

Experimental Analysis of Engine Exhaust Waste Energy Recovery Using Power Turbine Technology for Light Duty Application

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ABSTRACT

An experimental analysis was executed on a NA (Natural Aspirated) 4-stroke gasoline engine to investigate the potential of exhaust waste energy recovery using power turbine technology for light duty application. Restrictions with decreasing diameter were mounted in the exhaust to simulate different vane positions of a VGT (Variable Geometry Turbine) and in-cylinder pressure measurements were performed to evaluate the effect of increased exhaust back pressure on intake- and exhaust pumping losses and on engine performance. Test points in the engine map were chosen on the basis of high residence time for the vehicle during the NEDC (New European Driving Cycle). The theoretically retrievable power was calculated in case a turbine is mounted instead of a restriction and the net balance was obtained between pumping power losses and recovered energy.

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INTRODUCTION

Many solutions are related to increasing the efficiency of the internal combustion engine (ICE) itself. However, the technical limit for improvement of current engine designs is nearing and will not be sufficient to meet future fuel economy targets without additional measures. Other technologies are focused on transmission design or the recuperation of kinetic vehicle energy such as several electric hybrid drive-trains.

Potentials for efficiency maximization of a running and brake loaded engine is determined by all individual (in)efficiencies according to

$$\eta_b = \eta_{comb} \cdot \eta_{therm} \cdot \eta_{GE} \cdot \eta_{mech} \quad (1)$$

In this Equation η_{comb} denotes the combustion efficiency, which is the fraction of the fuel energy supplied which is released in the combustion process. For equivalence ratios lower than one, the combustion efficiency is between 90 and 100 %. η_{therm} denotes the thermodynamic efficiency, which

relates the actual work per cycle to the amount of fuel chemical energy released in the combustion process. Typical values for the thermodynamic efficiency are nowadays around 30 %. Furthermore, η_{GE} denotes the gas exchange efficiency and indicates how efficient the gas moves through the different engine process cycle stages. Typical values for the gas exchange efficiency range from 50 up to 97 %, depending on load. Finally, η_{mech} is the mechanical efficiency, which is the ratio of the brake (or useful) power delivered by the engine to the net indicated power. Typical values vary from 0 to 90 % depending on engine speed and other parameters. By definition the mechanical efficiency is zero for 0 bar BMEP.

Further big improvements can only be obtained when the largest source of losses is addressed, namely exhaust waste energy. This loss source can be ascribed to the poor thermodynamic efficiency. As a consequence, about 40 % of

the ingoing fuel energy of today's ICE's is lost via the exhaust gas [1].

To recover some of these losses, different technologies are under investigation in which the Rankine cycle and power turbine technology cover the largest part. According to Ringler et al. [2], the recoverable amount of power in case of power turbine technology is driven by pressure gradients and/or kinetic energy while in case of the Rankine cycle a temperature difference or exhaust heat is addressed. For this reason the Rankine cycle has a higher potential in terms of heat utilization. However looking at complexity and costs, power turbine technology can be a more attractive solution. Beside that, the use of power turbine technology does not exclude the use of other thermal processes.

While axial turbines are mainly used in heavy duty and stationary applications because of their high efficiency, radial turbines are mainly used for light duty applications. In contrast to the axial flow turbine, for which the efficiency decreases with size due to relatively large blade clearance effects, the radial flow turbine has the great advantage that it maintains a relatively high efficiency when reduced to small sizes. It can efficiently handle a high expansion ratio and is also more robust and cheaper to manufacture than an axial turbine. Furthermore it is able to accept quite unfavorable unsteady inlet conditions like highly pulsating flows, found in the exhaust of a ICE (Internal Combustion Engine), without a complete deterioration of its performance [3].

In literature, several reports are presented on performance improvements or generated power with power turbine technology driven by exhaust energy [4, 5, 6]. However, in most of them the effect of increased exhaust back pressure on engine performance is treated very briefly, only simulated, or not mentioned at all.

During earlier performed experiments [7], no significant net improvement in specific fuel consumption was observed for overall system (engine with power turbine) performance which is in contrast with results from Michon et al. [4]. The presumed cause was that the increased level of exhaust back pressure, compared to the natural aspirated case, increases the engine pumping losses. In a second experimental session performed by the authors, the effect of exhaust back pressure on pumping losses and engine performance was analyzed in more detail. This is very important for further development of power turbine technology for exhaust waste heat recovery purposes, but also for any other technology that causes an increase in exhaust back pressure. The results from this analysis are presented in this paper.

METHODOLOGY

The research goal is to quantitatively and qualitatively determine the effect of exhaust back pressure on engine pumping losses and overall performance and to get insight into the theoretically maximum recoverable amount of power using power turbine technology.

Four engine operating points (see Table 1) were chosen on the basis of highest vehicle residence time according to the NEDC and for a significant amount of exhaust mass flow.

Table 1. Engine Operating Points

Engine Speed [rpm]	Engine Torque [Nm]	net IMEP [bar]
2300	20	1.9
2500	30	2.4
2750	50	3.4
2500	75	4.7

Restriction Hole Diameters [mm]							
45	35	25	20	15	12.5	11.25	10

The engine residence map is depicted in Figure 1. It is obtained by simulation of the NEDC with the specifications of a Nissan SR20DE engine and a RS5F70A manual transmission. The crosses denote the residence of the engine. Higher vehicle residence is indicated by increasing square color strength and the chosen engine operating points are highlighted in green. Furthermore, relevant vehicle speeds are indicated in the map.

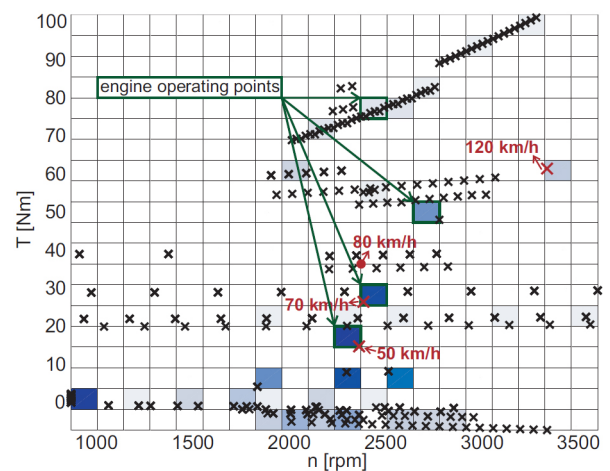


Figure 1. Chosen engine operating points highlighted in green in the engine residence map based on the residence time according to the NEDC

To increase the exhaust back pressure eight orifice plate restrictions with decreasing hole diameter (see Table 1) were mounted in the exhaust, where 45 mm is the exhaust diameter without restriction, and the in-cylinder pressure trace and other relevant parameters were acquired. This was done for all four engine operating points. Instead of a back pressure valve, a restriction was chosen for practical reasons and because merely stationary points were analyzed.

EXPERIMENTAL SET-UP

For the experiments a Nissan 2 L four cylinder NA 4-stroke gasoline engine was used. The properties of the engine are listed in Table 2.

Table 2. Engine Properties

Bore [mm]	86
Stroke [mm]	86
Compression ratio	10
Displacement [cm ³]	1998
Valve number	16
Injection system	PFI (Port Fuel Injection)

The in-cylinder pressure traces were acquired with a 0.3 mm Kistler piezo quartz pressure transducer integrated in the spark plug which was mounted in one cylinder. Both crank-angle resolved and time averaged data were acquired during the measurements. The exhaust back pressure was measured in the exhaust manifold, both crank-angle resolved and time averaged. Finally, the EGR (Exhaust Gas Recirculation) system of the engine was closed.

ANALYSIS

Crank-Angle Resolved Data

From the in-cylinder pressure trace, which was matched with intake pressure just before IVC (Intake Valve Closing), the aRoHR (apparent Rate of Heat Release) was calculated with the assumption of an ideal gas. Furthermore, net IMEP (Indicated Mean Effective Pressure) and PMEP (Pumping Mean Effective Pressure) were calculated. The pumping losses were individually calculated for the intake and exhaust stroke.

There are different conventions for the calculation of the pumping work. In this paper the 1:360° integration convention is used, described by Shelby et al. [8], which includes area B plus area C in the total pumping losses (Figure 2).

This convention implicitly assumes that the work associated with exchanging the exhaust gas with fresh charge occurs only during the exhaust and intake strokes with no impact on the indicated work. Shelby et al. proposed a new method to account for the inaccuracy introduced when valve timings deviate from conventional timings (e.g. late intake valve closing), however for the engine described in this paper this is not the case.

For NA engines the pumping work transfer is from the piston to the cylinder gases if the pressure during the intake stroke is less than the pressure during the exhaust stroke. For highly loaded turbo-charged engines, the exhaust stroke pressure is usually lower than the intake pressure and therefore the pumping work transfer is from the cylinder gases to the piston.

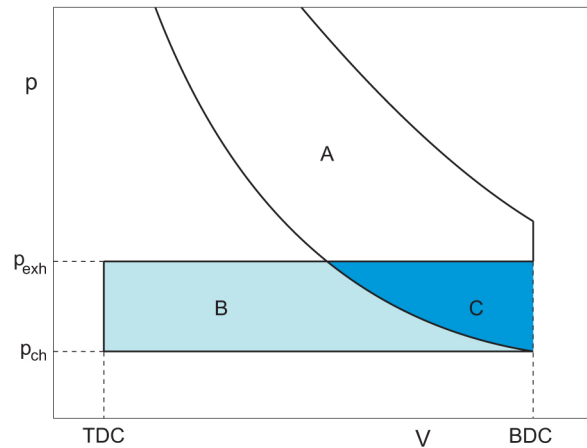


Figure 2. Pumping work in volume pressure diagram

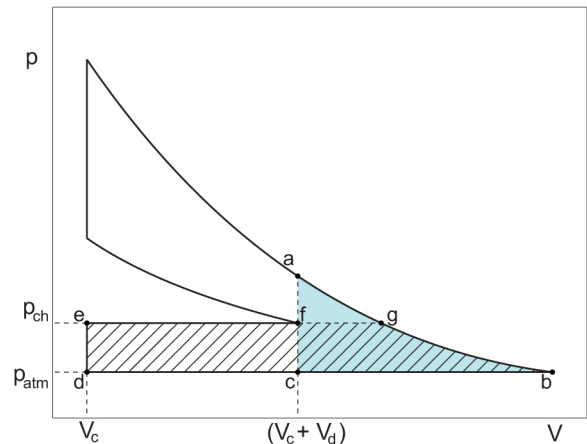


Figure 3. Available energy turbine

Considering Figure 3, ideally, according to [3] the energy available to the turbine is represented by area abc , which represents the engine blow-down pulse or work transfer produced by expanding the gas in the cylinder at exhaust valve opening to atmospheric pressure, and area $cdef$, which represents the work done by the piston displacing the gases remaining in the cylinder after blow-down. However, a positive pressure difference over the engine is normally difficult to have at low loads if a turbo-charger is not designed and optimized for this point. With the Miller and Atkinson cycle, an attempt is made to recover losses that are caused by the incomplete expansion (from point a to b). However, this comes at the cost of engine power density. With the constant pressure principle a large exhaust manifold volume connected to a turbine is used and the work represented by area $bdeg$ is available to the turbine. With the pulse pressure principle on the other hand, where a small and short exhaust manifold is used, an attempt is made to utilize the work represented by both areas. This results in a higher amount of energy available to the turbine for the pulse pressure principle.

Time Averaged Data

The available amount of power in the exhaust was calculated from the time averaged data using

$$P_{av} = \dot{m} C_p T_{exh} \left[\left(\frac{p_{atm}}{p_{exh}} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \quad (2)$$

[3] which is valid for reversible isentropic expansion of an ideal gas to atmospheric pressure. In the analysis the kappa was calculated as function of temperature.

The ISFC (Indicated Specific Fuel Consumption) was calculated from both crank-angle resolved and time averaged data.

RESULTS & DISCUSSION

Pulse or Constant Pressure

In Figure 4 the volume pressure diagrams for two cycles with varying restriction hole diameter are shown with a net IMEP of 4.7 bar. For the cycle with a mean exhaust back pressure of 1.1 bar a clear blow-down pulse is visible. For the cycle with a mean exhaust back pressure of 4.1 bar however, no blow-down pulse is visible, which suggests that conditions in the exhaust reflect those of the constant pressure principle.

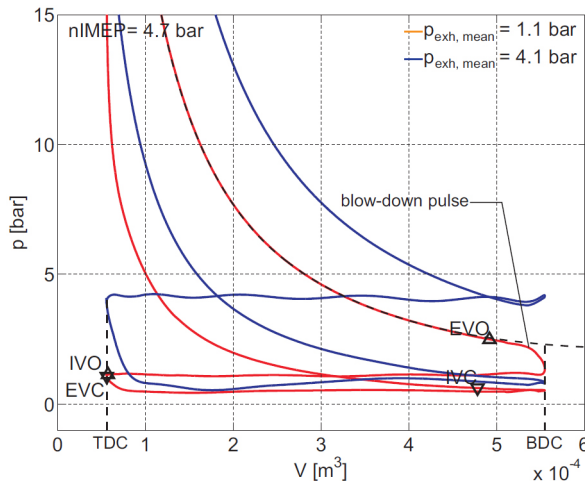


Figure 4. Engine blow-down pulse

For the experiments done it is not exactly clear whether exhaust conditions were according to the constant pressure or the pulse pressure principle. The first part of the exhaust was relatively small and narrow and would suggest that it was pulse pressure. While the pressure sensor was mounted just before the catalyst, with a much larger volume, in the first part of the exhaust, pressure waves were recorded. However, the pressure sensor was mounted at the end of a connecting tube instead of directly into the exhaust manifold. For this reason the tube probably functioned as a Helmholtz resonator and the amplitudes from the recorded pressure waves were weakened more than one order of magnitude compared to

realistic values. Beside that, interference and reflections with unknown origin are possibly superimposed on the signal which further impeded a detailed pulse pressure analysis. Therefore, blow-down pulses will be ignored in the remaining part of the paper. This makes the estimates from the impact of the total system on fuel economy somewhat conservative. Further research has to be conducted on this topic in future work.

Exhaust Back Pressure

When the restriction open area decreases, the exhaust back pressure increases as displayed in Figure 5.

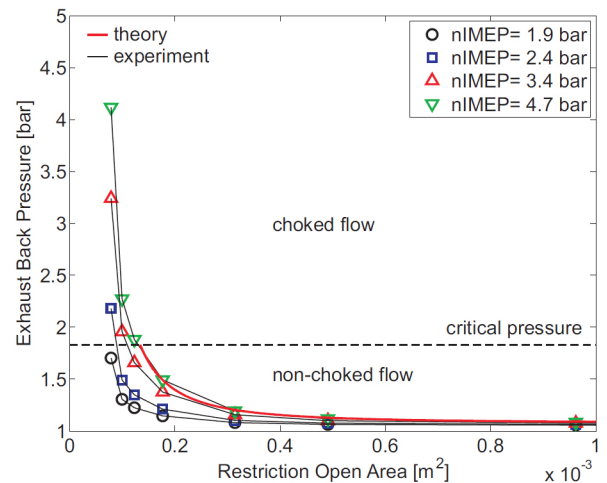


Figure 5. Exhaust back pressure as function of restriction open area

The exponential increase in exhaust back pressure corresponds to the theory of a compressible flow through a flow restriction. The maximum mass flow occurs when the velocity at the minimum area, or throat, equals the velocity of sound. This is true when

$$p_{exh}^* = p_{atm} \cdot \left(\frac{2}{\kappa + 1} \right)^{\kappa/(\kappa-1)} \quad (3)$$

[1] holds and the pressure for this to occur, p_{exh}^* , is called the critical exhaust back pressure. In this case the exhaust gas flow becomes choked and the mass flow is not able to increase any further. As a result, the maximum power of the engine decreases, with decreasing restriction hole area, because higher load implies higher mass flow.

Pumping Work and Recoverable Power

When the PMEP (Pumping Mean Effective Pressure) is evaluated for the different load cases (Figure 6) a clear linear trend is observed. When multiple regression is applied to the data, a linear fit with a r-squared of at least 99 % for all the load cases is found.