Effects of Multiple Injectors on Spray Characteristics and Efficiency in Internal Combustion Engines

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Abstract

High-pressure internal combustion engines promise high efficiency, but a proper injection strategy to minimize heat losses and pollutant emissions remain a challenge. Previous studies have concluded that two injectors, placed at the piston bowl's rim, simultaneously improve the mixing and reduce the heat losses. The two-injector configuration further improves air utilization while keeping hot zones away from the cylinder walls. This study investigates how the two-injector concept delivers even higher efficiency by providing additional control of spray -and injection angles. Three-dimensional Reynolds-averaged Navier-Stokes simulations examined several umbrella angles, spray-to-spray angles, and injection orientations by comparing the two-injector cases with a reference one-injector case. The study focused on heat transfer reduction, where the two-injector approach reduces the heat transfer losses by up to 14.3 % compared to the reference case. Finally, this study connected the two-injector approach to a waste-heat recovery system through GT-Power 1-D simulations, increasing the importance of heat transfer reduction. The final two-injector system then delivered a 54.4% brake thermal efficiency compared to 53% of the one-injector reference case.

Introduction

Heavy-duty (HD) diesel vehicles produce a quarter of the EU transportation sector's CO₂ emissions [1], corresponding to 6 % of the total EU emissions [2]. Anticipations conclude a 45 % HD CO₂ emissions increase by 2040 [3]. Thus, the European Commission set an EU-wide CO₂ emissions standard for HD vehicles [4]. This standard targets a 15 % fleet-averaged CO₂ emissions reduction by 2025, and a 30 % reduction by 2030, compared to 2019 EU levels.

Increased engine efficiency reduces the CO₂ levels by reduced fuel consumption. High engine efficiency follows from low energy losses, such as heat transferred through the cylinder walls. The Double Compression Expansion Engine (DCEE) concept aims to reduce all energy losses simultaneously by a split-cycle approach. Every cylinder is split into one High-pressure (HP) unit and one Low-pressure (LP) unit [5]. This approach allows high combined compression ratios of up to 100:1, where the combustion process occurs in the HP-unit at high in-cylinder pressures, around 300 bar, implying manufacturing and combustion constraints. The split-cycle approach further allows work

extraction from usually wasted exhaust energy by performing a second expansion in the LP unit.

One-dimensional simulations concluded a 56 % brake efficiency for the DCEE concept [6]. However, a high in-cylinder pressure renders a high in-cylinder temperature giving potentially high DCEE heat losses. High heat losses further follow in other engine concepts due to the current trend towards higher compression ratios giving higher in-cylinder temperatures [7,8]. To tackle this challenge, studies have suggested that multiple injectors reduce heat losses while keeping emissions at decent levels [9,10].

Senecal et al. [11] conducted computational fluid dynamics (CFD) simulations with different combustion chamber shapes and number of injectors. The study concluded that both two and three injectors per cylinder reduced NO_x emissions. However, the two-injector case enlarged the fuel-rich zones, increasing soot emissions. The three-injector case solved this issue where the angle between sprays mattered for the soot-NO_x trade-off. Uludogan et al. [12] further explored the use of multiple injectors through CFD using six single-hole injectors at the piston bowl's rim to increase the diesel engine power density while maintaining low pollutant emission levels. They concluded that the spray geometry conditions affect the soot and NO_x levels. Those results indicate that the engine emissions depend on the spray geometry. Furthermore, the spray pattern, umbrella angle, spray-to-spray angle, and spray-orientation affect the wall heat losses, the exhaust losses, and the indicated thermal efficiency.

Uchida et al. [9] showed how two additional injectors, placed at the piston bowl's rim, reduce the heat losses by 2.6 % points in an HD engine. Further multiple-injector studies concluded a total efficiency increase of 2.6 % from lower heat losses in the DCEE concept [13]. Here, a single-cylinder HD engine combined two injectors with a flat bowl to reduce the DCEE HP unit's heat losses. The lower flat bowl surface area rendered lower convective heat losses with further heat loss reductions by a lower boundary temperature gradient.

This paper investigates the multiple injectors' configuration regarding the spray orientation, umbrella angle, and spray-to-spray angle, using CFD simulations to optimize the multiple injectors system. In addition, this study explores the complete efficiency analysis of the DCEE concept, using 1-D simulations.

Methodology

CFD Simulation Setup

A renormalized group k-epsilon RANS turbulence model was used to perform HD diesel engine simulations in CONVERGE v3.012 software. The high-pressure unit of the DCEE concept has been modeled as the firing cylinder of a Volvo D13 HD engine.

N-heptane was used as a surrogate for the diesel fuel, but diesel's lower heating value was used in the simulations. The liquid fuel injected and liquid droplets had diesel's chemical and thermo-physical characteristics. Thus, only the evaporated parcels have been considered as n-heptane for chemical reactions. The Kelvin-Helmholtz Rayleigh-Taylor (KH-RT) approach was implemented to capture the fuel injection and spray breakup [14]. The KH and RT breakup time constants used in this study were 7 and 1, respectively. The fuel injection process follows a discrete model with 200,000 spherical parcels using the Frossling droplet evaporation correlation [15] for diameter estimations. The Rosin-Rammler cumulative probability distribution predicted the droplets' size distribution. The reduced kinetic model AramcoMech for compression-ignition, with 61 species and 270 reactions, was used in this study. This model is appropriate for simulations of complex TPRF surrogate fuels [16].

A base grid size of 2 mm was used in all three directions for the mesh, with 471,056 cells at BDC. Fixed embeddings of scale 3 for the injectors were added to account for the near-nozzle flow, which is of interest to this study. The study does not consider the interior nozzle. Also, fixed embeddings of scale 1 were used for the walls (piston, liner, and head). Level 2 adaptive mesh refinement (AMR) was utilized based on velocity and temperature gradients inside the combustion domain. To reduce the computational cost, the law-of-the-wall temperature boundary condition calculated near-wall behavior where the heat transfer was modeled using the Angelberger model [17].

GT-Power Simulation

The software GT-Power, version 2019, simulated 1D cases for the full DCEE concept. The complete DCEE concept's HP-cylinder here consisted of similar geometrical parameters as the Volvo D13 HD engine.

For the full DCEE concept, the simulation procedure followed an earlier model developed by Volvo. A 2-4-2 layout was implemented with a 2-stroke compressor, a 4-stroke combustion cylinder, and a 2-stroke expander, as shown in Figure 1. The cylinder size order follows, from largest to smallest, as: expander, compressor, and combustor. An LP tank after the compression cylinder and an HP tank before the expander performed an isobaric gas exchange between the cylinders following from the tanks' large volumes. For more details on the DCEE 2-4-2 layout, please refer to Lam et al. [18].

Woschni GT heat transfer model calculated the combustion cylinder convective heat transfer losses. For compressor and expander units, the heat transfer multiplier was kept at a low value of 0.1 because of expected lower heat losses than the combustion cylinder. The friction losses were calculated using a linear correlation, as proposed by a previous study [6]. The CFD simulation results provide the pressure trace and rate of heat release (RoHR), which was used as an input to the combustion cylinder of the DCEE concept.

Figure 1. A 2-4-2 layout of the double compression expansion engine (DCEE) concept modeled using GT-Power

CFD Models Validation

The simulations followed the experiments' setup, with a single-cylinder 4-stroke Volvo D13 engine at 1200 rpm. At 1500 bar injection pressure, a standard central-mounted fuel-injector was utilized to inject the diesel fuel directly into the combustion chamber. The engine used an 11.5 compression ratio giving a larger chamber volume at the top dead center (TDC) to mimic the DCEE conditions [5]. Tables 1 and 2 present the engine specifications.

Table 1. Geometric engine specifications

Cylinder bore	0.131 m
Stroke	0.158 m
Con. Rod length	0.255 m
Crank offset	0 m
Compression ratio	11.5 : 1
Fuel system	Common-rail direct-injection

Table 2. Central injector geometry and specifications

Number of nozzle holes	7
Nozzle diameter	255 μm
Spray cone angle	10 deg
Spray umbrella angle	150 deg
Discharge coefficient	0.94

In-cylinder parameters at IVC, such as wall temperatures and trapped gas composition, are important to set up the simulations. Since the initial and boundary conditions are difficult to obtain from the experimental engine, a 1-D GT-Power simulation was accounted only for the combustion cylinder. This simulation was set to fit the experimental in-cylinder pressure trace and RoHR, and the resulted

initial and boundary conditions are presented in Table 3. No swirl is also considered since the swirl level's effect has demonstrated not to be consistent over a load range [19]. Finally, this parametric study was conducted in a typical mid-engine load (FuelMEP in Table 3). The engine's speed was kept constant at 1200 rpm.

Table 3. Initial and boundary conditions from GT-Power used for simulations

In-cylinder pressure at IVC	4.454 bar
In-cylinder temperature at IVC	368 K
Engine speed	1200 rpm
Injection pressure	1500 bar
Piston temperature	550 K
Liner temperature	450 K
Cylinder head temperature	550 K
In-cylinder gas composition at IVC (mass fractions)	
CO ₂	0.0059
H ₂ O	0.0021
N ₂	0.7657
O ₂	0.2263
Injection strategy	
Start and duration of the pilot injection	-5 ATDC (208 μ s or 1.5 CAD)
Start and duration of the main injection	-1 ATDC (1520 μ s or 11 CAD)
Injected fuel mass	192 mg/cycle
Fuel temperature	363 K
FuelMEP	~40bar
IMEP _{gross}	17.1 bar

A single-injector simulation validated the CFD models against experimental data on the engine geometry described in Tables 1 and 2, with initial -and boundary conditions reported in Table 3. All simulations used only the closed-volume part of the cycle, from IVC (-160 CAD ATDC) to EVO (140 CAD ATDC). Thus, the simulations evaluated the combustion chamber without any exhaust or intake systems.

The validation performed the following steps: adjusting the effective compression ratio to match the motored pressure trace; introducing the in-cylinder residual gases to reach the fired case's pressure before fuel injection; adjusting the injection duration and injection timing to account for injector hydraulic delay and the lower heating value for diesel compared to n-heptane, and altering heat transfer models to match the experimental in-cylinder pressure trace.

Figure 2 presents the experimental and computational in-cylinder pressure, RoHR, integrated heat release, in-cylinder temperature, and the computational rate of fuel injection (ROI). The simulated pressure trace, integrated HR, and in-cylinder temperature match the experimental results. Table 4 shows the RoHR predicted CA10, CA50, and CA90, which slightly differs each other. Table 4 also shows that the simulated IMEP_{gross} differs by 1.3% compared to the experimental one, mainly due to the late unwanted fuel addition presented in the experimental in-cylinder pressure trace. The second hump in the experimental RoHR is attributed to a possible injector dribbling observed previously at the end of injection in optical engine experiments [20], which has a minor effect on the validation.

Figure 2. CFD validation of In-cylinder pressure trace, RoHR trace, and In-cylinder temperature

Table 4. Comparison of the combustion phasing and gross indicated mean effective pressure (IMEP_{gross}) obtained experimentally and numerically.

	Experiments	Simulations
CA 10	4.8 deg ATDC	4.91 deg ATDC
CA 50	12 deg ATDC	12.3 deg ATDC
CA 90	20.8 deg ATDC	23.1 deg ATDC
IMEP _{gross}	17.1 bar	17.33 bar

Design of Experiments

This study aimed to compare two-injector and single-injector geometries with two different piston bowl shapes. To utilize the two injectors optimally (and with the assumption that the single-injector case is already optimized), the spray's angles were changed following the convention presented in Figure 3 where: the first term is the spray-to spray angle (β), the second term is the umbrella angle (ψ), and the third term is the orientation angle (θ). The cases are then named as $\beta \times \psi \times \theta$, where the x represents the corresponding angle. The study sweeps the β , ψ , and θ angles, as illustrated in Table 5. One particular case with different ψ angles, namely 70, 80, 80 in Table 5, was simulated for each β and θ combination. In this case, the ψ angle of the closest spray to the liner is set to 70 degrees, shooting the fuel towards the piston bowl instead of the squish zone. Additionally, some cases

direct both side-injectors towards the piston bowl center. These cases are henceforth labeled "Frontwards" directed.

Figure 3. Schematic of side injectors spray angles and side injector location

Table 5. Angular swept parameters from the side injectors

Beta (β) [deg]	Psi (ψ) [deg]	Theta (θ) [deg]
20	80	110
		120
	70	130
		140
	70, 80, 80	150
30	80	110
	70	120
	70, 80, 80	130
		Frontwards
40	80	Frontwards
	70	
	70, 80, 80	

To allow an identical flow rate, the central injector yields six holes compared to the three-holed side-injector. The central-injector case was performed with the standard piston bowl shape (Figure 4a). Since the same injector conditions (injection pressure, ROI, and injected fuel mass) were used for both the 7-hole and 6-hole injectors, the study assumed small combustion differences. The hole diameter was increased from 255 μm (7-hole) to 270 μm (6-hole), to account for hole number reduction and discharge coefficient detriment. It was deemed necessary to utilize an even number of holes for this study to perform a fair comparison between two-injector and single-injector cases.

Several side spray angles were simulated using a flat bowl piston (Figure 4b). Previous works showed that a flat bowl improved the

multiple injector concept's efficiency [13,21]. Since no differences for spray pattern, injection, wall temperature, and initial conditions were induced between the two piston bowls, this study assumed small combustion differences. Thus, no further validation procedure was performed with the flat piston bowl. In a similar way, the study "extrapolates" other parameter changes, such as spray angles. The simulations aim to find an optimum combination of side spray angles in GIE and left-over exhaust energy. The latter could be used for a second expansion in the expander unit of the DCEE concept.

Figure 4. Injector and spray configuration. a. Standard case (Standard bowl piston). b. Side injector case (Flat bowl piston)

All simulations utilized the injection strategy and conditions in Table 3, where IMEP_{gross} changes with efficiency.

A merit function (MF) was defined to find the most suitable cases for high GIE and exhaust energy (to be utilized for a second expansion in the DCEE concept). Mathematically, the MF follows as:

$$MF = GIE + \eta_{Exp} * \left(\frac{ExhMEP}{FuelMEP} \right)$$

The first term of the right-hand side is the gross indicated efficiency produced in the combustor unit. The second term is the combination of the expander efficiency (η_{Exp}) and the left-over exhaust energy normalized by the fuel energy. Here, the DCEE recovers exhaust energy at a 50 % efficiency based on previous evaluations at Volvo. It should be noted that the 50 % exhaust recovery is not valid for all load conditions, which is why this study includes a full DCEE system analysis.

Finally, only the ten cases with the highest MF values are analyzed. The GT-Power simulations of the whole DCEE concept are performed on the top and lower MF cases, as well as the standard CDC, to assess the influence of the spray pattern on the efficiency and energy distribution of the concept.

Results and Discussion

This section presents and explains the results of the parametric simulation study. The study evaluated several cases, but here only the top ten cases are presented in terms of MF. Furthermore, this section describes the multiple injector performance in terms of MF, in-cylinder pressure, RoHR, energy distribution, and heat transfer losses. Finally, this section presents the energy distribution analysis of the complete DCEE concept.

Merit Function

This function evaluates the optimal spray pattern for the DCEE concept. Figure 5 presents the MF values and the contribution from each equation part (GIE and recovered exhaust energy). Three of 33 multiple-injector cases increase the MF compared to the standard CDC, showing the importance of a correct spray pattern. In general, the two-injector cases experience a decay in GIE (compared to the standard CDC). However, the increased exhaust energy, rendering work in the DCEE waste-heat recovery (WHR) system, improves the two-injector MF values. This is in line with previous studies [9,10,13]. Thus, the MF value remains sensitive to the WHR efficiency, where a higher efficiency favors the two-injector cases.

Figure 5. Merit Function distribution for best cases

Figure 5 shows that almost all the two-injector cases render higher exhaust energy than the standard CDC case. The maximum MF, at 0.6551, follows from the angles: $\beta=20^\circ$, $\psi=80^\circ$, and $\Theta=150^\circ$ (see Figure 6, left). This MF value corresponds to a 1.8 % increase compared to the standard CDC. As a comparison, the lowest MF (of the ten cases here) follows from $\beta=30^\circ$, $\psi_{\text{closest to liner}}=70^\circ$, $\psi=80^\circ$, and $\Theta=120^\circ$ (see Figure 6, right).

The comparison in Figure 6 shows that for the lowest MF case, the fuel hits the piston bowl earlier during the cycle, leading to a higher amount of fuel going into the squish zone close to the liner. This larger combustion zone into the squish zone produces higher heat transfer

(HT) losses by the liner and cylinder-head than the maximum MF case. By keeping the hot zones into the combustion chamber center, the best case reduces these losses and piston HT losses, allowing both higher GIE and exhaust energy.

To summarize, the results of this CFD study showed that an increase in the total efficiency of 1.8% could be achieved using a DCEE expander unit efficiency of 50% ($\eta_{Exp}=0.5$).

Figure 6. Maximum (left) and lowest (right) Merit Function cases of the top ten cases

Performance

Figure 7 illustrates the in-cylinder pressure trace, RoHR, and mean in-cylinder temperature during the cycle for the top-MF five (left) and bottom-MF five (right) cases presented. Several two-injector cases experience a faster initial RoHR (compared to the standard CDC), with a following maximum peak-pressure increase at around 5 bar. Nevertheless, the pressure during expansion for the multiple injector cases is lower than the standard CDC case, leading to the detriment in IMEP_{gross}. The top-MF case further shows an earlier RoHR peak than the standard CDC, which favors the GIE by a longer remaining stroke. However, the cases with lower MF than the reference case presented lower RoHR and longer late combustion, increasing the HT losses, which will be discussed later.

The mean in-cylinder temperature shows values either close or lower to the standard CDC case near to TDC; as a result of this, the HT losses are reduced for the multiple injector concept, as it will be discussed in the following section.

Figure 7. In-cylinder pressure, RoHR, and In-cylinder temperature for the top-MF five (left) and bottom-MF five (right) cases

Energy Distribution and Heat Transfer Losses

The multiple injection concept was introduced first to increase the engine's energy density [11,12]. However, in the last decade, this concept has been implemented to reduce HT losses [9,22] and increase the exhaust energy [10,13,21]. Figure 8 presents the energy balance for the ten cases tested here, where combustion losses are omitted since the combustion efficiency was always higher than 99.76 %. All the cases with higher MF than the standard CDC case have lower heat transfer losses and higher exhaust energy, obtaining 2.03 %-points (14.3%) lower HT losses and 1.81%-points (4.25%) higher exhaust energy for the top-MF case ($\beta 20 \psi 80 \ 0150$) compared with the standard CDC. Although the indicated work is only higher for the first case, the combined effect of the lower HT losses and higher exhaust energy makes the multiple injector concept potentially more suitable for a WHR system such as the DCEE concept.

The WHR efficiency importance becomes evident in Figure 8, where some two-injector cases with a high MF have low GI work levels combined with exhaust energy. Thus, a higher WHR efficiency would increase the MF, compared to the standard CDC, for these cases. With a lower WHR efficiency, the standard CDC MF rises compared to the two-injector cases due to its higher GI work level, with only one two-injector case delivering higher GI work at this load condition.

Connecting to the previous discussion in Figure 7, the low-MF cases presented late combustion, producing higher HT losses. This is also shown in Figure 8, where the HT losses increase combined with a decreased GI work level for these cases. The early two-injector peak RoHR for the top-MF cases discussed earlier does not deliver higher

GI work, as seen in Figure 8, with only the top-MF case producing higher GI work than the standard CDC. The changed piston geometry could explain this but needs to be investigated further.

Figure 8. Energy distribution of the high-pressure unit

Figure 9. Heat transfer losses for the MF ten top cases

Figure 9 shows the heat losses from different boundaries where the piston surface is responsible for more than 50% of these losses. Thus, although the two-injector cases experience similar liner -and head HT losses (except for the top-MF (the lowest) and lower-MF (the highest) cases), the total HT decreases as an effect of the reduced piston HT. This is a direct effect from the side-injections aiming towards the combustion chamber center instead of towards the piston surface, which will be discussed later. Figure 9 again shows the importance of a correct spray pattern where some two-injector cases increase the heat losses, although the injector-wall distance is longer.

To explain the lower two-injector HT losses, Figure 10 presents the piston's near-wall gas temperature distribution for the four cases with maximum MF at different CAD ATDC. The standard CDC experiences significantly larger areas covered by high-temperature gases for each CAD than the two-injector cases. Furthermore, the two-injector cases experience a smaller fuel-impinged area, which explains the lower piston HT losses for these cases.

Although swirl motion was not imposed (swirl ratio = 0), the tangential injection from the bowl's rim produced a slightly swirling motion that enhanced the air utilization and dragged the hot combustion products from the piston's surface. The lower residence time of the high-temperature gases in contact with the cold piston surface might further explain the lower HT losses produced for the multiple injector concepts compared with the baseline case.

Comparing the two-injector cases, Figure 10 shows an earlier interaction between the fuel and the piston's wall for those cases with a lower umbrella angle for the closest spray to the liner ($\psi_{70} 80 80$). The fuel is injected directly towards the piston wall producing an upward motion, allowing fuel and hot combustion products to go into the squish zone. This explains the results presented in Figure 9, where the maximum MF case, with a large umbrella angle, showed lower liner and head HT losses.

Figure 10. Piston's near-wall gas temperature distribution for the four best cases

It can be concluded that the best-case spray angles combination keeps most of the combustion process and hot gases centered into the combustion chamber as well as far from the liner and the head. This is observed in Figure 11, where both the liner and head boundaries have earlier HT losses for the second and third cases ($\beta_{30} \psi_{70} 80 80 0130$ and $\beta_{20} \psi_{70} 80 80 0150$) compared with the maximum MF case. Results in Figure 11 also supports the hypothesis that the lower interaction with the piston's wall reduces the liner and head HT losses. It can be seen that for those cases with strong interaction between fuel and piston's wall ($\beta_{30} \psi_{70} 80 80 0130$ and standard CDC), the liner HT losses start earlier and faster than the best case, resulting in higher losses at the end of the expansion stroke.

Figure 11. Liner (solid lines) and head (dashed lines) accumulated heat transfer losses for the first four best cases

Energy Distribution Analysis of the Complete DCEE Concept

Figure 12 shows the energy distribution of brake efficiency, friction losses, exhaust losses, and heat transfer losses for the first-top case ($\beta 20 \psi 80 \theta 150$), the fourth-top case (Standard CDC), and for the tenth-top case ($\beta 30 \psi 70_80_80 \theta 150$). Here the plot for energy distribution is calculated for the complete DCEE concept, i.e., compression, combustion, and expansion cylinders. As mentioned in the methodology section, the heat transfer losses and friction losses are estimated using the Woschni equation and a linear correlation, respectively. The exhaust losses, however, are calculated from balancing the total fuel energy input.

Figure 12. The energy flow distribution of the complete DCEE concept for $\beta 20 \psi 80 \theta 150$, CDC and $\beta 30 \psi 70 \ 80 \ 80 \theta 150$ cases

It can be seen that the $\beta 20 \psi 80 \theta 150$ case had the highest BTE, followed by the CDC and $\beta 30 \psi 70 \ 80 \ 80 \theta 150$ cases, consistent with Figure 8. Another advantage of using two side injectors with the selected configuration of $\beta 20 \psi 80 \theta 150$ is the reduced heat transfer losses, which is found to be 1.8 %-points (12.86%) and 2.9 %-points

(19.2%) lower than CDC and $\beta 30 \psi 70 \ 80 \ 80 \theta 150$ cases, respectively. It is to be reminded that a different convection heat transfer multiplier for the combustion cylinder was used for these cases to match the in-cylinder pressure and rate of heat release. The friction losses for the $\beta 20 \psi 80 \theta 150$ case were 0.2 %-points higher than the other two cases due to higher friction mean effective pressure and peak cylinder pressure. The exhaust losses for $\beta 30 \psi 70 \ 80 \ 80 \theta 150$ were found to be minimal as the exhaust temperature was the lowest for this case, resulting in the lowest exhaust enthalpy.

Figure 12 was further dissected to understand how the energy flow was distributed among the three cylinders of the 2-4-2 layout of the DCEE concept using the top MF case, i.e., $\beta 20 \psi 80 \theta 150$ case and is shown in Figure 13. The net indicated efficiency, friction losses for each cylinder were calculated using the combustion cylinder given input fuel energy. The compressor's 1st stroke represents the ambient air induction from TDC to BDC and the re-expansion process after the isobaric gas transfer to equalize the in-cylinder pressure to ambient conditions. The 2nd stroke indicates the ambient air's compression process and the transfer of gases to the LP tank. The net indicated efficiency from the compressor is found to be -11.5 %.

The combustion cylinder works on a conventional 4-stroke cycle. The 3rd stroke represents the suction of compressed air from the LP tank. The compressed air undergoes a second compression process during the 4th stroke, which follows the first expansion process during the 5th stroke and the transfer of exhaust gases to the HP tank in the 6th stroke. The net indicated efficiency from the combustor lead to 35.4 %.

The expander cylinder, similar to the compressor, works on a two-stroke cycle. The 7th stroke denotes gas induction from the HP tank to the expander unit and the second expansion process until the gas reached the ambient pressure. The 8th stroke denotes the expulsion of fully expanded gas to the exhaust manifold, followed by a re-compression process to match the HP tank's in-cylinder pressure. The expander cylinder showed a net indicated efficiency of 33.4 %.

It was found that ~21 % of the combustor unit's exhaust energy resulted in 33.4 % of useful work from the expander unit. The combined net indicated efficiency from the compressor and expander units is 21.9 %, which, when summed to the combustion cylinder efficiency of 35.4 %, resulted in an overall net indicated 57.3 % efficiency. Due to lower friction losses in the compressor and expander units, high mechanical efficiency of ~99 % was achieved.

Figure 13. An example of the breakdown of the energy distribution for compression, combustion, and expansion cylinders of the DCEE concept for $\beta 20 \psi 80 \theta 150$ case

Conclusions

This study performed three-dimensional RANS CFD simulations at mid-load conditions in a heavy-duty Volvo D13 modified single-cylinder engine. A parametric study evaluated different injection -and spray angles to optimize two-injector concept's performance, especially when utilizing waste-heat recovery systems. Thus, the merit function consisted of both the gross indicated efficiency and exhaust energy (to be used in the waste-heat recovery system). Finally, GT-Power simulations assessed the influence of the two-injector configuration on the Double Compression Expansion Engine concept's energy distribution. The most important findings follow as:

1. Two-injector configurations delivered higher efficiency than the reference one-injector case when utilizing a waste-heat recovery system due to lower heat losses and faster combustion.
2. The side-injector cases reduced the heat transfer losses by up to 14.3 % compared to the one-injector reference case, mainly due to decreased piston convection attributed to lower gas temperatures close to the boundaries.
3. The two-injector cases produced a gentle swirl motion enhancing the air utilization and keeping the hot gasses away from the piston boundary, leading to reduced heat losses.
4. The top merit function case produces at BTE of 54.4 % when connected to the DCEE concept, which is 1.4 %-points higher than the standard CDC case. Again, this is mainly due to the overall lower heat transfer losses and higher exhaust energy obtained by the two-injector, with waste heat recovery from the Double Compression Expansion Engine concept.

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Abbreviations

CA10	Crank angle at 10% of cumulative heat release
CA50	Crank angle at 50% of cumulative heat release
CA90	Crank angle at 90% of cumulative heat release
CAD ATDC	Crank angle degree after top-dead-center
DCEE	Double Compression Expansion Engine
HD	Heavy-Duty
HP	High-pressure
LP	Low-pressure
IVC	Inlet Valve Closing
EVO	Exhaust Valve Opening
RoHR	Rate of Heat Release
ROI	Rate of fuel injection
IMEP_{gross}	Indicated Mean Effective Pressure gross
FuelMEP	Fuel Mean Effective Pressure
β	spray-to spray angle
ψ	umbrella angle
θ	orientation angle
MF	Merit Function
WHR	Waste-heat recovery
CDC	Conventional Diesel Combustion
HT	Heat Transfer
GIE	Gross Indicated Efficiency
GI	Gross Indicated
η_{Exp}	Expander efficiency