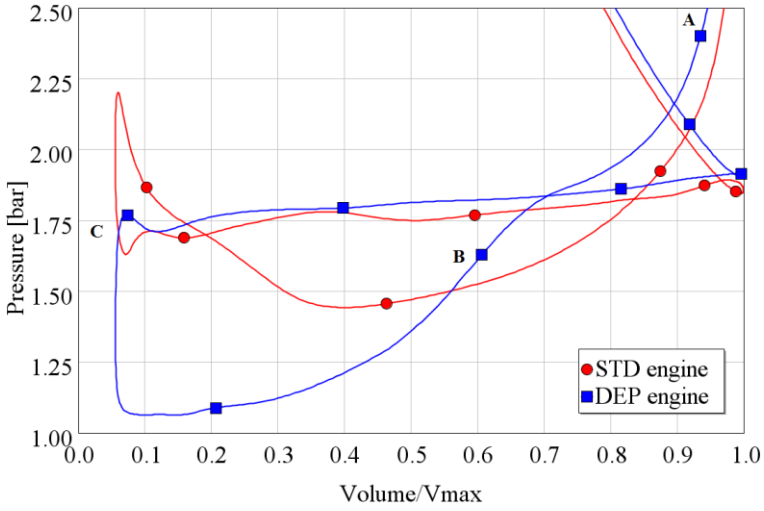

FIGURE 5.1. Cylinder pressure vs. volume for the non-EGR baseline and DEP engine during the gas exchange loop for load point A7.

Comparing the baseline and DEP pumping loop, the process is similar as it moves from exhaust valve opening at point (A) to point (B). In this blow-down phase the mass flow is mainly driven by the pressure difference between cylinder and exhaust port. In both cases this blow-down pulse feeds the turbine. The reason for the higher cylinder pressure for the DEP case is due a smaller turbine size and the fact that only one exhaust valve is open, restricting the outgoing flow. However, when moving from point (B) to point (C) the cylinder pressure curves starts to deviate. The mass flow in this scavenging phase is mainly driven by motion of the piston. In the baseline case, the turbine will create a backpressure for the piston to work against as it moves towards TDC. Close to

TDC the blow-down pulse from the next cylinder in firing order will interrupt the scavenging process causing the backpressure to increase. This is seen as a small pressure peak at the end of the exhaust stroke. For the DEP case instead, the blow-down valve will close and the scavenging valve open at point (B). Since the scavenging valve is connected to a manifold with ambient pressure the cylinder pressure drops quickly. The piston can then expel the remaining exhaust gas against ambient pressure instead as it moves towards TDC. At point (C) both exhaust valves are closed and the intake valves open, increasing cylinder pressure to charge pressure level. The intake pressure difference between baseline and DEP is in this case 0.04 bar and is due to the optimization procedure. As PMEP equals the enclosed surface areas in Figure 5.1, one can see that PMEP for the DEP case is larger than the baseline case. PMEP is in this case increased from 0.04 bar to 0.22 bar resulting in a BSFC reduction of 2.1 %.

5.2.2 Turbocharger performance

The reason why it is possible to keep the charge pressure at the same level even though the mass flow to the turbine is reduced can be explained by comparing the turbine performance maps. Figure 5.2 shows the performance map for the baseline engine with the full cycle and cycle mean values of reduced mass flow and pressure ratio. The turbine efficiency is represented by the contour lines. Due to the highly pulsatile exhaust flow, the mass flow and pressure ratio will vary within a wide range in the performance map, consequently affecting the turbine efficiency. The same performance map for the DEP engine is presented in Figure 5.3. The mass flow capacity or size of the DEP turbine is 38 % smaller than baseline. Here one can see that the mass flow and pressure ratio varies with a narrower range within the performance map. When only the high pressure blow-down pulse is directed to the turbine, the flow will be less pulsatile but with the same peak pressure. With a proper valve timing strategy it is possible to control the mass flow and pressure ratio of the turbine to make it operate in a high efficiency area to larger extent than the baseline engine. Furthermore, the scavenging phase that is present in the baseline case but not in the DEP case, does not contribute much to the total turbine work since it has low mass flow, low pressure ratio and low efficiency. To conclude, even though the mass flow is reduced in the DEP case it is possible to produce the same turbine work since the average pressure and efficiency is higher due to the less pulsatile flow.

While this is true for load point A7, the same trend is valid for the rest of the load points. In the baseline case the turbine will work with reduced efficiency at off-design load points. However, in the DEP case it is possible to control the turbine mass flow and pressure ratio with valve timing strategies, making the

turbine work in its high efficiency area to larger extent. More details on this can be found in **Paper 1**.

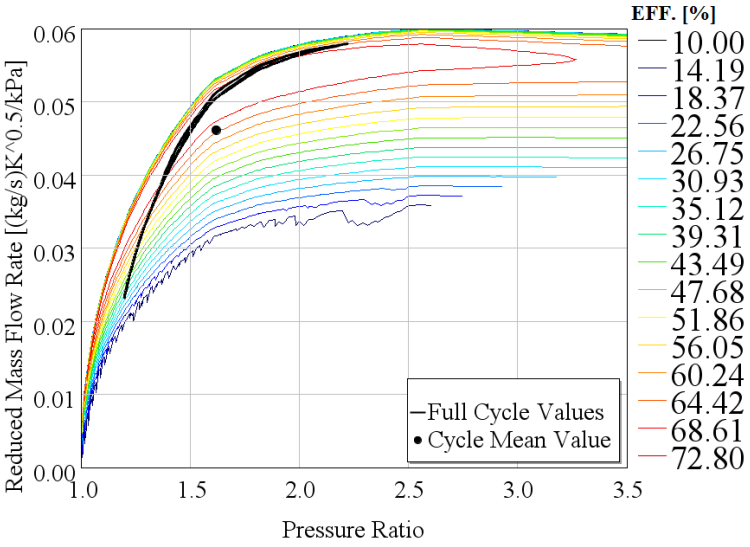

FIGURE 5.2. Turbine performance map for the baseline engine operating at load point A7.

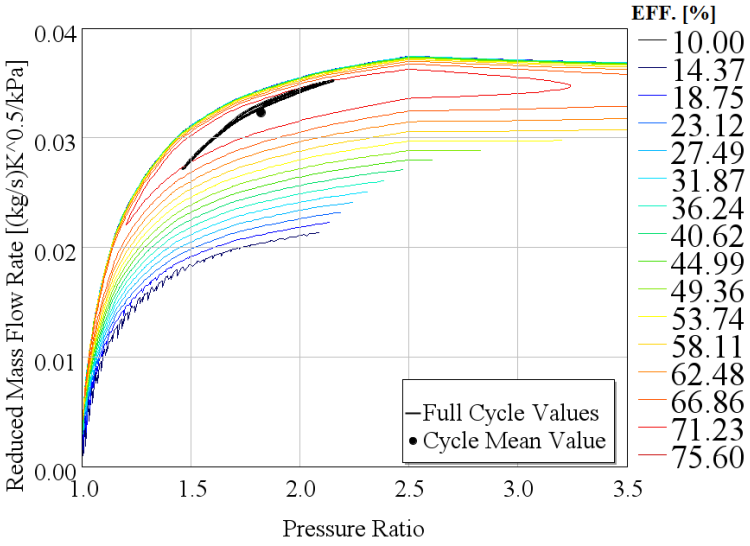

FIGURE 5.3. Turbine performance map for the DEP engine operating at load point A7.

5.2.3 Full operating range

Based on the hardware chosen for load point A7 the remaining load points were optimized solely by changing valve timing. Fuel consumption was minimized while keeping charge pressure at the level close to the baseline engine. The

change in BSFC for all load points investigated can be seen in Figure 5.4. Note that this figure is presented in **Paper 1**, but is showing the BSFC reduction when comparing to the baseline engine with standard exhaust valve size. Figure 5.4 presents the BSFC reduction when both engines use 30 % larger exhaust valve area, for a more fair comparison. As can be seen, the difference between Figure 5.4 and Figure 6a in **Paper 1** is not that large which means that the DEP concept benefits much more from larger valves than the baseline engines does.

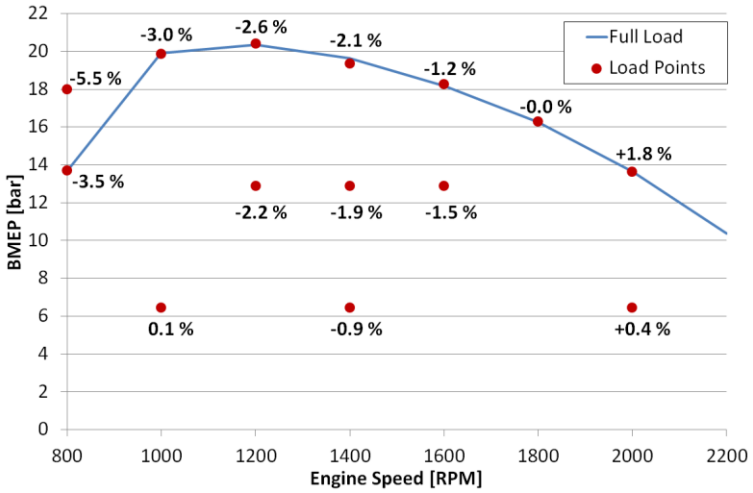

FIGURE 5.4. Change in BSFC for the DEP engine compared to the baseline engine, both with 30 % larger exhaust valve area.

The general trend is that at low load points, the backpressure is low and there is less to gain from the DEP concept. At high loads where the backpressure is higher, the potential to reduce this backpressure is higher. Therefore the BSFC reduction is higher at high loads. When changing engine speed the trend is that at low engine speed the BSFC reduction is higher than at high speed. The reason for this is due to flow restriction in the valves. Since only one exhaust valve is open at a time in the DEP case, it becomes more prone to choking at high engine speeds and higher mass flow rates. At low engine speed the valves are less choked and it is easier to expel the exhaust gas during the scavenging process, hence the BSFC gain is higher. Load point A14, which is above the full load curve, shows that with a smaller turbine the boost pressure is increased and it is possible to increase the full load torque for the same λ value.

5.3 The SR-EGR case

This section presents the main results from the SR-EGR case study. Results presented here concerns the pumping loop for the optimized load point, a study

of different valve area and ratios, and fuel consumption for all investigated operating conditions. For more detailed information regarding this case, the reader is referred to both **Paper 1** and **Paper 2**.

5.3.1 Gas exchange process

The procedure for this case was the same as for the non-EGR case. The hardware was optimized for load point B9 and valve timing was based on minimizing the BSFC while maintaining the same λ value and EGR rate as the baseline engine. In this case the valve area was kept identical to the baseline engine but the ratio between the blow-down and scavenging valve area was changed, where the optimal ratio was shown to be 2.7. The corresponding pump loop for the optimized case can be seen in Figure 5.5.

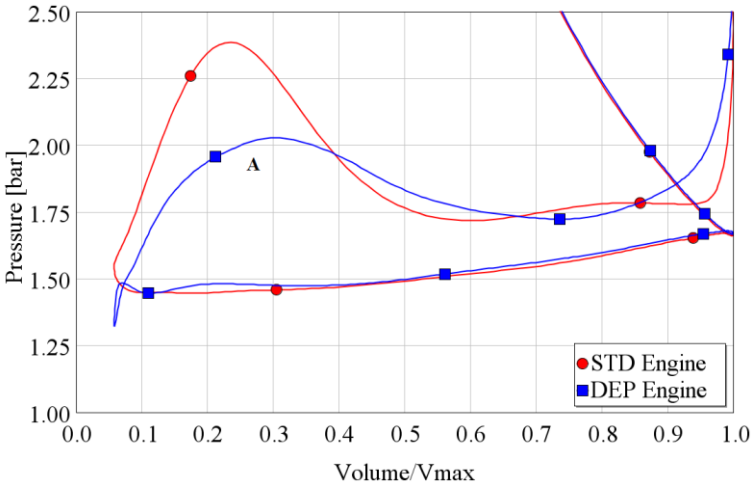

FIGURE 5.5. Cylinder pressure vs. volume for the SR-EGR baseline and DEP engine during the gas exchange loop for load point B9.

For the SR-EGR case the blow-down valve closes and the scavenging valve opens at around point (A). This is much later than the non-EGR case, and is due to the EGR system. Since the blow-down pulse is now used for both turbine work and EGR flow, the duration of the blow-down valve needs to be increased. The duration of the scavenging valve in turn, will affect the pressure in the turbine manifold and will therefore control the ability to drive EGR flow to the intake side. This combined will cause less freedom in choosing a valve timing strategy that maximizes both pumping and turbine work, since valve timing now also controls EGR flow. Another reason for the long blow-down valve duration is due to choking in the valves. If the scavenging valve duration is increased the mass flow through that valve will increase and it will eventually cause a flow restriction. If the scavenge valve size is increased to avoid this choking, the

blow-down valve needs to be decreased, and choking will occur there instead. More details on this can be found in **Paper 2**.

Even with late scavenging it is possible to reduce PMEP also in the SR-EGR case. As can be seen in Figure 5.5, the baseline engine has a large pressure peak at the end of the exhaust stroke. This pressure peak is caused by the blow-down pulse from the next cylinder in firing order. Since the baseline engine has a single entry VGT turbine, this pressure peak will not be mitigated as it was in the non-EGR case. In the DEP case instead, even though it also uses a single entry turbine, it is possible to mitigate this pressure peak since the blow-down valve is closed and scavenging valve opened at this point. Furthermore the DEP concept allows the EGR rate to be controlled without a VGT turbine. The BSFC reduction for this load point is 1 % when using exhaust valve area equal to the baseline engine.

5.3.2 Effects of changing valve area ratio

Since it was clear that the valve size was of high importance for the DEP concept, investigations were made on different settings of blow-down to scavenging valve area ratios. These sweeps were performed for a number of different total valve area settings aside from the baseline area. Three settings are with reduced intake area, where the exhaust area was increased correspondingly. Two settings are with increased exhaust area. These sweeps can be seen together with the baseline engine in Figure 5.6.

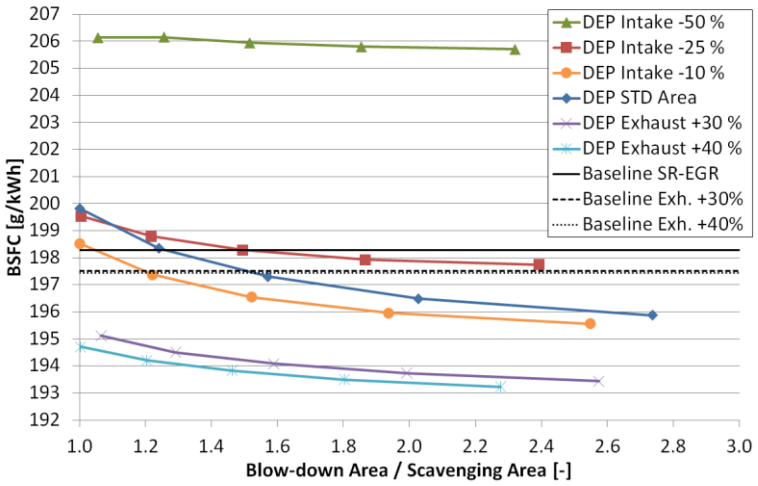

FIGURE 5.6. BSFC as a function of blow-down to scavenging area ratio for different settings of total intake and exhaust area on the SR-EGR engine.

It is clear that decreasing the intake area only gives a benefit for the *DEP Intake - 10 %* case. When decreasing the intake area further, the intake losses will be higher than the benefit of using larger exhaust valves. If the intake area is not changed, but the exhaust area is increased, there is reduction in BSFC. When compared to the baseline engine, one can see that the baseline engine benefits less from increased exhaust area. This is due to the fact that DEP engine, with only one exhaust valve open at a time, is closer to choked flow than the baseline engine. Therefore it becomes much more important for the DEP engine to increase the exhaust valve area. For all area sweeps the trend is the same, increasing the area ratio decreases BSFC. This is due to the long blow-down valve duration requirement for sufficient EGR flow. With long blow-down valve duration, this valve needs a larger area due to the higher mass flow through this valve. As mentioned above, with standard valve area the BSFC reduction is 1 %. When exhaust area is increased by 30 % the BSFC reduction is 2 %.

5.3.3 Full ESC cycle results

Figure 5.7 presents the BSFC reduction for the ESC and 100 kW load points. All load points in the DEP case have identical EGR rate and λ was kept within 4 % of the baseline engine. As mentioned above, the valve timing does not only control the PMEP vs. turbine work trade-off, but also the EGR driving pressure and mass flow. Hence there are more limitations to how valve timing can be set. This also causes the general trends discussed for Figure 5.4 to be invalid for the SR-EGR case. Here the BSFC reduction is basically due to the mitigation of the blow-down pulse interference only, and the pumping work improvement is a function of which EGR rate that is required. Higher EGR rates will lead to longer blow-down valve duration and less improvement of the pumping work. Even so it is possible to achieve an efficiency increase with the DEP concept, and this is done without the use of a VGT turbine.

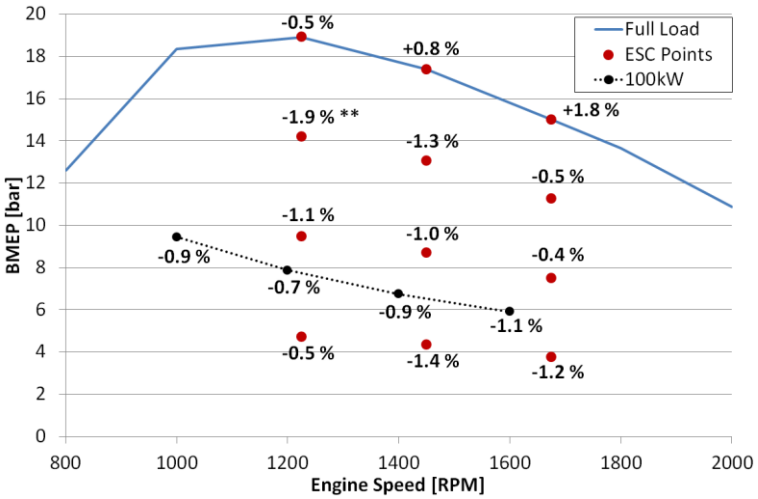

FIGURE 5.7. Change in BSFC for the DEP engine compared to the baseline engine. Load point marked ** exceeds target λ by 17 %.

5.4 The LR-EGR case

This section presents the main results from the LR-EGR case study. Results presented here concerns the pumping loop for the optimized load point, a study of different valve area and ratios, and fuel consumption for all investigated operating conditions. For more detailed information regarding this case, the reader is referred to **Paper 2**.

5.4.1 Gas exchange process

For the LR-EGR case there was no fuel consumption benefit when using standard exhaust valve area for the optimized load point (B9). More results regarding valve area and ratio sweeps will be presented in the next subsection. Only when increasing the valve area by 40 % was there a benefit in PMEP, for reasons that can be explained with the help of Figure 5.8. For the baseline engine there is a small reduction in backpressure with 30 % increased exhaust area, but increasing it further does not affect the backpressure notably. This tells us that the valves are not the restriction anymore, but the backpressure of the turbine. For the DEP engine instead, it is clear that changing the exhaust valve area has a larger effect. With standard exhaust valve area, there is a need to scavenge quite late (SEVO at 300 CAD). Because the area ratio in this case is large, the scavenging valve duration is short to avoid choking in this valve. If the duration is increased the scavenging valve will be choked and there will be re-compression of the cylinder gas at the end of the exhaust stroke. If scavenging area is increased (decreasing the blow-down area), choked flow in the scavenging valve

will be avoided but will occur in the blow-down valve instead. For this case the fuel consumption is identical to the baseline engine.

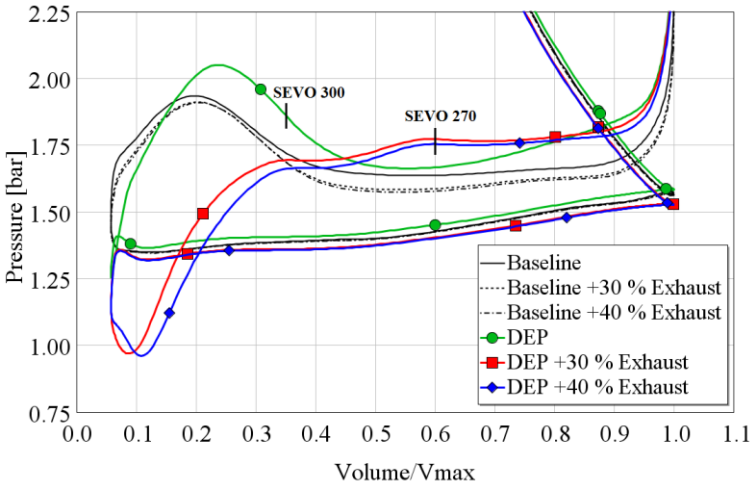

FIGURE 5.8. Cylinder pressure vs. volume for the LR-EGR baseline and DEP engine for standard, +30 % and +40 % exhaust area

If the total exhaust area is increased by 30 % the pump loop changes its appearance. Since the scavenging area can be increased without decreasing the blow-down area, it is possible to scavenging much earlier (SEVO at 270 CAD) without choking the scavenging valve. As can be seen in Figure 5.8 the cylinder pressure drops down to ambient levels as soon as the blow-down valve closes as well. If the area is increased by 40 % the backpressure is reduced further, especially during the scavenging phase. Again it is clear that the DEP concept benefits much more from increased valve area than the baseline engine does.

Even though the increased exhaust area gives higher pumping work, the benefit compared to the baseline engine is still very small in this load point. With +40 % increased exhaust area, BSFC is reduced with merely 0.3 %, To summarize, the DEP concept on the LR-EGR engine does not show the same PMEP benefit as was shown previously in the non-EGR engine and the SR-EGR engine. This is mainly due to generally higher pressure levels compared to the non-EGR case, making this case more prone to choked flow. It is also due to the fact that the twin-entry turbine in the baseline LR-EGR engine already mitigates the blow-down pulse interference, which the VGT did not in the baseline SR-EGR case. In the LR-EGR case, more mass flow is passed through the turbine and compressor compared to the SR-EGR case, thus increasing the required turbocharger power output. This might also have an effect on the overall

performance of the turbocharger in combination with the DEP application, but requires further investigation.

5.4.2 Effects of changing valve area ratio

Valve area and ratio sweeps were also performed for the LR-EGR engine in the same manner as explained in 5.3.2. These sweeps are presented in Figure 5.9. As can be seen, there is no benefit from decreasing the intake area to allow larger exhaust area. Again, this is due to the intake losses being larger than the benefit received with larger exhaust area. With standard and +30 % exhaust area, BSFC is identical to the baseline engine. Not until the area is increased by 40 % is there benefit in a BSFC. The trend when changing area ratio is for the standard area case similar to the SR-EGR case, increasing the ratio decreases BSFC, for reasons explained above. Late scavenging requires larger blow-down area. If scavenging occurs early instead, the scavenging valve will be choked for large ratios, or the blow-down valve will be choked for small ratios. Furthermore, early scavenging (low ratio) requires a smaller turbine to keep the boost pressure, due to the lowered turbine mass flow, which in turn leads to higher backpressure. When increasing the exhaust area by 30 % and 40 % the trend changes, increasing the ratio will increase BSFC. This is because at a certain point there is a tipping effect where the scavenging valve will not be choked and it is more beneficial to scavenge early. With early scavenging the area ratio needs to be smaller since the duration of the scavenging valve is increased and more mass flow will pass through it. More on this can be found in **Paper 2**.

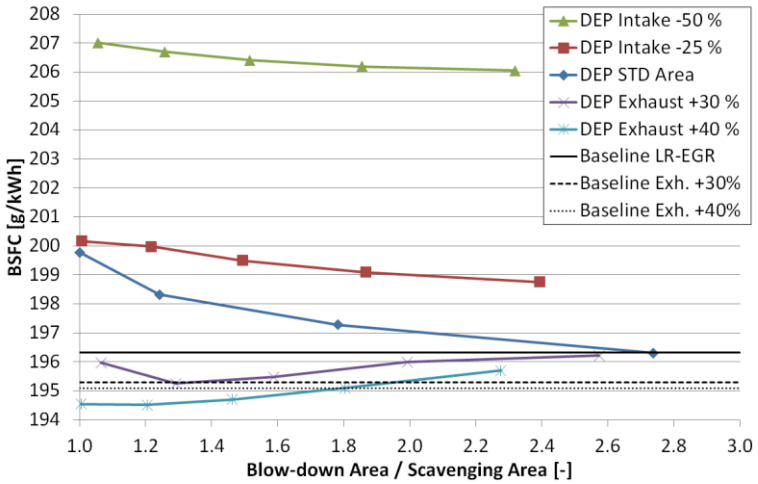

FIGURE 5.9. BSFC as a function of blow-down to scavenging area ratio for different settings of total intake and exhaust area on the LR-EGR engine.

5.4.3 Full ESC cycle results

Figure 5.10 presents the BSFC reduction for the ESC and 100 kW load points. All load points in the DEP case have identical EGR rate and λ was kept within 4 % of the baseline engine. In both the baseline and DEP case the exhaust valve area is increased by 40 %. As can be seen the BSFC benefit is not as promising as in the previous two cases. It is only at low engine speeds that there is a notable gain using the DEP concept since the mass flow restriction in the valves is less of a problem than at high engine speeds where the valves will get choked. As explained above, this is mainly due to two reasons. Firstly, the twin-entry turbine in the baseline engine already mitigates the blow-down pulse interference between the cylinders, which the VGT turbine did not in the SR-EGR case. Secondly, the pressure levels are generally higher in the LR-EGR engine compared to the non-EGR engine, making it more prone to choked flow already at medium load and engine speeds.

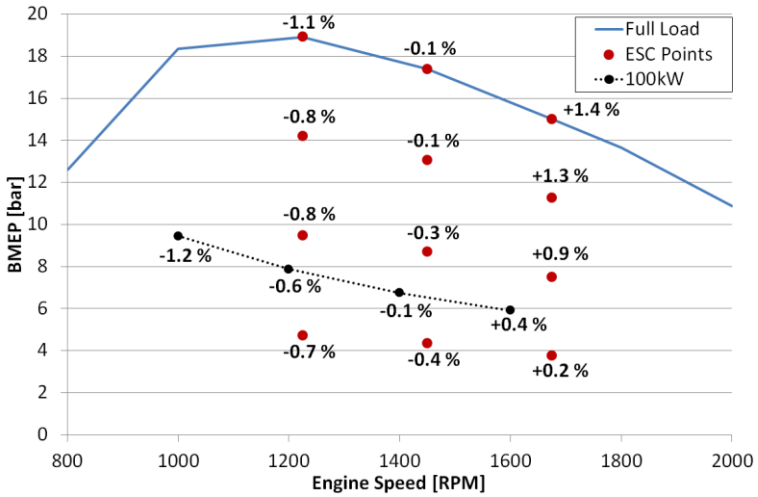

FIGURE 5.10. Change in BSFC for the LR-EGR DEP engine compared to the baseline engine.

5.5 Complementing results

This section presents some complementary results not included in the appended papers. Results presented here concerns transient response, exhaust temperature and a sensitivity study to errors in valve discharge coefficients.

5.5.1 Transient response

To investigate the positive effects of the DEP concept on a load step, transient simulations were performed on the same engine speed as the optimized load point (A7) going from 430 Nm to full load at 1800 Nm. This transient load step

had been measured in a test bed and the model had been calibrated for the original engine in a previous work [44]. The strategy for these simulations was to achieve high boost pressure as fast as possible using proper valve timing. This was done for the DEP engine using the same inertia as the baseline turbocharger and using the real inertia of the new smaller turbocharger. The reason for using these two inertia values was to be able to separate the change in transient performance based on gas exchange effects only and the change based on the fact that a smaller turbine is used for the DEP engine. For both the original and the DEP engine a λ limit of 1.4 has been used during the transient. The result in transient response, or the time to reach full load, is shown in Figure 5.11.

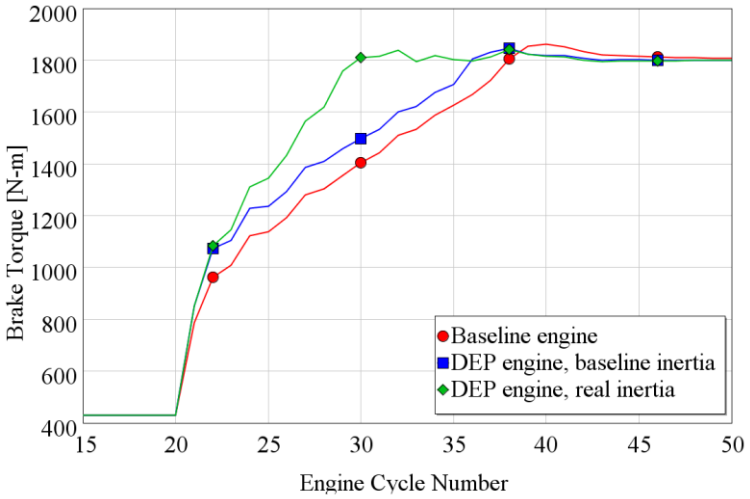

FIGURE 5.11. Comparison of transient response for the non-EGR baseline and DEP engine.

The load step takes 18 engine cycles for the original engine. When using the inertia of the baseline turbocharger on the DEP engine the load step takes 16 engine cycles. This corresponds to an improvement of 11 % on the transient response. The improvement is both based on increased pumping work and the fact that the turbocharger works in a more efficient operating point as has been explained previously. For the second case when the inertia is set to the real inertia value of the new turbocharger the load step takes 10 engine cycles. This corresponds to an improvement of 44 % on the transient response compared to the baseline engine. The reason for the drastic change is obvious; as the inertia is lower the change in turbocharger rotational speed for each cycle is equally larger for the same amount of excess torque produced by the turbine. The conclusion of this study is that the biggest improvement to the transient response comes from the fact that a smaller turbine is being used and not the DEP concept in itself, even if that improves the response as well.

The valve timing strategy for the simulations was as follows. At low load the valve timing that gives the lowest BSFC has been used. As the transient starts the valve timing is changed to very late scavenging to send all exhaust gases to the turbine. Compare this to closing a VGT during a transient. Even with very late scavenging, the scavenging valve opens shortly at the end to puncture the cylinder pressure. Looking at Figure 5.1 for example it is clear that the cylinder pressure increases quite fast at the end of the exhaust stroke for the original engine. If the scavenging valve is instead opened the cylinder pressure will drop to close to ambient level right before the intake valve opens and there will be a better filling of the cylinder. When the target load is reached the valve timing is again changed to a setting that gives the lowest BSFC for that specific load. Investigations of changing the valve timing strategy also during the transient were performed but without improving the result notably.

5.5.2 Exhaust temperature

Paper 1 presents results of change in in-cylinder and exhaust temperature due to the DEP concept for the non-EGR case. The conclusion was that no drastic changes occurred due to reduced residuals, since the amount of residuals was low in the baseline case to start with. However, reduced injected fuel mass (for the cases with increased efficiency) resulted in lower peak cylinder temperature and lower exhaust temperature for the DEP case. Furthermore, a more efficient expansion over the turbine in the DEP case lowered the exhaust temperature for some load points. There was no drastic change in turbine inlet temperature. However, what was not presented was the temperature difference between the blow-down and scavenging manifold, which can differ quite much. Table 5.1 presents the mass averaged temperature difference between the manifolds for all three studied cases. Note that the speed and torque differs between load points A and B and cannot be compared to each other directly (see Figure 4.1 and Figure 4.2).

For all cases the blow-down manifold has a higher temperature than the scavenging manifold. When the blow-down valve opens the cylinder gas still has high temperature and pressure but as the cylinder gas expands the temperature will decrease, according to adiabatic expansion, resulting in lower temperature during the scavenging phase. For the non-EGR case the trend is that at higher loads the temperature difference is larger. This is due to the trend discussed in section 5.2.3, at higher loads scavenging occurs earlier causing a larger part of the cooler scavenging flow bypass the blow-down manifold. This leads to higher temperature differences for early scavenging. With late scavenging more of the cooler scavenging flow will pass through the blow-down manifold, and the

temperature difference will be evened out more. This can especially be seen for the SR-EGR case, where all load points runs with very late scavenging causing the temperature difference between the manifolds to be much lower than the non-EGR case. The LR-EGR case runs with scavenging timing somewhere in between the other two cases, resulting in a temperature difference that is generally higher than the SR-EGR case, but lower than the non-EGR case. This ability to control the exhaust gas temperature between the manifolds, with valve timing strategies, could be an important aspect to consider from a thermal management or aftertreatment point of view.

TABLE 5.1. Difference in mass averaged temperature between blow-down and scavenging manifold for the non-EGR, SR-EGR and LR-EGR case. Positive ΔT means higher temperature for blow-down side.

Load Point	non-EGR ΔT [K]	Load Point	SR-EGR ΔT [K]	LR-EGR ΔT [K]
A1	249	**B1**	121	144
A2	97	**B2**	80	100
A3	294	**B3**	71	80
A4	190	**B4**	72	83
A5	269	**B5**	53	64
A6	109	**B6**	59	55
A7	182	**B7**	54	64
A8	239	**B8**	78	94
A9	171	**B9**	84	102
A10	215	**B10**	95	119
A11	164	**B11**	135	171
A12	87	**B12**	103	139
A13	143	**B13**	102	116
A14	327	**B14**	117	133
-	-	**B15**	120	171
-	-	**B16**	113	220

Table 5.2 presents the mass averaged temperature difference between the turbine outlet and scavenging manifold for all three studied cases. Here it is clear that the temperature difference is smaller due to the temperature drop over the turbine. The general trend is that load points with high load and speed have a higher temperature drop over the turbine (up to 120 °C). In the non-EGR case, which has a large temperature difference between the two manifolds, the temperature is still higher at the turbine outlet compared to the scavenging manifold. Only for the highest load and speed points is the temperature difference almost equalized. For the two EGR cases, which have a smaller temperature difference between the two manifolds, the temperature difference between the turbine outlet and scavenging manifold is almost equalized for all cases.

TABLE 5.2. Difference in mass averaged temperature between turbine outlet and scavenging manifold for the non-EGR, SR-EGR and LR-EGR case. Positive ΔT means higher temperature for turbine outlet.

Load Point	non-EGR ΔT [K]	Load Point	SR-EGR ΔT [K]	LR-EGR ΔT [K]
A1	207	**B1**	85	103
A2	88	**B2**	43	57
A3	224	**B3**	33	30
A4	146	**B4**	22	19
A5	183	**B5**	32	40
A6	78	**B6**	31	21
A7	115	**B7**	16	18
A8	133	**B8**	6	16
A9	92	**B9**	22	39
A10	95	**B10**	45	61
A11	51	**B11**	40	84
A12	20	**B12**	10	44
A13	25	**B13**	1	9
A14	276	**B14**	-12	7
-	-	**B15**	-7	51
-	-	**B16**	-5	113

5.5.3 Sensitivity to change in C_D

When changing the valve area the valve discharge coefficient, C_D, remains the same as the standard valve size. Since it is clear that changing valve area has a big effect on the DEP concept it was of importance to investigate how potential errors or a change in C_D would affect BSFC. This study was done for the non-EGR case for the full load curve at different engine speed, since it is at full load valve choking occurs. The results for the baseline case can be seen in Figure 5.12, where C_D was increased and decreased by 10 and 20 % respectively.

It can be seen that low engine speeds are affected the least, while at high engine speeds a change in C_D have a larger effect on BSFC. Higher engine speed gives higher mass flow and more restriction in the valves. Since the valves are closer to choked flow at higher engine speeds, a small change in C_D has a larger impact than it has on lower engine speeds. It is also clear that BSFC is increased more with lower C_D than what it is decreased with higher C_D. This is due to the fact that once choked flow is avoided a further increase in C_D does not affect the pumping work as much. The closer the valves are to choked flow, the more dramatic the effect will be of decreasing C_D. For the highest engine speed, an increase or decrease of C_D by 20 % results in change of BSFC by -1 % and +2 % respectively.

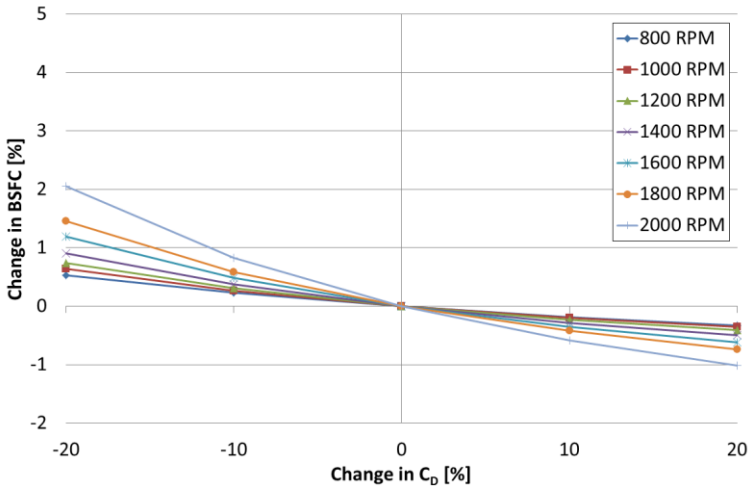

FIGURE 5.12. The effects on BSFC by changing the exhaust valve discharge coefficient C_D for the baseline non-EGR engine at full load for different engine speeds.

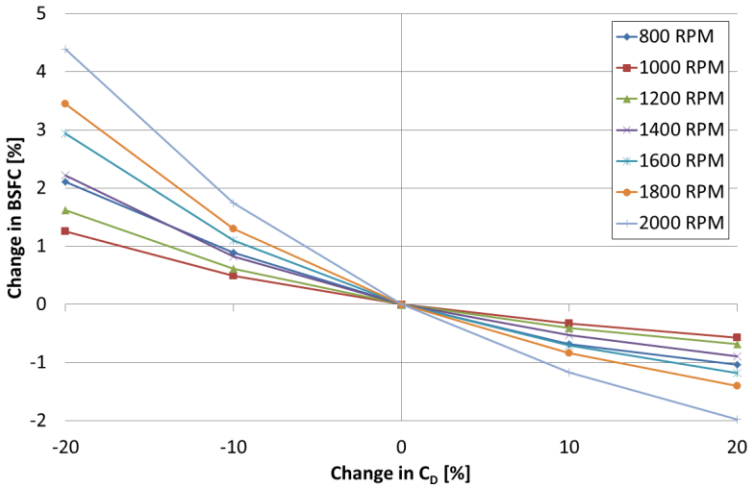

FIGURE 5.13. The effects on BSFC by changing the exhaust valve discharge coefficient C_D for the DEP non-EGR engine at full load for different engine speeds.

The results for the DEP case can be seen in Figure 5.13, where C_D was increased and decreased by 10 and 20 % respectively. Here the trends are the same, high engine speeds are more sensitive to a change in C_D. The case of 800 RPM is an exception to this trend. This is due to a very long scavenging valve duration, which causes a large restriction in the scavenging valve which makes it more sensitive to changes in C_D. It is clear also here that the slopes of the curves are steeper for reduced C_D than for increased C_D. As explained above, the closer the

valve gets to choked flow, the more dramatic is the effect of lower C_D. When comparing Figure 5.12 and Figure 5.13, it is clear that a change in C_D has a much larger impact for the DEP case than for the baseline case. This is in analogy with changing valve area as explained above. The DEP concept is much more sensitive to changes in valve area or C_D since only one valve is open at a time which makes them more likely to choke. For the highest engine speed, an increase of C_D by 20 % results in a change of BSFC by -2 % and a decrease of C_D by 20 % results in a change of BSFC by and +4.4 %.

5.6 Discussion

In the previous sections the three cases have been discussed individually and some differences have been highlighted. This section will instead present a summary and discussion on what they have in common, that is inherent in the DEP concept. All cases have shown some benefit to pumping work and results from this have been presented above. However, this section will focus on aspects of DEP that are not only connected to the pumping work and some conditions are presented for when DEP could be a feasible concept to be considered.

(a) Choking valves

From the results presented here and in the appended papers it is clear that the exhaust valve size is of paramount importance to the DEP concept. Since only one exhaust valve is open at a time the DEP concept becomes very sensitive to flow restriction in the valves. A comparison between the three cases shows that even though the EGR system was disconnected from the DEP system in the LR-EGR case, valve timing strategies were still limited by the valve size. With small total exhaust area there was a need for larger blow-down valve size and duration compared to the non-EGR case. With increased valve area, both the scavenging valve area and duration could be increased. Hence, there is a strong interaction between valve size and duration, which in turn limits valve timing strategies. Furthermore, it is also clear that there was a need for larger exhaust area on the LR-EGR case compared the non-EGR case to get an efficiency benefit. This is due to the generally higher boost levels in the LR-EGR case, leading to larger flow restriction in the valves already at medium engine speeds. As shown in **Paper 2**, with lower boost demand the choking could be partly avoided and a larger increase of pumping work could be achieved. To summarize, choking valves is the single biggest issue for the concept and needs to be addressed either through increased valve area or decreased flow losses across ports and valves.

(b) Down-speeding

Another solution to the choked valves is to run on lower engine speeds. As shown above, there was a larger BSFC benefit at lower engine speeds than at high engine speeds. Higher engine speed brings higher mass flow and the exhaust valves are more likely to choke when only one valve is opened at a time. At lower engine speed this becomes less of a problem and there is more to gain from the DEP concept. Consequently, the DEP concept should be considered with simultaneous down-speeding. If down-speeding is combined with a dual clutch gearbox, which narrows the engine operating speed range, choked flow could probably be avoided to a large extent.

(c) Wastegate

The DEP concept makes the use of a traditional wastegate obsolete. Instead of opening a wastegate to limit boost pressure, the scavenging valve has the same function. Furthermore, a smaller turbine can be used and at high loads the scavenging valve opening can limit the turbine mass flow instead. With a wastegate operated turbine there will still be a backpressure which is avoided in the DEP case. Hence, the scavenging valve works as an improved wastegate.

(d) Variable geometry turbine

As explained more in detail in **Paper 1**, the DEP concept adds the functionality of a VGT but with a fixed geometry turbine. With a VGT the geometry of the turbine is changed to adapt to the mass flow. In the DEP case the mass flow is adapted to the geometry of the turbine, by means of suitable valve timing. This allows the turbine to always work in a condition with high efficiency for a wide range of load points.

(e) Transient response

As explained above, the combination of using a smaller turbine with the VGT functionality of the DEP concept, the transient response of the engine is improved. Firstly, a smaller turbine has a lower inertia which shortens the speed-up time of the turbocharger. Secondly, the smaller turbine in combination with a late scavenging valve timing strategy will results in a higher starting boost pressure than in the baseline case. Consequently, more fuel can be injected before reaching the λ limit, resulting in a faster torque buildup.

(f) Pulse interference

Adding to the functionality described in *(d)* the DEP concept makes it possible to avoid pulse interference between cylinders even if a single entry turbine is used. Since it is only the scavenging valve that is opened at the end of the exhaust stroke, the blow-down pulse of the next cylinder in firing order will not

interrupt the scavenging process. In the SR-EGR case where the baseline engine uses a single entry VGT, this was shown to have a big impact on the pumping work.

(g) Residuals

Continuing on the above; the fact that only the scavenging valve is open, at the time of intake valve opening, will reduce the residual gas fraction in the cylinder. Since the intake manifold pressure will always be higher than the scavenging manifold pressure, there will be an improved scavenging process of burnt residuals. This in turn allows for longer overlap duration between scavenging and intake valve for optimal volumetric efficiency. Lower amount of hot residuals results in lower in-cylinder temperature at start of combustion, which could have an effect on *NOx* formation. Furthermore, this will reduce the risk of knock in Otto-type applications and makes it possible to advance the spark timing and increase compression ratio for increased efficiency. Reduced residuals could also reduce the risk of pre-ignitions which was apparent in a diesel dual fuel application of the Scania engine [48].

(h) Scavenging of HC emissions

Emissions of un-burnt hydrocarbons due to release from the crevice volumes in the end of the exhaust stroke is a well-established mechanism in SI engines [4]. This can also be seen in time resolved measurements in the above mentioned diesel dual fuel application and the amount of emissions exhausted during this phase can be substantial [49]. With the DEP concept it could be possible to direct the exhaust flow from the scavenging manifold (which contains these crevice hydrocarbons) as EGR and in this way re-burn the HC.

(i) Thermal management

The DEP concept could be considered from a thermal management point of view as well, to aid the aftertreatment system. With variable exhaust valves it could be possible to combine DEP with different measures to increase the exhaust temperature, such as early exhaust valve opening/closing or exhaust valve opening during the intake stroke [50]. Furthermore, with the possibility to let all exhaust flow bypass the turbine, which acts as a heat sink, catalyst light-off time could be reduced or filter regeneration could be made more efficient.

(j) Internal EGR

In analogy with above, the external EGR system could be replaced with internal EGR. This could for example be done through a late scavenging valve closing, combined with late intake valve opening (since the intake pressure is higher than the scavenging manifold pressure). Early scavenging valve closure could also be

used to introduce internal EGR. Alternatively, the blow-down valve could be opened during the intake stroke. In this case the intake valve can keep its original opening time since there is a backpressure in the blow-down manifold. However, the issues that uncooled internal EGR brings in terms of increased in-cylinder temperature, which affects volumetric efficiency and emission formation, need to be addressed further.

CHAPTER 6

Conclusions and Outlook

This chapter will present the main conclusion that can be drawn based on the results presented in the thesis and the appended papers. Suggestions for possible future work will be given as well.

6.1 Conclusions

Simulations of the DEP concept show potential to decrease fuel consumption. With proper valve timing and turbocharger matching, a part of the backpressure created by the turbine can be avoided during the last part of the exhaust stroke and pumping work is increased. However, it is apparent that the efficiency gain is largely dependent on the exhaust valve size and valve configuration. Since only one exhaust valve is opened at a time, the DEP engine is more prone to choke during the exhaust phase. To avoid choked flow in the valves, the DEP engine needs to either run at lower engine speeds or have its exhaust valve size increased. Exhaust valve size and configuration is also largely dependent on the valve timing strategies. Depending on the individual duration of the two exhaust valves, their size needs to be adapted accordingly to avoid choked flow.

By adjusting valve timing towards either late or early scavenging, the trade-off between turbine power and pumping work can be controlled. This in turn allows control of boost pressure. This essentially gives the fixed geometry turbine the function of a variable geometry turbine, since the inlet conditions can be controlled with the valve timing. For low engine speed and mass flow essentially all exhaust gas can feed the turbine. At high engine speed and load, the mass flow can be limited by opening the scavenging valve. This has the same effect as a wastegate, but without the added backpressure. Furthermore, since the low pressure and low mass flow phase of the exhaust stroke is not passed to the turbine, it operates with high efficiency to a greater extent. The possibility of controlling the mass flow to the turbine over the whole engine speed and load map greatly extends the range for which the turbine operates with high efficiency.

The DEP concept benefits from using a smaller turbine than baseline. This greatly improves the boosting capabilities at low engine speeds, improving the low-end torque curve. The smaller turbine (lower inertia) together with the functionality of controlling the mass flow to the turbine, which in turn controls the boost pressure, greatly improves the transient response during load steps.

By removing the backpressure during the overlap period between intake and scavenging valve, the residual gas fraction was reduced. The reduced backpressure also allows larger overlap durations without increasing residuals. With proper adjustment of the overlap duration the ability for more complete cylinder filling and higher volumetric efficiency was possible.

The choice of EGR system proved to impact the outcome of the DEP concept. When no EGR is used, the valve timing strategies are based solely on the optimization of the turbine work vs. pumping work trade-off. With the SR-EGR system, the valve timing strategies also need to include control of EGR driving pressure. This greatly hampered the flexibility of the valve timing strategies. However, since a fixed geometry turbine could be used instead of a VGT, blow-down pulse interference could be reduced, increasing pumping work. With a LR-EGR system, the valve timing strategies are again independent of the EGR flow control and can be adjusted for optimal efficiency as in the non-EGR case. However, due to higher pressure levels on the LR-EGR engine causing choked flow already at medium engine speeds and load, a fuel efficiency benefit was not found until the exhaust valve size was greatly increased.

6.2 Suggestions for future work

Based on the results and conclusions presented in this thesis, these are some suggestions on possible paths for future work:

- The simulations should be validated against measurements on a DEP prototype engine. As a starting point this could be done with a fixed cam based valve train, with a discrete set of different cam lobes before a VVT system is implemented to keep the cost down.
- Investigations could be performed on how much the exhaust valves can be increased or how much the flow characteristics of the ports can be improved when implementing DEP on a real cylinder head.
- Based on the results, DEP should be considered with simultaneous down-speeding. Hence, the simulations could be re-run with the same optimization procedure but with the purpose of optimizing the concept

at a lower speed range. Possibilities of combining this with a dual clutch gearbox could also be included.

- Development of control strategies for the variable valve timing could be investigated. Valve timing was shown to largely depend on speed and load and one can imagine that engine calibration or control strategies during transient operation or during real driving cycles could possibly be quite complex.
- A complete mapping of the variable valve timing strategies could be performed for the whole operating range of the engine, both stationery and transient. With this as a base it could be investigated if it would be possible to make some simplifications on the variability of the valves, to avoiding the need for a FVVT.
- An investigation could be made on the cost trade-off between the extra energy required to operate a specific VVT system and the fuel efficiency that is gained when implementing DEP.
- Further investigations should be made as to how turbine efficiency is affected by the different exhaust pressure pulse train that the DEP concept generates. According to GT-Power calculations it is possible to maintain the same turbine power with lower mass flow, but this needs to be verified by extending the performance map under pulsatile flow.
- Further studies on valve modeling should be performed. In this study the two exhaust valves are allowed to change size. Furthermore, compared to a standard engine the two exhaust valves open at completely different cylinder pressure. However, both exhaust valves use the same discharge coefficient which is measured for very low pressure ratios. A more in-depth analysis how this affects the real application is needed.

The suggestions from the two last bullets could be investigated by means of measurements on the CCGEx pulsating flow rig. There, both turbine performance measurements for different pulse modulation and discharge coefficient measurements for different pressure ratio could be performed. Preferably the discharge coefficients could be instantaneously measured during the valve opening and closing event to investigate dynamic effects. The groundwork for these kinds of studies and the corresponding simulation in GT-Power of the same experiments has already been started. Simulations of the pulsating flow rig, with and without a turbine have shown good agreement. More information on these simulations can be found in Appendix A.

APPENDIX A

Pulsating Flow Rig

This appendix deals with the 1D simulation with GT-Power [40] of the pulsating flow rig developed within the CICERO laboratory at KTH CCGEx. The purpose of this study was to investigate the possibilities of correctly simulating the behavior of the highly pulsatile flow in the rig both with and without a turbocharger. The flow rig model can then be used as a complementary tool for future experiments concerning gas exchange related topics in the flow rig laboratory. A short introduction to the experimental and simulation setup will be given and results from measurements and calculations on experiments both with and without a turbocharger will be presented.

A.1 Experimental setup

The flow rig system consists of compressors and tanks feeding the rig with compressed air of up to 0.5 kg/s at 5 bar through a pipe system. Upstream the flow rig, the air is heated in an electric heater to avoid the temperature from falling below the dew point downstream a turbine for instance. The flow is then directed to a ball valve where the ball, planed on two opposite sides, is able to rotate within a circular pipe. The ball is rotated with a frequency controlled AC motor causing the valve to open twice per revolution. In this way a pulsatile flow downstream the ball valve (chopper) is achieved. A by-pass line with a control valve in parallel with the chopper allows the pulse amplitude to be varied. Downstream the chopper it is possible to mount any measurement devices. For more details regarding the flow rig and its measurement equipment the reader is referred to [51]. Two sets of experiments were performed with corresponding simulations in GT-Power. The first concerns pulsatile flow with a contraction downstream the chopper, and the second concerns pulsatile flow with a turbine downstream the chopper.

A.1.1 Setup with contraction

The first set of experiments was performed with a contraction, which acts as turbine, downstream the chopper. Static pressure was measured upstream the chopper and at three locations downstream the chopper (p_2 not used due to faulty sensor), while temperature was measured directly downstream the chopper, see Figure A.1. Temperature was measured both averaged and time resolved. Average mass flow rate was measured upstream the electrical heater, at the laboratory inlet line, with a hot film flow meter. Measurements were performed both without and with a contraction (25 % of the pipe area) at the outlet orifice. Tests were run for two different pulse frequencies (60 and 100 Hz) and mass flow rates ranging from 58 to 146 g/s for a total of 22 cases.

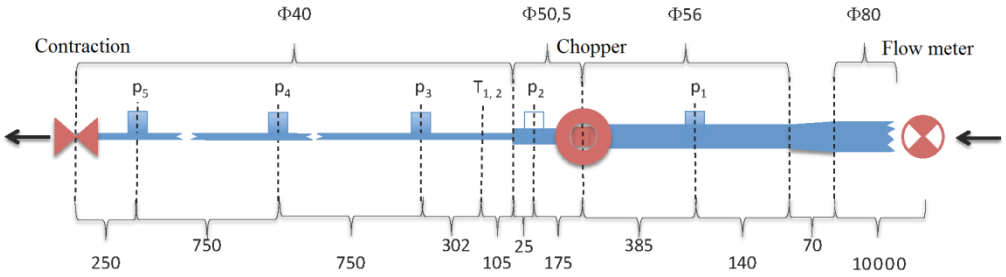

FIGURE A.1. Schematic representation of the flow rig setup and its instrumentation with the contraction. p_i is pressure sensors (i=1, Kistler 4045A10; i=3-5 Kistler 4045A5). T_i is temperature sensors (i=1, thermocouple; i=2, cold wire). Lengths and diameters in [mm].

A.1.2 Setup with turbocharger

The second set of experiments were performed with a turbine connected downstream the chopper, see Figure A.2. Time resolved static pressure (p_{1T}, p_{2T}), total pressure (p_{01T}), stagnation temperature (T_{01T}, T_{02T}) and flow velocity (U_{1T}, U_{2T}) were measured upstream and downstream the turbine. Average pressure and temperature were measured on the compressor side but are not included in this study. Time resolved pressure was measured with piezo-resistive pressure transducers and temperature with cold wires (Ø5µm wolfram). The time resolved flow velocity upstream and downstream the turbine was measured with vortex shedding meters. With measured pressure and temperature the density was calculated using the gas law. Density and velocity gives in turn the instantaneous mass flow rate. Measurements were performed for three different pulse frequencies (40, 60 and 80 Hz) and four mass flow rates (55, 80, 105, 130 g/s). The turbocharger used was a Garret GT17 NS. For more details regarding the instrumentation, measurement techniques and results from these measurement the reader is referred to [51].

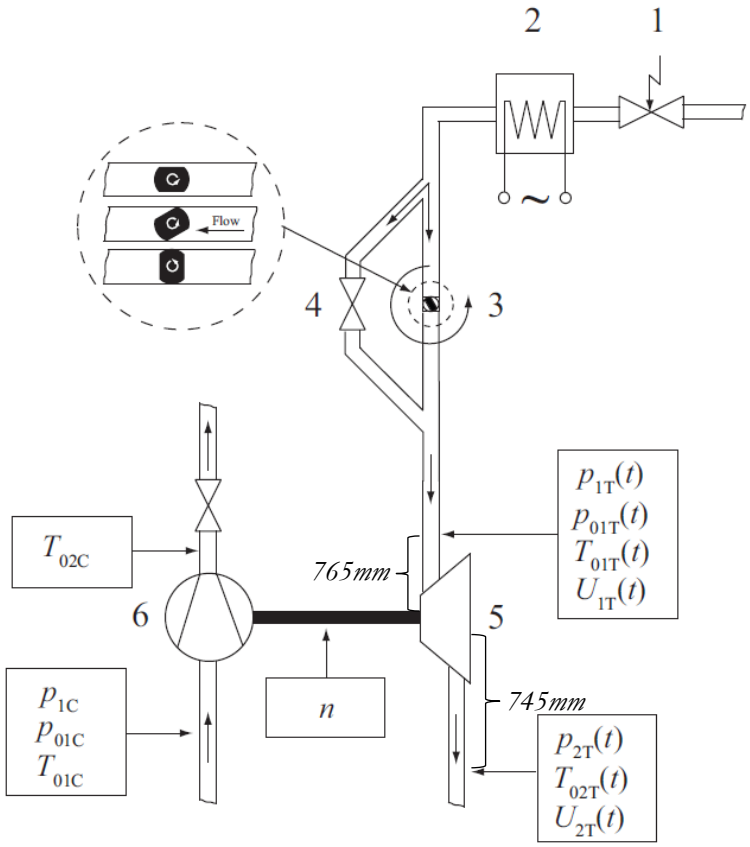

FIGURE A.2. Schematic representation of the flow rig setup and its instrumentation with a turbocharger. 1. Inlet regulator valve, 2. Electric heater, 3. Pulse generator, 4. By-pass line, 5. Turbine, 6. Compressor [51].

A.2 Simulation setup

The GT-Power model of the flow rig is built up from pipe objects 3 meters upstream the chopper and 10 m downstream the turbine. In the case with the contraction the model ends at the contraction location. The by-pass line (not used in the measurements) and electrical heater is not included in the model. The pipe objects represent the real geometry of the flow rig piping system and surface roughness and heat transfer coefficients are set according to values for smooth steel. The turbine is modeled with the corresponding performance map for the Garret GT17 NS turbocharger according to standard methods, see Section 3.4. More details on the modeling of the chopper and a short note on the boundary conditions used will be presented below.

A.2.1 Chopper modeling

As described in Section 3.3, all valves are modeled in GT-Power through the input of discharge coefficients, C_D. This can be a single value or a function of one or more parameters like valve lift, angle, pressure and so forth. For simple geometries the coefficient can be set by the code (tabulated from database of known geometries) but for more complex geometries, e.g. valves, C_D needs to be entered by the user. The discharge coefficient for the chopper valve was calculated from Equation 3.6-3.8. Input for these calculations came from measurements performed on the flow rig under steady state conditions for a range of mass flow rates, valve angles and pressure ratios over the chopper, see Figure A.3. Average mass flow rate and temperature was measured upstream the chopper and static pressure was measured directly upstream and downstream of the chopper. For total inlet pressure, the dynamic pressure was estimated with the help of the mass flow rate, temperature and pipe area. With these measurements, Equations 3.6-3.8 were used to calculate C_D as a function of valve angle and pressure drop over the chopper. Due to measurement uncertainties at low mass flow rates and pressure drops, C_D did not show a smooth trend in this range and a curve fit was applied as input for GT-Power. When calculating the mass flow rate, using the curve fit of the C_D values, result still show good agreement with the measured mass flow rate, see Figure A.3.

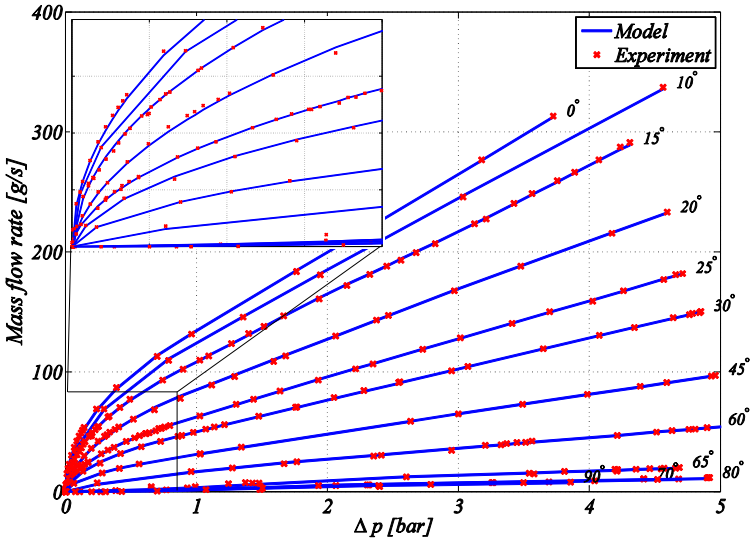

FIGURE A.3. Measured and modeled values for mass flow rate as a function of valve angle and pressure drop over the chopper.

A.2.2 Boundary conditions

Inlet boundary conditions are averaged temperature and instantaneous pressure from measurements for the case with the contraction. In the turbocharger case the instantaneous pressure upstream the chopper was not available but the average mass flow rate was used instead. Outlet boundary conditions are in both cases ambient pressure and temperature. For the pipe wall an isothermal boundary condition was used where the temperature was set equal to the measured inlet air temperature. The turbine was set with a fixed rotational speed given by the measured turbocharger speed.

Since the instantaneous pressure upstream the chopper was not available for the simulations of the turbocharger setup, this model required some tuning. To capture the correct pulse amplitude downstream of the chopper the length of the pipe upstream the chopper had to be adjusted. Since the distance between the inlet boundary and the chopper will greatly affect the pulse propagation behavior, caused by the reflecting pressure pulses as the valve closes, the length required tuning for good agreement with measurements. Note that this tuning to get the correct pressure pulse propagation was not required in the case with the contraction since the measured instantaneous pressure could be used as boundary condition. Note also that no further tuning or calibration of the model, in terms of pressure loss, heat transfer, friction coefficients etc., was performed for the results presented below.

A.3 Results and discussion

This section will present some examples of results from both the case with the contraction and with the turbocharger to show the validity of the simulations when compared to measurements.

A.3.1 Contraction results and discussion

A highlight from the results when comparing the measured temperature and pressure to the simulated ones, are presented in Figure A.4. Temperature and pressures are the measured and simulated values at the locations according to Figure A.1. Since the measured pressure upstream the chopper (p_1) is used as the boundary condition, the simulated value is the same and is not included. The two cases presented are with 60 and 100 Hz pulse frequency and a mass flow rate of 110 g/s. The contraction area is 25 % of the connecting pipe area. It is clear that the GT-Power simulation agrees very well with the measurements and is able to capture both the pressure waves generated by the chopper and the reflecting waves caused by the restriction. There are slightly larger discrepancies between

measured and simulated temperature, at least for the 60 Hz case, but the general trend is captured within reasonable limits. The same agreement can be observed for the remaining 20 cases where the simulation model is able to capture the pressure pulsations also for other mass flow rates and contraction areas. For higher mass flow rates (>146 g/s), the discrepancy is slightly larger between measurements and simulations, this is especially true for the temperature, but the reason for this has not yet been found. It could be an error in the discharge coefficient at choked conditions but needs to be confirmed by re-evaluating the measurements and calculations performed for establishing the C_D values.

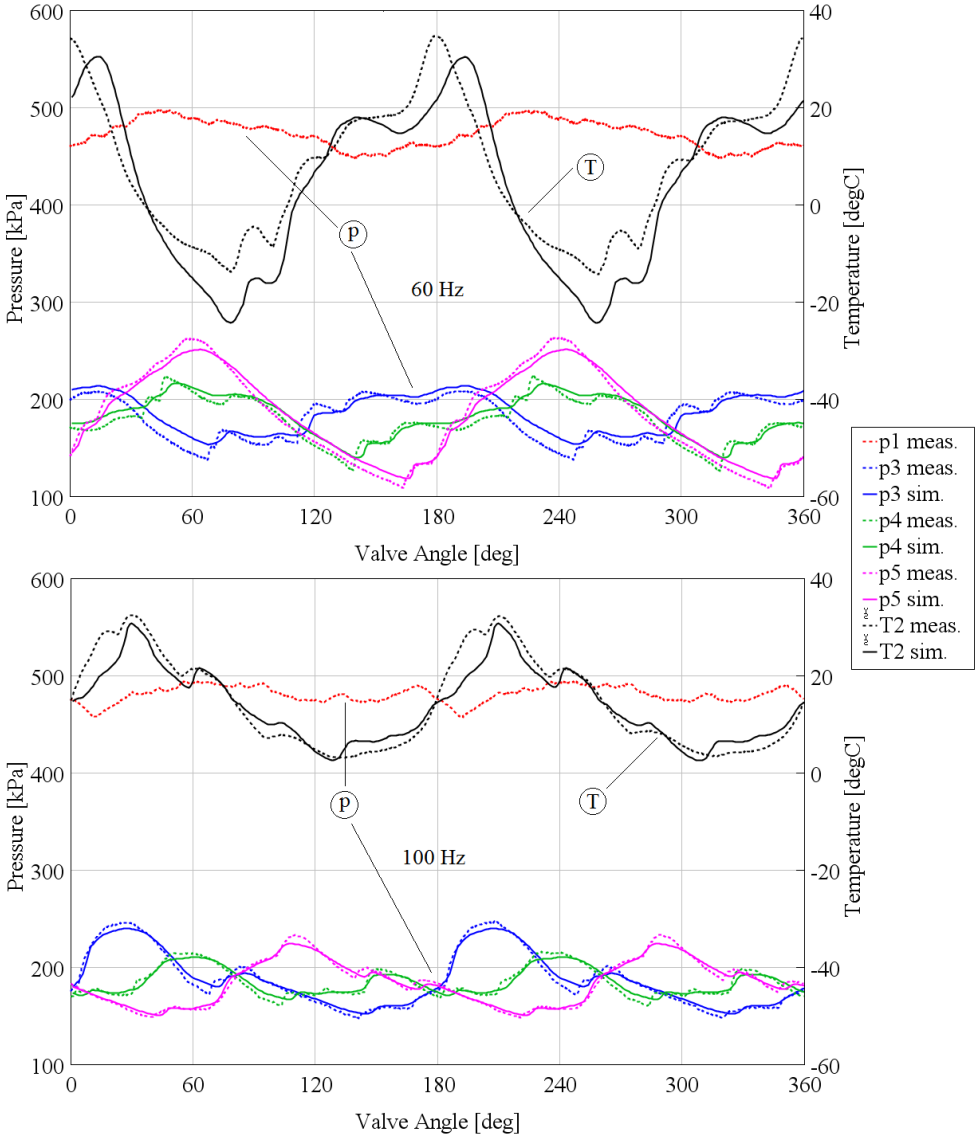

FIGURE A.4. Measured and simulated temperature and pressure at three locations downstream the chopper for two pulse frequencies (60 and 100 Hz), with mass flow rate of 110 g/s and outlet contraction of 25 % of total pipe area.

A.3.2 Turbocharger results and discussion

A highlight from the comparison between measurement and simulations for the case with a turbocharger downstream the chopper is presented in Figure A.5. Here instantaneous mass flow rate upstream the turbine is shown as a function of the total to static pressure ratio over the turbine. Simulated values are from the same location upstream and downstream the turbine as the measurement locations. The performance map entry are the mass flow and pressure values at the location of the turbine object in the GT-Power model. Note that the turbine object is not represented by any volume and consequently there is no hysteresis effect caused by mass accumulation in the volute. The reason there is a slight area enclosed by the simulated trace is due to the slope of the pressure pulse being steeper on the downside than the upside of the pulse peak, thus causing a small phase shift between the mass flow and pressure trace. These values are the input for the turbine performance map which in turn predicts the turbine efficiency and power.

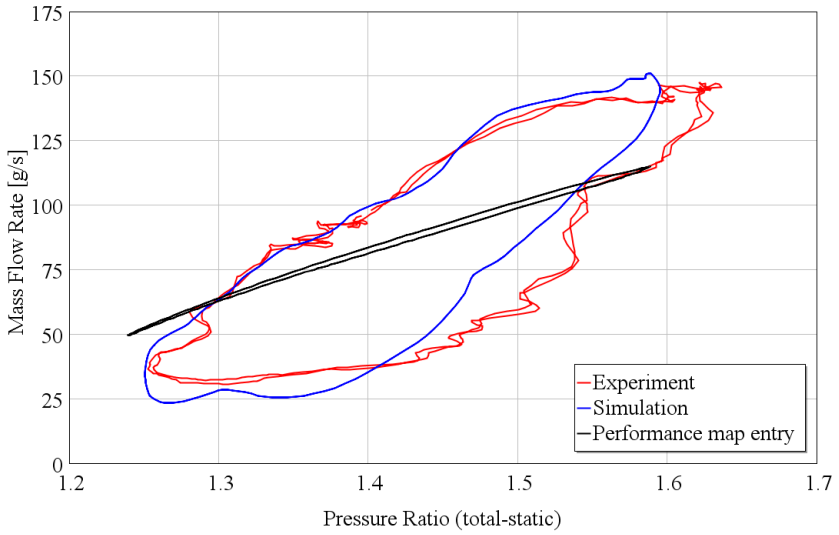

FIGURE A.5. Measured and simulated mass flow rate vs. total to static pressure ratio, with average mass flow rate of 80 g/s and pulse frequency of 60 Hz. *Experiment* and *simulation* values from measurement location, *performance map entry* from turbine location.

As can be seen in Figure A.5 the simulations agree quite well with the measurements. The discrepancies are mainly from the simulated pressure traces and not the instantaneous mass flow rate, which agrees very well with measurements. The reason the agreement with the pressure traces is worse is due to the inlet boundary condition discussed above. Since a fixed averaged mass flow rate is used as inlet boundary condition, it is difficult to match the

simulation to the measurements solely by tuning of the inlet pipe length for proper pulse propagation. Furthermore, this result in a situation where this pipe tuning is required for all cases individually, e.g. the pipe length used for this case will not give good agreement for other mass flow rates or frequencies. This issue could likely be resolved by using instantaneous pressure measurements upstream the chopper as inlet boundary condition, as was done in the case with the contraction.

A.4 Summary

To summarize, the GT-Power model of the flow rig appears to agree well with measurements when comparing both the case with the contraction and with the turbocharger. This is done without any major tuning or calibration of the model. However, there was a need to measure the discharge coefficient of the chopper valve for accurate prediction of the pulse amplitude. Issues with a correct prediction at higher mass flow rates were found and could be due to erroneous C_D values for the chopper at critical flow cases. Furthermore, it was clear that an instantaneous pressure measurement upstream the chopper should preferably be used as an inlet boundary condition. This is to avoid the need for pipe length tuning for individual cases, and a correct prediction of pressure pulse propagation both upstream and downstream the chopper. In addition to this, it would be of interest to measure the instantaneous mass flow rate closer to the turbine. In this setup the mass flow rate was assessed quite far from the turbine inlet and outlet (app. 15 pipe diameters). This makes it difficult to analyze the mass accumulation effect in the relatively small volute volume, since the pipe volume between measured mass flow rate and turbine is much larger.

With this GT-Power model in combination with future experiments on the pulsating flow rig with instantaneous mass flow rate measurements, further studies can be made on turbocharger performance behavior under un-steady conditions. For future studies it would be of interest to also measure turbine power or torque. Investigations could then be made if GT-Power can correctly predict the efficiency and power produced also under pulsating conditions. If not, the error caused by the quasi-steady assumption and negation of the mass accumulation effect in the volute could be assessed further.

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