

**WAGNER MATOS SANTOS**

**Heat Rejection Avoidance in Combustion Engines**

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**Heat rejection avoidance in combustion engines**

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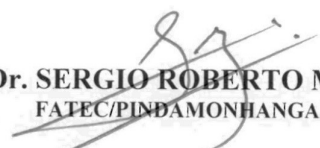
  
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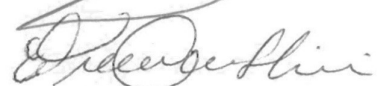
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## **ABSTRACT**

Recent studies show a tendency of waste heat recovering from engines exhaust gases. Through the analysis of those cases, it was possible to identify the high complexity to adapt a Rankine cycle to the known vehicle systems, in combination with combustion engines. Since a high amount of energy is lost from the cooling system, the present study proposes a better use, and analyzes the feasibility of recovering part of it. It also seeks to present a solution for steam generation using part of the energy lost through the combustion chamber walls, as an alternative solution for the conventional exhaust gases heat exchanger. A heat exchanger was developed to substitute the conventional combustion chamber walls, in order to heat a working fluid and activate an expander of a Rankine cycle. A simulation was done using the software GT-Power and the result shows that the purposed systems is able to remove the rejected heat from the combustion chamber as well as the conventional cooling systems. Additionally, the present study shows an Exhaust Gases Recirculation system able to work without a heat exchanger. The gases are admitted from a low pressure line into the intake high pressure line using a Laval Nozzles, that is simulated using an Excel base VBA routine. The described actions are the combination to demonstrate that it is possible to recover energy from the engine cooling system. The simulations show that it is feasible to use an engine gas recirculation system without a cooler, and to recover energy from the cooling system. The combination of those actions leads, in a conservative view, to more than 7% reduction in fuel consumption.

**KEYWORDS:** ORC. Liners. Super Heater. Supersonic. EGR. Heat Recovery. Cooling.

## RESUMO

Estudos recentes mostram tendência de desenvolvimento de sistemas capazes de recuperar energia dos gases de escape. Apesar de ser de fato uma fonte de energia, os sistemas para recuperação instalados na linha de exaustão mostram grande complexidade e necessidade de muito espaço de instalação. O presente estudo demonstra a viabilidade de recuperação de energia do sistema de arrefecimento com espaço de instalação reduzido. O estudo propõe a geração de vapor através de perfis instalados nas paredes da câmara de combustão e sua aplicação em um ciclo Rankine para gerar trabalho através de expansor. A simulação realizada com o software GT-Power demonstra a possibilidade de extrair calor das câmaras com a mesma eficiência de um sistema convencional de arrefecimento por água e aditivo. Paralelamente, o estudo propõe um sistema de recirculação de gases de exaustão sem o uso de um trocador de calor, patenteado pela empresa patrocinadora, complementando a estratégia para evitar o uso desnecessário de energia em um motor a combustão. Os gases provenientes da linha de baixa pressão da exaustão são admitidos pela linha de ar limpo, em alta pressão, através do uso de uma válvula Laval. Os resultados são provenientes de uma rotina de cálculo em Excel, base VBA. O sistema permite que os gases fluam por fora da turbina e elimina a necessidade de um trocador de calor, resolvendo o maior problema dos sistemas convencionais de recirculação de gases em baixa pressão. A combinação das duas soluções, baseado em estimativas conservadoras, pode levar a redução de consumo de combustível de acima de 7%.

**PALAVRAS CHAVE:** Rankine Orgânico. Camisa de pistão. Expansor. Supersônico. Recuperação de energia. Arrefecimento.

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## **LIST OF ABVBREVIATION**

|        |                                 |
|--------|---------------------------------|
| BDC    | Bottom Dead Center              |
| CAC    | Charge Air Cooler               |
| EGR    | Engine Gas Recirculation        |
| ELR    | Engine Load Response            |
| ESC    | European Stationary Cycle       |
| ETC    | European Transient Cycle        |
| HP-EGR | High Pressure EGR               |
| ICE    | Internal Combustion Engine      |
| LP-EGR | Low Pressure EGR                |
| OEM    | Original Equipment Manufacturer |
| PC     | Passenger Cars                  |
| CV     | Commercial Vehicles             |
| SCR    | Selective Catalytic Reduction   |
| SI     | Spark Ignition Engine           |
| TCO    | Total Cost of Ownership         |
| TDC    | Top dead center                 |
| WHR    | Waste Heat Recovery             |

|              |                                                                 |           |
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# 1 INTRODUCTION

## 1.1 SCOPE DEFINITION

The definition of the scope for this project ran around many researches until it was possible to define the boundaries of the PhD Degree at issue.

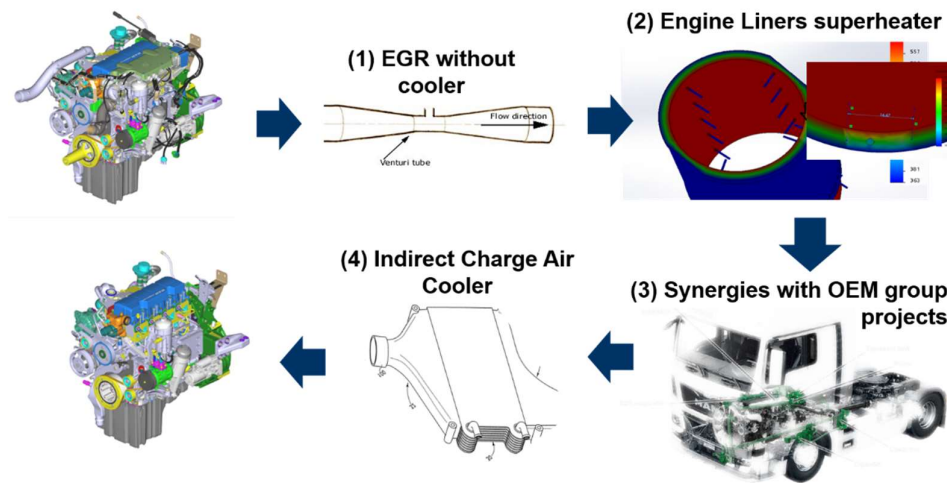
The first idea was set to recover energy from the Heat exchanger used on an Engine with Gas Recirculation system (EGR). See section 2.2.2 for details. It consisted of the development of a super heater able to last until the end of life of the Engine. The suggestion was to develop a heater using the same available space from the current EGR cooler. The thermal fatigue principles would be used to develop an optimized heat exchanger with alternative materials. The Materials and Technology Department of the UNESP university (Universidade Estadual Paulista) was expected to contribute to this investigation.

The bibliographic researches revealed high complexity systems with huge heat exchangers, and a lot of thermodynamics theory. It leaded to a deep analysis of the heat transfer behavior on the existing cooler, the Energy Department was invited to take part of the research. In the meantime, a new insight was postulated: A super heater around the liners could be used instead of the EGR cooler, which meant a smaller and cheaper super heater. The second idea was created in 2016.

The analysis followed towards energy recovery, demonstrating that it would be academically feasible. However, an Original Equipment Manufacturer (OEM) environment would not support such a technological solution without a feasible business case, and with an extreme long lead time for development and implementation.

At this point, synergies where found inside the OEM group. A demonstration truck has been used to support researches from and new ideas were welcome to improve the prototype efficiency. The plan to integrate the proposals on a demo truck would be firstly implementing the EGR solution, secondly to integrate a superheated around engine liners to recovery energy from the cooling system. The system could be integrated on the Demo truck as the third step, and finally, as a fourth step, the use of an indirect charge air cooler could integrate all systems: original WHR (Exhaust), EGR without cooler and super heater around liners (see Figure 1)

Figure 1: Intermediate steps for an OEM waste heat recovery system



Source: My own authorship

Besides the technical discussion this kind of project needs also to be economically feasible for market acceptance. If the customer accepts to pay additional costs for it, the actions are self-driven, but if not, the system shall be able to reduce fuel consumption without adding variable cost to the vehicle and needs also the cost increase for the introduction of new features.

## 1.2 MOTIVATION

This project was motivated by a new trend in the automotive industry: Vehicles with lower CO<sub>2</sub>, NO<sub>x</sub>, particulate and other emissions, and the reduction of Total Cost of Ownership (TCO) by reducing fuel consumption. This term will be further explained in the following chapters.

A project must be profitable for its approval within a company. Considering exclusively the product, the following would be the possible ways:

- Cheaper product to increase margin;
- Same product costs, but increased sales volume;
- Extra costs that the customer wants to pay for;

With the developed market for vocational vehicles and equipment, customers are also evaluating a parameter called Total Cost of Ownership (TCO), which is offered by the OEM based on technical and empirical data from the product. This criterion involves preventive

maintenance, fluids and fuel consumption, and market value for selling the equipment after using it for a specific period.

It is expected that the whole system will be able to reduce at least 7% of fuel consumption, making the TCO worth it for the customer, even if the product itself costs more.

Because a thermal machine is a low efficiency equipment, there are many actions to be taken before electrification. One of them is the use of wasted heat. Even though WHR systems are widely explored, a mass production proposal is not yet in place, mostly due to costs and packaging available on the vehicle. Systems are usually big, heavy, and the added costs to the product don't make it worthy.

As most researches about WHR are geared to the exhaust system (see section 2.3.1), a challenge was set to demonstrate the feasibility of recovering energy from the engine cooling system, which is nowadays poorly explored by the scientific community.

In addition, making a thermal machine more efficient doesn't mean just recovering the lost energy. It also means avoiding heat to be lost. For the actual implementation step, it will be the first action to be taken.

Carstensen et al., 2019 (item 2.5.3) has an important contribution to this work motivation. The MAN system turns the EGR cooler into an evaporator that works in parallel with an evaporator downstream the exhaust system. By removing the EGR cooler, the vehicle as a closed system would have more energy available and by introducing the superheater around the liners, an additional steam source would be available and would overcome the old EGR cooler energy offer to the WHR system.

### 1.3 OBJECTIVE

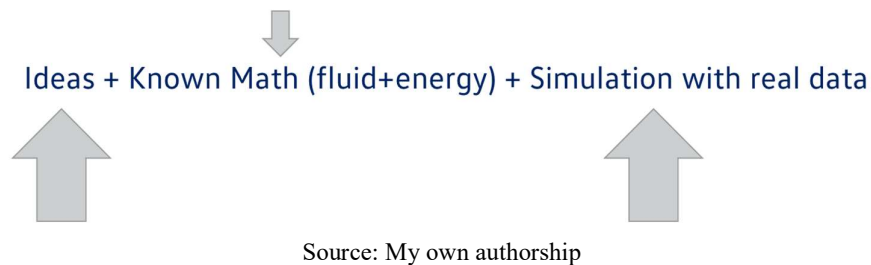
This work aims to show two innovative ideas and two further steps for integrating it with an existing system. In order to bring a flavor of what would it be an cooling system fully ruled from a ORC working fluid, an Cold EGR system flowing outside the turbocharger is simulated and a super heater is modelled by using GT power (see section 3.4.2). To ensure its feasibility, a further calculation is performed in order to integrate both systems with an exhaust gases WHR system.

## 1.4 SCIENTIFIC CONTRIBUTION

The purposed Cold Engine gas Recirculation system (see section 3.3) gives an important scientific contribution as soon as it removes the cooler and remains gas flow through outside the turbocharger, which proves its main finding and its clear innovation when compared with a low pressure EGR (see section 2.2.3).

The super heater around liners also brings innovation since it explores a source of heat not usually explored, showing feasibility when analyzed from the thermal exchange. Using energy from the cooling system in the proposed direction brings large advantage when compared with WHR systems using exhaust gases energy. The necessary space and hardware installation is potentially lower when compared with the heat exchangers built on the vehicle environment instead of inside the engine.

Figure 2: Overall contribution to technical community



The main contribution are the ideas itself, supported with the simulations by using known Math methods. This combination aims to demonstrate that the evolution from air cooled to “water” cooled engines could have taken a more effective way: instead of air, another kind of “gas” should have been chosen to recover heat, and not to reject it more efficiently with “water”.

Besides the contribution to scientific community, it is in line with the continuous improvement mind set suggested by the OEM which supports it.

## 2 LITERATURE REVIEW

Vehicle weight and drag reductions can contribute both in the short and long terms. The longer-term options for moving beyond such improvements currently appear to be plug-in electric hybrids and electricity, and fuel cells and hydrogen. Compelling visions of efficient low emission transportation to use these two energy carriers are still under development for series production, and for a reasonable balancing of vehicle cost and fuel consumption reduction (BANDIVADEKAR, 2008).

The short-term changes (weight, aerodynamics and tire rolling resistance) can be achieved as a result of vehicle redesign, vehicle size reduction, and the use of lighter materials. While some aspects of vehicle functionality may be diminished, the basic mobility offered to consumers by personal transportation can be maintained. Even so, they still result in a vehicle cost increase.

More complex or advanced technologies, such as hybrids, will take longer to result in significant overall reductions, and to reach better TCO due to their higher cost and slower deployment. Significantly different technologies, such as plug-in hybrids and hydrogen and fuel cells, could take over 30 years to be developed to a point where they are feasible to the market, and able to be deployed in substantial numbers. The additional costs of these advanced vehicles are higher, but are proving to be feasible compromises every day. The development and introduction of advanced technology vehicles are first of all motivated by an environmental appeal and, therefore, explored by researchers (BANDIVADEKAR, 2008).

Policies developed to reduce vehicle fuel consumption will need to take into account the trade-offs of vehicle performance, size, weight, and fuel consumption, providing market “pull” for these attributes. It means that those combinations shall also match customer preferences. In passenger cars, this characteristic is memorable, but in commercial vehicles, the emphasis on performance and fuel consumption improvements are determinative.

The contribution of biofuels, such as ethanol and biodiesel, are likely to be constrained by land availability, as well as by biomass yields, their environmental impacts, and costs. It is therefore important that policy efforts focus simultaneously on measures that improve both energy security and carbon and NOx emissions.

Fuel consumption and emissions can be significantly reduced. How rapidly that reduction occurs depends on the determination of major stakeholder groups to vigorously undertake the actions required: vehicle and fuel suppliers, vehicle and fuel purchasers, users and governments.

As worldwide demand for transportation services continues to grow, no single major development is foreseen that could solve the problems of vehicle fuel consumption and gas emissions alone. Short-term actions, such as renewing running fleet, would bring positive results, but new developments are at the front load aiming this result.

An especially promising opportunity is the development and deployment of more efficient propulsion system engines and transmissions. Government policies normally further the overall objectives of road transportation system, as well as the reduction of its energy and environmental impacts. OEM are commonly involved in the decisions, since the market must be carefully analyzed in order not create products that wouldn't be naturally accepted by customers, and would negatively impact the whole economy. Alongside regulatory instruments, incentive-based policies such as fee bates, taxes, pay-as-you-drive insurance, and scrappage incentives can be used (BANDIVADEKAR, 2008).

Most believe that not much will happen without appropriate policies to push and pull improved technologies and greener alternative fuels into the marketplace in high volumes, but fuel consumption reduction attracts the market, and is directly proportional to the raw emissions reduction. In the following pages, alternative ways of improving energy use will be presented, aiming customer satisfaction and not only government regulations.

This chapter presents references about diesel combustion engines, its main components, and the description of some existing systems for the fulfillment of emissions legislation with the correspondent test benches applications.

WHR systems, as well as boilers concepts, expander types, and other WHR applications are also described. Finally, there is an overview of a brief discussion about the behavior of materials exposed to thermal fatigue.

## 2.1 INTERNAL COMBUSTION ENGINES

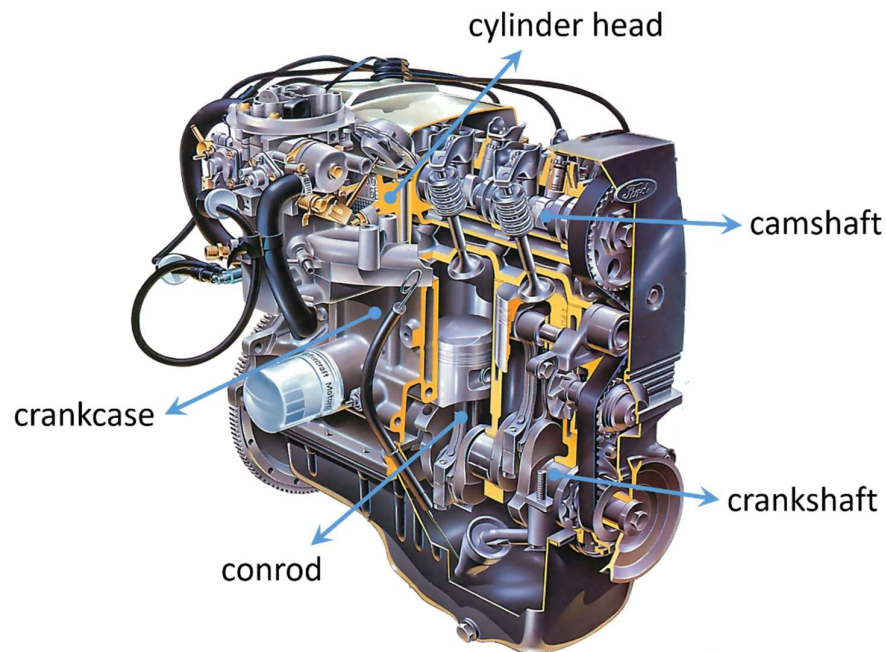
Internal Combustion Engines (ICE) are thermal machines able to convert chemical energy from fuel combustion into mechanical energy. It can be used for directly driving vehicles powertrains with mechanical power, or to run generators that can provide electrical energy for several systems, including hybrid powertrains.

Heywood (1988) makes a wide description about internal combustion engines showing its main types, components and functions. They can be classified by application: Automobile, truck, aircraft, marine, stationary or power generation; by design: Reciprocating in "V" form,

inline, radial and opposite, or rotary engines like Wankel and other geometries; by working cycles: four or two cycles, both varying between natural aspirated, turbocharged or supercharged; by valve location: overhead or under head; fuel type: gasoline, oil, natural gas, alcohol, and others; by method of ignition /cycle: spark (Otto) or compression (Diesel) ; by cooling method: water or air cooled engines.

Conventional Internal Combustion Engines (ICE) have 5 basic components, known as the 5 Cs, as shown in Figure 3:

Figure 3: Engine 5Cs



Source: Adapted from Howacarworks (2019).

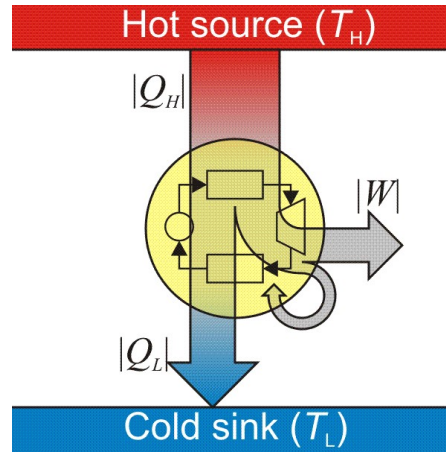
Many different configurations are available on conventional reciprocating ICE. Individual or camshafts on cylinder Head, on Crankcase. Engines from 1 to multiple cylinders. Iron or aluminum casted crankcases. Even though, all 5Cs are always present.

In order to help demonstrating the principle that the wasted heat from coolant can be recovered, it is important to know the engine heat balance and some features about the combustion chamber, where all the energy is converted from chemical to thermal.

### 2.1.1 Heat Balance

As typical thermal machines, ICE are low-efficiency systems, and heat losses are inherent from its concept. Thermal machines heat flow can be observed in Figure 4.

Figure 4: Thermal machine heat flow

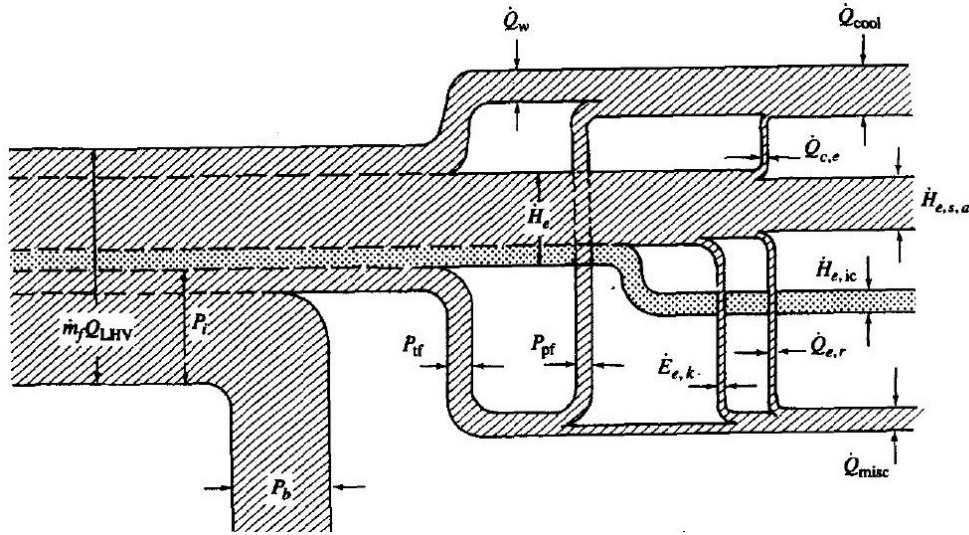


Source: MIKEY (2019).

Making an analogy with combustion engines, The hot source can be considered the chemical energy ( $Q_H$ ) provided by the fuel combustion. The cold sink represents the environment where the ICE works. The mechanisms from the ICE convert the chemical energy into mechanical work ( $W$ ) and the remaining energy is rejected mainly through exhaust and cooling systems. Friction has also an important influence on those losses and are explored by Loureiro (2009) with a parametric model from the engine moving parts (conrod, crankshaft and piston).

There are many sorts of lost energy not converted into mechanical work. The engine energy balance is shown in Figure 5. On the left side it is possible to observe the energy generated by the burned fuel. To the right side all engine energy losses are pointed and one of them is especially interesting for the present study:  $(\dot{Q}_{cool})$  is the energy that flows through the combustion chamber walls to the cooling system. ;  $\dot{H}_{e.s.a}$  and  $\dot{H}_{e.ic}$  are also important, since it represents the energy lost to the exhaust system, which is better explored in section 2.5.

Figure 5: Energy Flow diagram for IC Engine



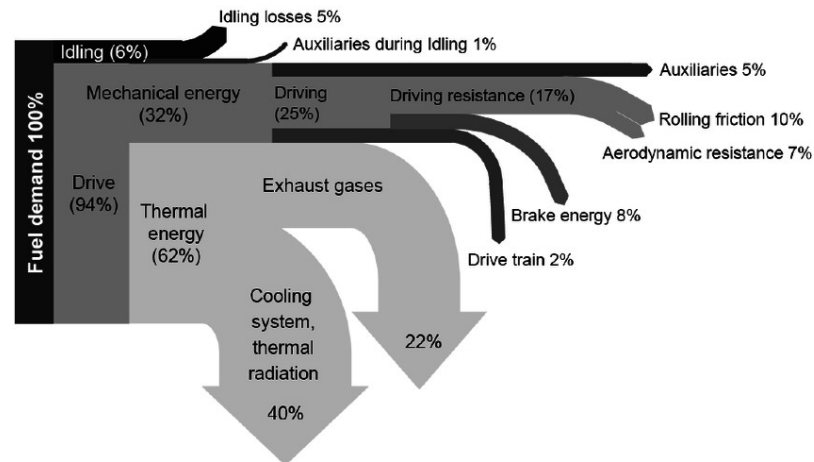
Source: Source: Adapted from Heywood (1988).

Energy flow diagram for IC engine:  $(\dot{m}_f \dot{Q}_{LHV})$  = fuel rate x lower heating value;  $\dot{Q}_w$  = heat transfer rate to combustion chamber wall;  $\dot{H}_e$  = exhaust gas enthalpy flow;  $P_b$  = brake power;  $P_{tf}$  = total friction power;  $P_i$  = indicated power;  $P_{pf}$  = piston friction power;  $\dot{Q}_{cool}$  = heat rejection rate to coolant;  $\dot{Q}_{c,e}$  = heat transfer ratio to coolant in exhaust ports;  $\dot{H}_{e,s,a}$  = exhaust sensible enthalpy flow entering atmosphere;  $\dot{H}_{e,ic}$  = exhaust chemical enthalpy flow due to incomplete combustion;  $\dot{Q}_{e,r}$  = heat flow radiated from exhaust system;  $\dot{E}_{e,k}$  = exhaust kinetic energy flow;  $\dot{Q}_{misc}$  = sum of remaining energy flow and transfers.

Another interesting way to observe the heat balance is by analyzing the vehicle as a whole. Hay et al. (2011) describes it in his paper and concludes that ca. of 62% of the fuel demand is converted in thermal energy, which is currently wasted. The baseline engine heat flow is described in section 3.2.

There are many sort of energy lost on the engine itself, but the vehicle also contributes on it, as can be observed on Figure 6. Notice that the Mechanical Energy can be compared to the value  $P_b$  from Figure 5.

Figure 6: Vehicle Energy flow diagram



Source: Hay et al. (2011).

In this example, approximately 40% of the generated energy is lost to the cooling system. Most of the WHR systems, described in section 2.5, use the exhaust gases as heat source. As it demands a lot of space in the vehicle, this project suggests a solution for energy recovery from cooling system with low packaging needs.

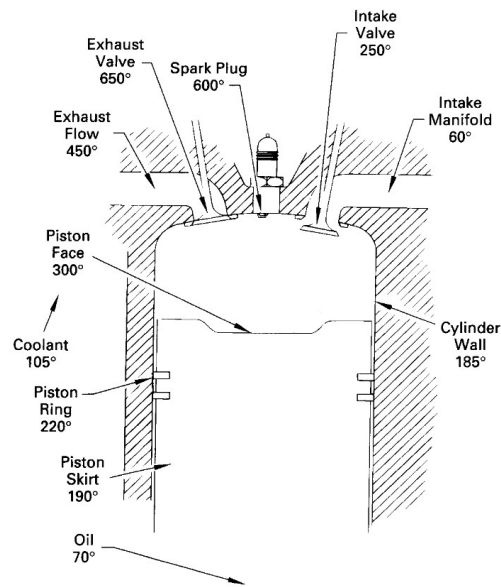
### 2.1.2 Engine Cylinder

The combustion chamber is the variable volume available in each engine cylinder to perform a controlled fuel and air combustion, so that energy can be transferred to the piston head. This region is limited by the engine cylinders, which are described by Heywood (1988) as part of the engine crankcase structure. Some applications use removable cylinder sleeves, which are commonly called liners. Depending on the construction, they can be wet (direct contact with coolant) or dry (indirect contact with coolant).

Both cylinder sleeve types were analyzed to evaluate the level of contact from cylinders with coolant, and can be seen in details in section 3.4. In this section, the presence of a liner doesn't directly mean a wet version.

The movement of the pistons inside the chamber happens between the Bottom Dead Center (BDC) and the Top Dead Center (TDC). The linear difference between those points determines the piston stroke and, consequently, the engine volume by using the cylinder round area and number of cylinders. Pulkrabek (2003) shows typical temperature values found in a Spark Ignition (SI) Engine operating at a Steady state condition in Figure 7.

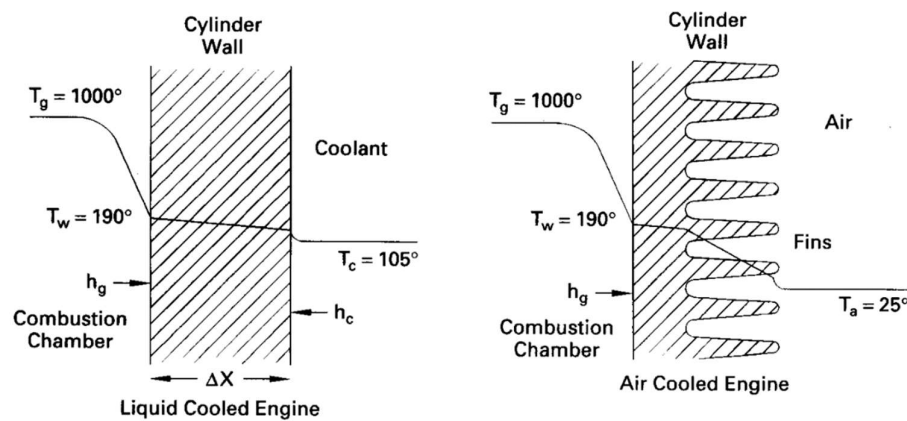
Figure 7: Typical Temperature Values found in a SI Engine



Source: Pulkrabek (2003).

The wall temperature decreases linearly in direction to the Colling galleries varying from 185° to 105°. In section 4.2, empirical data shows temperature of 190° on TDC for the diesel engine in study. The same principle can also be observed on air cooled engines concepts like shown in Figure 8.

Figure 8: Heat Flow through the combustion chamber cylinder wall



Source: Pulkrabek (2003).

The cylinder gas temperature  $T_g$  and convection heat transfer coefficient  $h_g$  vary over ranges for each engine cycle, while the coolant temperature  $T_c$  (or air temperature  $T_a$ ) and heat transfer coefficient  $h_c$  are fairly constant. As a result, heat conduction is cyclic for small depth into the cylinder wall on the combustion chamber side (PULKRABEK, 2003).

## 2.2 SYSTEMS FOR DIESEL EMISSIONS REDUCTION

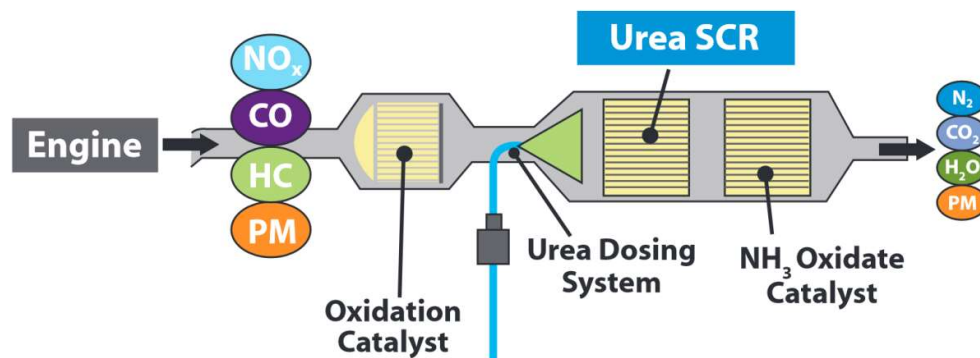
At constant vehicle performance and size, a 30–50% reduction in the fuel consumption of new light-duty vehicles is feasible over the next 20–30 years. The greater uncertainty lies with the time necessary to achieve these changes, rather than the technological options available to achieve them. In the short-term, a combination of improved gasoline and diesel engines and transmissions, and gasoline hybrids, can achieve reductions on this path (BANDIVADEKAR, 2008).

One of the ways for reaching those targets are increasing systems efficiency. The actions suggested for thermal machines improvements can be converted either on increasing liquid power ( $W$ ) or decreasing fuel consumption ( $Q_H$ ), which would mean directly decreasing emissions. However, there are technical limits for it and not only actions inside the ICE are conducted to decrease emissions. Add-on are nowadays required for reducing emissions and are installed in series with the combustion engine. This chapter shows three examples of after treatment systems used in diesel cycles.

### 2.2.1 SCR System

A Selective Catalytic Reduction (SCR) system is a technology that injects urea (a liquid-reductant agent) through a controlled injector into the exhaust stream of a diesel engine (see Figure 9). The urea sets off a chemical reaction that converts nitrogen oxides into nitrogen and water, which is then expelled through the vehicle tailpipe (DIESEL FORUM, 2019).

Figure 9: SCR system.

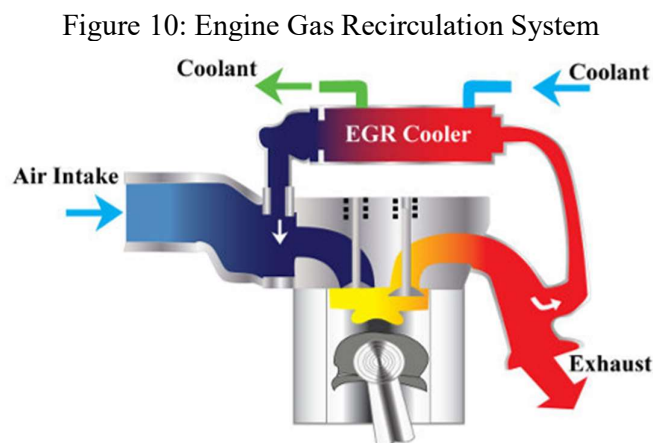


Source: Diesel Forum (2019).

SCR technology is one of the most cost-effective and fuel-efficient technologies available to help reduce emissions. SCR can reduce NO<sub>x</sub> emissions up to 90%, while simultaneously reducing Hydro Carbonite (HC) and CO emissions by 50-90%, and Particulate Mass (PM) emissions by 30-50%. SCR systems can also be combined with a diesel particulate filter to achieve even greater emission reductions for PM (DIESEL FORUM, 2019).

### 2.2.2 High Pressure EGR System

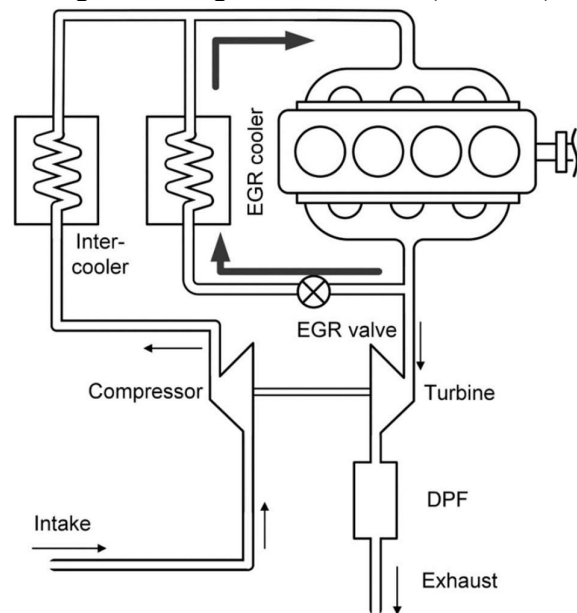
The EGR system showed in Figure 10 reduces NO<sub>x</sub> by reducing the temperature at which fuel burns into the combustion chamber. Engines employing EGR recycle a portion of engine exhaust gases, mixing it with the clean engine air intake. The oxygen-depleted exhaust gas (inert gas) is mixed to the engine intake fresh air. The reduction in oxygen produces a lower temperature burn, which leads to less NO<sub>x</sub> emission. The recycled exhaust gas can also be cooled, which further reduces NO<sub>x</sub> emissions (HOUNTLAS et al., 2007).



Source: Toyota Parts Center (2019).

In this case, the recirculation is possible because the exhaust gases are taken from the pressure line before the turbocharger. This EGR system variation is called High Pressure EGR, which schema can be observed in Figure 11. During the Exhaust phase, the pistons pressurize the gases to expel it. Therefore, the pressure on this line is high enough to flow through the cooler and to be inserted into the fresh air line.

Figure 11: High Pressure EGR (HP EGR)



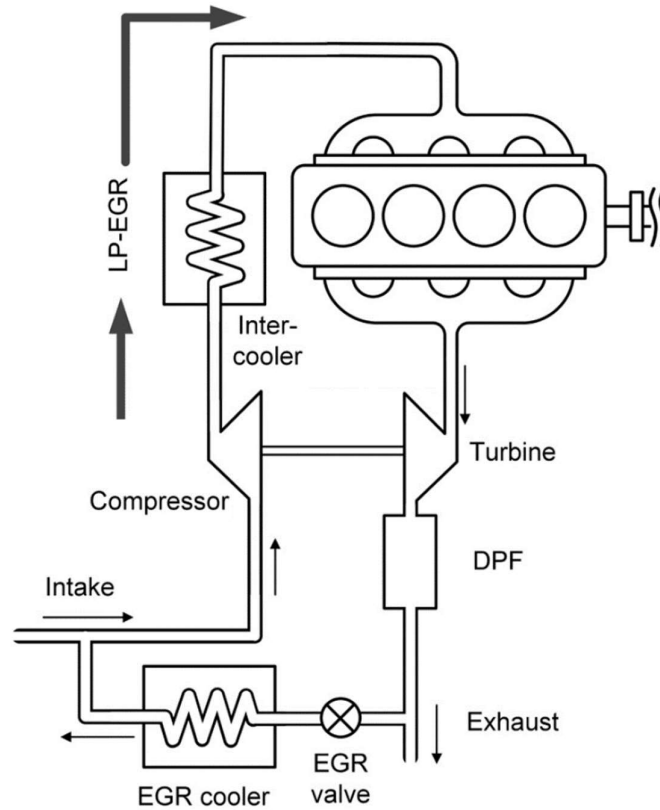
Source: Adapted from Kim et al. (2017).

Notice that the system needs the gases to be cooled down to a temperature level which when mixed with intake fresh air, still fulfills the temperature limits for combustion chambers intake.

### 2.2.3 Low Pressure EGR system

The Low Pressure EGR (LP-EGR) introduces exhaust gases from the downstream side of the turbine to the upstream of the compressor. The exhaust gases after the turbine mixes with fresh air before the compressor. It can be cooled before entering compressor and have further cooling into Char Air Cooler (CAC), which can also be called intercooler. LP-EGR has advantages in both reducing the energy loss from the exhaust gases drain, and the possibility of eliminating the ERG cooler. However, the water condensation and the slow response are the demerits of LP-EGR. Condensation of water due to low temperature of the LP-EGR involves the risk of corrosive attack of all the wet components, such as the compressor wheel. The response of the EGR rate change is slower than that of HP-EGR during an abrupt load change because the loop of LP-EGR is longer than that of the HP-EGR. LP-EGR and HP-EGR applications depend on the load conditions.

Figure 12: Low Pressure EGR (LP EGR)



Source: Adapted from Kim et al. (2017).

Low pressure EGR seems to be the easier solution for mixing inert gases with the clean air HP line, but a the side effect has a huge impact on the turbocharger life. The cold side (compressor) works in contact with particulate, creating deposits on the wheel creating soot deposits (oil cooking) in it due to the high temperatures involved. In section 3.3 a system, that is not a Low pressure EGR, is purposed to solve the turbo charger contamination problem.

## 2.3 SOURCES FOR ENERGY RECOVERY

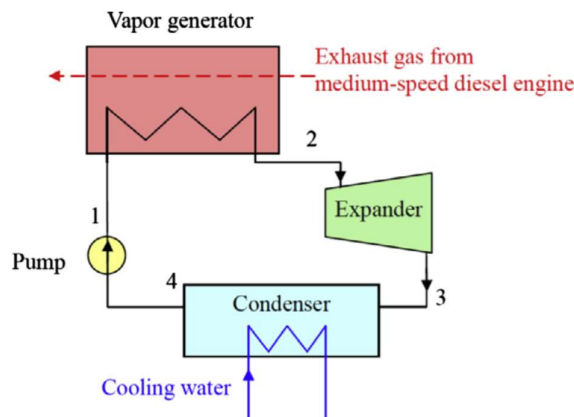
Energy sources are explored with references to boilers and combustion engine heat balance. To use this energy, conventional and organic Rankine cycles are researched; to recovery it, the different kinds of steam turbines are also referenced.

### 2.3.1 Exhaust system

The exhaust system is basically the routing between the Engine and the tail pipe, where gases are expelled to the atmosphere. Due to legislation requirements, this routing started as part of the engine logic instead of just a gas track to the atmosphere. Therefore, nowadays it is referred to as the After-Treatment System, which keeps its very first identity, including even many other functions, as explained in section 2.2.

Yang et al., 2017 describe a Waste Heat Recovery (WHR) system applied to a marine medium-speed diesel engine, briefly represented in Figure 13.

Figure 13: Exhaust gases WHR system.



Source: Yang et al. (2017).

The exhaust gases flow through a heat exchanger that transfer heat to the Rankine cycle working fluid. This heat, vaporizes the working fluid that, in contact with the expander, generates mechanical power. The remaining energy flows through the condenser which uses the sea water (in this application) changing the working fluid completely back to the liquid phases. Finally the pump rises the fluid pressure and the cycle starts over.

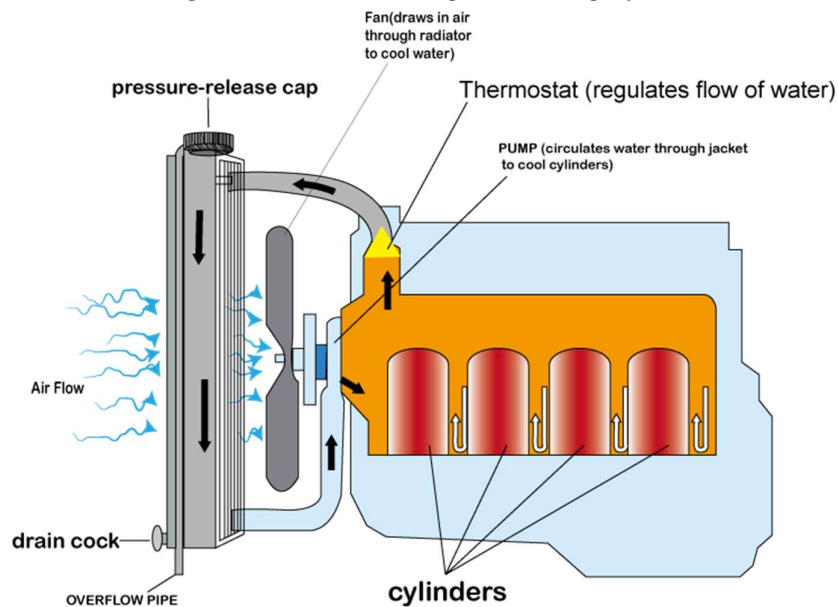
Although an important amount of the enthalpy is recovered at the turbocharger, the remaining exhaust gas energy is still high. Even so, under the new requirements of the legislation, this energy cannot be fully utilized for recovery because high exhaust line temperatures are required due to the need for catalytic reactions (DIESEL FORUM, 2019).

This new legislation scenario leads to lower emissions, but consumes more system energy, leading to higher fuel consumption. To meet TCO requirements, customers would no longer spend on fuel and the system itself. Therefore, efforts to recover energy also need to be carried to other systems.

### 2.3.2 Engine Cooling Systems

Thermal machines reject heat as a working principle. The ICE cycles are not different. To remove heat from the Engine, the cooling system can be based on Air, Water or can be an evaporative system. The air cooled Engine were common in the past for passenger car and are highly used for motorcycle engines. Its principle is based on fins outside the combustion chambers which enlarger the contact with the surrounding air promoting heat exchange by convection. The air flow through the fins can be natural (by vehicle movement) or can be forced via a fan. The water cooled engines includes chambers on the crankcase (see Figure 90 in section 4.2) in which water + coolant flows. A common engine water cooling system is analyzed in Figure 14.

Figure 14: Common Engine Cooling System

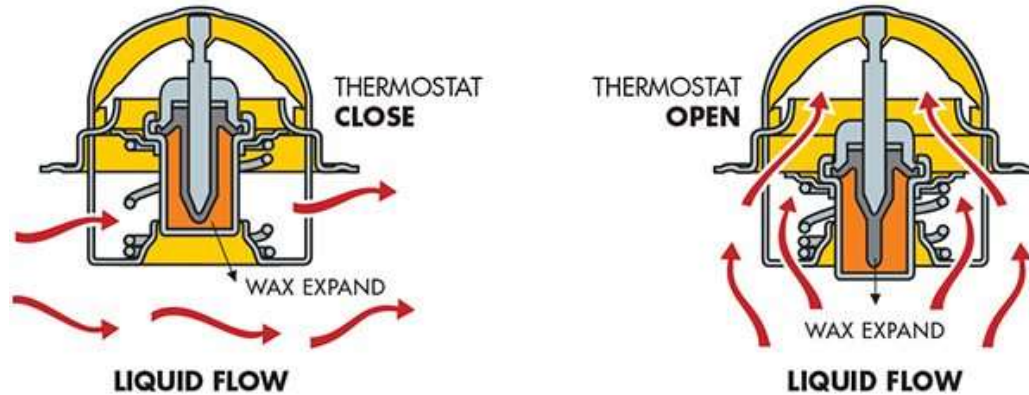


Source: Easyautocooling (2019).

As the cylinders reject heat from the coolant, its temperature rises and the thermostat valve (see Figure 15) opens to transfer heat to the radiator, which rejects it to the environment. See the coolant flow and main heat exchanges in Figure 72, section 3.4.1.

If the engine heat rejection is high, the coolant will increase the temperature, releasing the flow to the radiator, but if it is low, the valve will remain closed, keeping coolant flowing through the valve back into the engine, keeping the system at the same temperature until the cycle starts again.

Figure 15: Thermostat valve close and open positions



Source: TU-POLY (2019).

The thermostat valve controls the engine working temperature, which is specific to each engine design. It limits an upper and bottom temperature range with CLOSE and OPEN positions. Higher working temperatures lead to more efficient thermal machines, but materials and fluid boundaries limit it. The valve is normally closed using a spring that allows free flow of coolant into the engine chambers until the idling engine operating temperature is reached. An internal mechanism acts with a temperature-sensitive wax that expands and allows the thermostat valve to open, releasing fluid to the radiator. Once the ideal working temperature is reached, the cycle begins again.

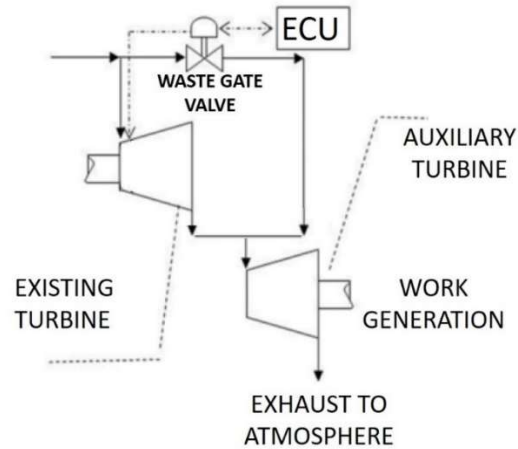
The evaporative cooling system is barely used, but its concept has been explored in many patents requests and papers available for study. NISSAN, 1987, stated and evaporative system using a coolant jacket into which coolant is introduced in liquid state from which the coolant is discharged in gaseous state. This system needs an condenser to change the coolant state back into liquid and a tank to collect the liquid state coolant for starting over the cycle. This is a known system in academic researches, but has not wide commercial use for automotive vehicle.

### 2.3.3 Turbo compound

Turbo compound is the term used for an alternative engine that employs a turbine to recover energy from the exhaust. The turbine has a mechanical link to the output shaft, which converts enthalpy into mechanical energy. All features included in compliance with the exhaust flow can produce back pressure in the system, but many researches, such as Cipollone et al. (2013), have shown some benefits of its use.

ICE works in transient conditions and, therefore, the air intake conditions and fuel amount are not constant. In this way, it is also clear that the exhaust gases available to the turbocharger are also not constant. Considering the main turbine (“existing turbine” in Figure 16) as an Brayton cycle, the necessary energy generation to insert more gases into the engine is also not constant. Therefore, a “waste gate valve” is used to bypass the turbine in cases where engine boost is not necessary. Figure 16 shows a turbo compound (auxiliary turbine) that works in the exhaust gases line.

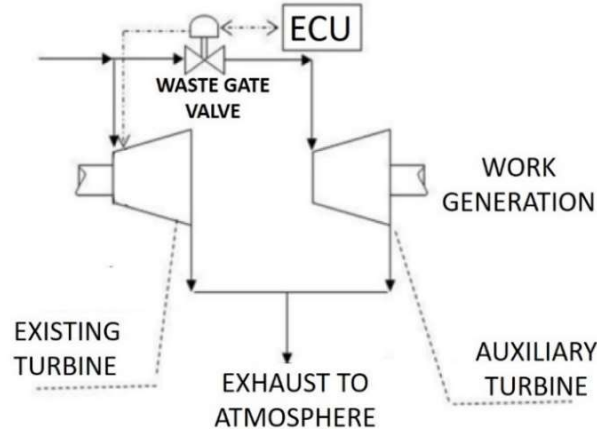
Figure 16: Conventional turbo compound system



Source: Adapted from Cipollone et al. (2013).

This configuration adds a pressure drop on the exhaust line when the auxiliary turbine is working. The direct use of the bypass gases is represented in Figure 17, what means an alternative not to create back pressure during load conditions. This layout remains the main turbine outlet free.

Figure 17: Parallel Turbo compound system



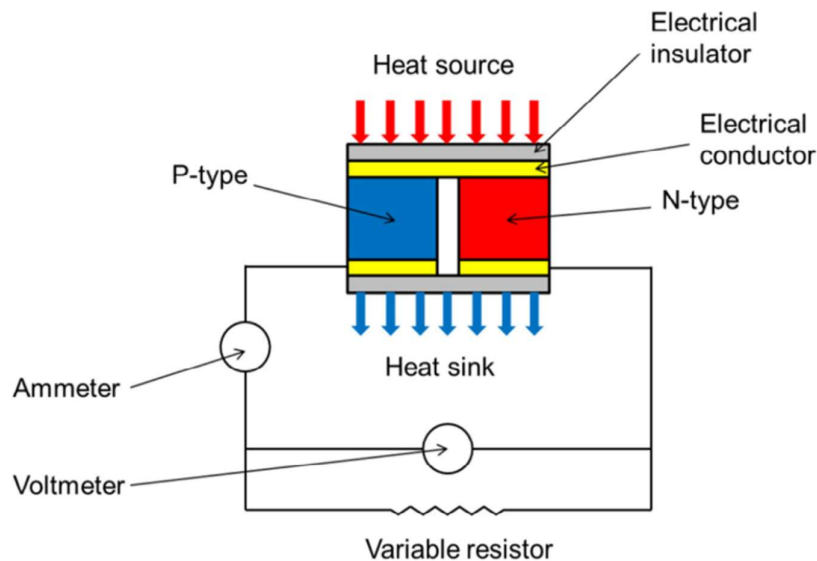
Source: Adapted from Cipollone et al. (2013).

Exhaust gas cylinders have two main energy contributions: the first is related to its pressure, the second to its temperature. Therefore, the work can be recovered by direct expansion, considering that gas pressure leaving the cylinders is higher than the ambient pressure and by a heat recovery, with the gas temperature higher than the ambient temperature (CIPOLLONE et al., 2013).

### 2.3.4 Peltier / Seebeck Heat Recovery

Orr et al. (2016) affirm that between 20% and 45% of a combustion Engine is converted in work. The remaining 55% to 80% is commonly wasted as heat in both coolant and exhaust gases. They suggest using heat pipes in combination with Thermal Energy Generators (TEGs) to recover part of the wasted energy. Using the seebeck effect, they assemble the TEG electrically in series, but thermally in parallel, as shown in Figure 18.

Figure 18: Seebeck Effect

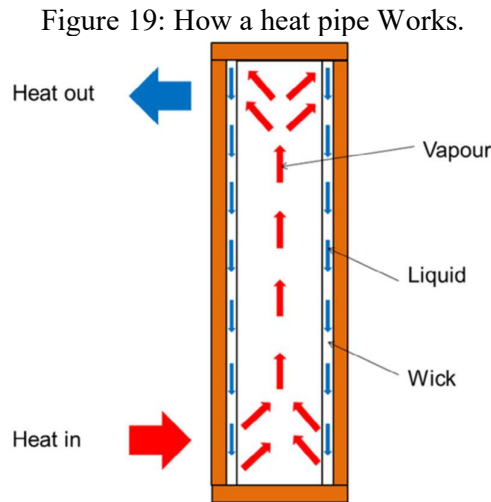


Source: Orr et al. (2016).

Even with low efficiency (around 5%), TEGs have some advantages, such as reduced size and no noise emission. As no moving parts and no chemical reactions are required, this equipment has a high expectation of durability and low maintenance (ORR et al., 2016).

Heat pipes are also used in combination with TEGs. The heat pipes work as a solution to increase the available area for the installation of TEGs. This area is normally limited to the exhaust manifold and pipe surfaces.

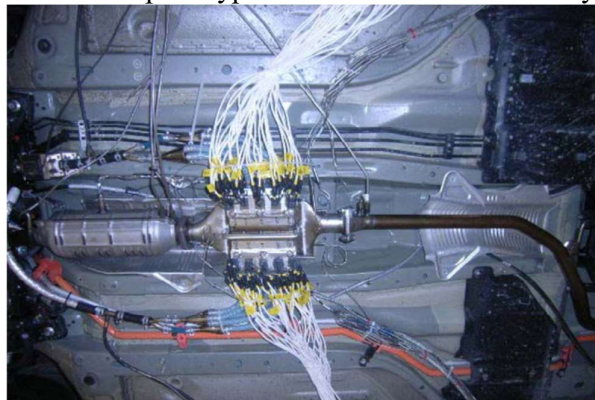
A heat pipe, shown in Figure 19, has a simple principle: it consists of a sealed metal cylinder filled with a fluid that works in a saturated region based on a negative pressure that decreases its boiling point. This equipment is used to “transport” heat over relatively long distances with high thermal conductivity.



Source: Orr et al. (2016).

Orr et al. (2016) exposes the recovery achieved by some OEMs (Original Equipment Manufacturer) in experimental analyzes. Using a shell and tube heat exchanger, BMW was able to produce 700 W. Using thermoelectric materials in contact with exhaust gases and cooling water to provide temperature gradient, Ford was able to recover 400 W. The Renault model expected to produce approximately 1000 W using a counter flow heat exchanger, combined with cooling fluid from a diesel engine. Honda also used a heat exchanger (Figure 20) combined with coolant, producing 500 W, which translated to a 3% reduction of fuel consumption.

Figure 20: Honda prototype TEG exhaust heat recovery system



Source: Orr et al. (2016).

TEGs are typically placed on the tail pipe surface. Saqr et al. (2008) show some kinds of heat exchangers, such as the rectangular and hexagon shape, respectively shown in Figure 21 and Figure 22.

Figure 21: Rectangular exhaust heat exchanger

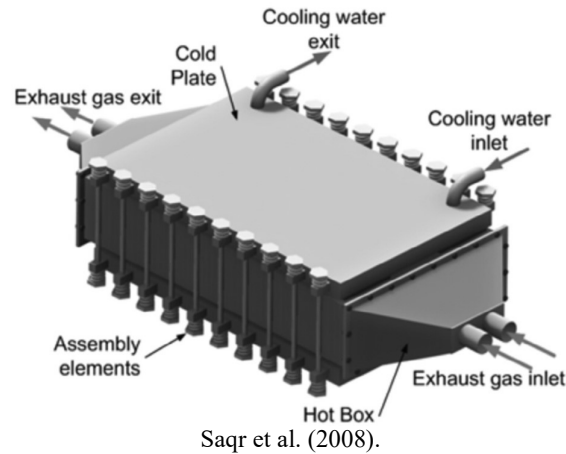
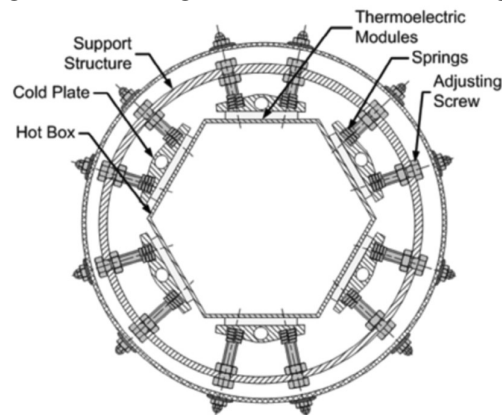


Figure 22: Hexagonal exhaust heat exchanger



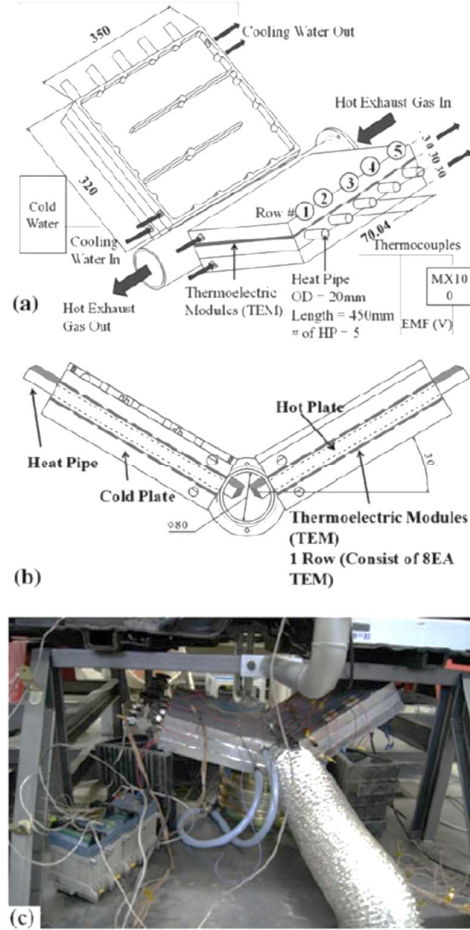
Source: Saqr et al. (2008).

Kim et al. (2011) and Baatar et al. (2011) developed a heat exchanger based on heat pipes and TEGs that was able to replace the vehicle radiator using the same packaging. The purposed systems consist of 72 TEGs and 128 Heat pipes. When the vehicle was stopped, it was possible to generate 28 W of power with 20 °C difference between the cold and hot sides. By 80 km/h, the system was able to generate 75 W, where the difference from the hot to cold side was about 45 °C.

Kim et al. (2011) also developed a heat recovery system using TEGs and heat pipes on exhaust tubes, showed in Figure 23. The heat pipes were transversally inserted into an aluminum block. One side of the TEGs was heated with the exhaust gases, while the other side

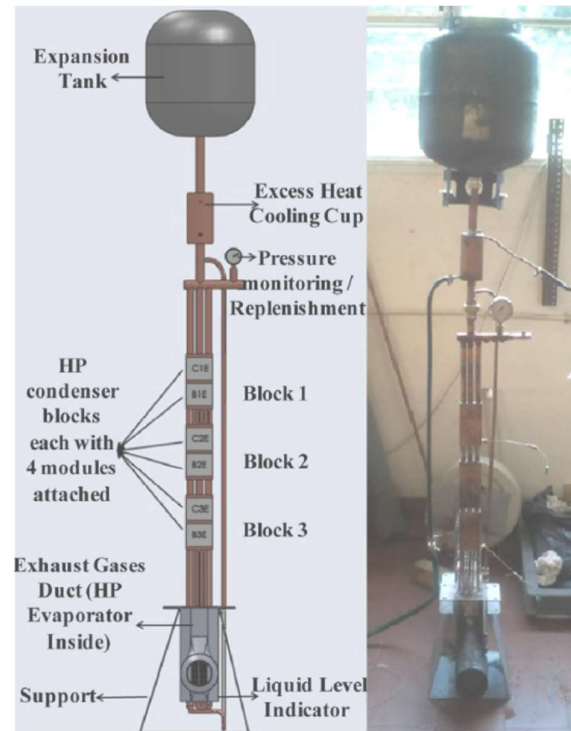
used a water heat sink to cool the TEGs' temperature down. "This system generated a maximum of 350 W using 112 40 mm × 40 mm TEGs".

Figure 23: Exhaust heat recovery



Source: Orr et al. (2016).

Figure 24: Exhaust heat recovery with controlled TEG

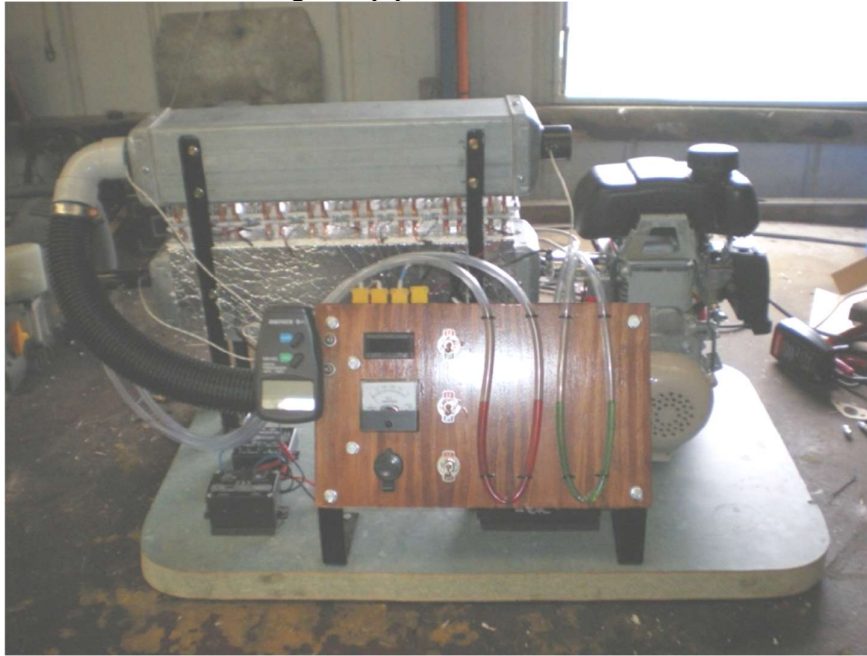


Source: Orr et al. (2016).

A similar system was suggested by Goncalves et al. (2010), and can be seen in Figure 24. The main difference is the usage of a Variable Conductance Heat Pipe (VCHP). A VCHP handles a constant operating temperature, which leads to increased performance and protection of TEGs from overheating.

The system shown in Figure 23 uses heat pipes on both sides of the TEG to transfer heat both to and from the TEG. The system is air cooled using a fan to simulate air flow on a moving car. A counter flow heat exchanger arrangement was used to maximize the heat transfer rate. Exhaust gases were supplied from a small 50 cc gasoline engine. A total of 8 TEGs 40 mm × 40 mm in size were used to generate approximately 6 W of power (ORR et al. 2016).

Figure 25: Exhaust WHR using heat pipes to transfer heat both to and from the TEGs



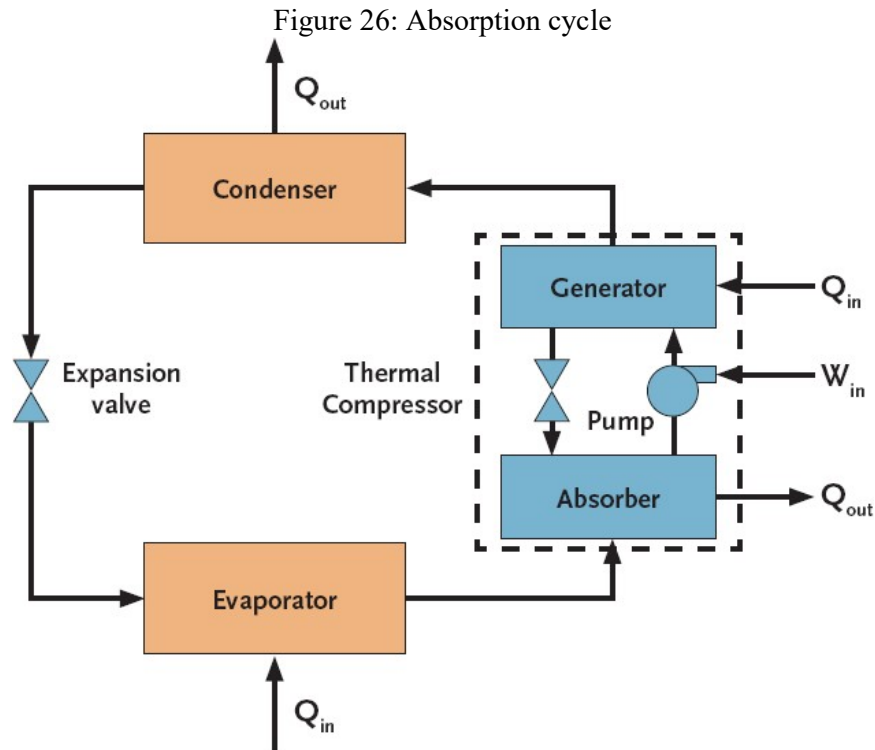
Source: Orr et al. (2016).

Orr et al. (2016) further developed the design shown in Figure 25 to be able to handle higher exhaust temperatures. In this case, thick-walled copper water heat pipes were used to increase the temperature capacity of the heat pipes. TEGs with higher temperature rating and thermal resin with higher temperature rating were also used. A naphthalene heat pipe pre heat exchanger was suggested to reduce the exhaust gas temperature, and to protect the downstream TEGs and copper water heat pipes from overheating. As the naphthalene heat pipes have a working temperature range between 250 °C and 450 °C, when the exhaust gas temperature is low, no heat will be removed. But when the exhaust gas temperature is high, heat will be removed. The naphthalene heat pipes work as a temperature regulator for the incoming exhaust gases. When 8 TEGs 75 mm × 75 mm in size were used, the system produced 54 W of electricity.

Orr et al. (2016) show that the association of TEGs and Heat pipes is feasible for use with combustion engines. However, the system has a low Heat recovery rate in comparison with Rankine Heat Recovery Cycle application systems.

### 2.3.5 Heat Recovery by absorption

Absorption machines are devices that use residual heat of systems to convert it in cold or hot energy sources. A conventional absorption cycle is shown in Figure 26.



Source: Cibsejournal (2019).

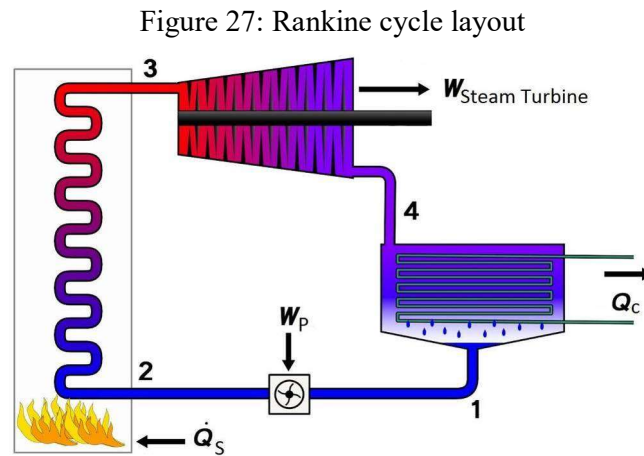
Martinelli Junior (2008) suggests an absorption refrigeration system recommended for applications in the air conditioning systems of buses and trucks, which have a restricted space available for installation. Such system is applied as an alternative vehicular air conditioner. The author demonstrated that the residual heat released by the bus is enough to activate the alternative air conditioning system.

## 2.4 RANKINE CYCLES

To convert heat into mechanical energy, the most suitable thermal machine is the Rankine Cycle. Its use can increase the efficiency of reciprocating engine cycles.

A Rankine cycle contains (Figure 27):

- Super Heater (2-3): Components able to transfer the energy source ( $Q_s$ ) to the cycle working fluid;
- Expander (3-4): Component able to convert working fluid energy into mechanical energy ( $W_s$ ) by decreasing the pressure and temperature (enthalpy);
- Condenser (4-1): Element necessary to remove latent heat from the working fluid, by changing its state from gas to liquid;
- Pump (1-2): Element able to increase liquid pressure by sending it back to the super heater.



Source: Adapted from Ploucha (2019).

The Rankine cycle produces mechanical energy on the expander output shaft, to be used for powertrain drive or secondary functions. Its application to combustion engines is linked to the engine capacity to create the heat source  $Q_s$ .

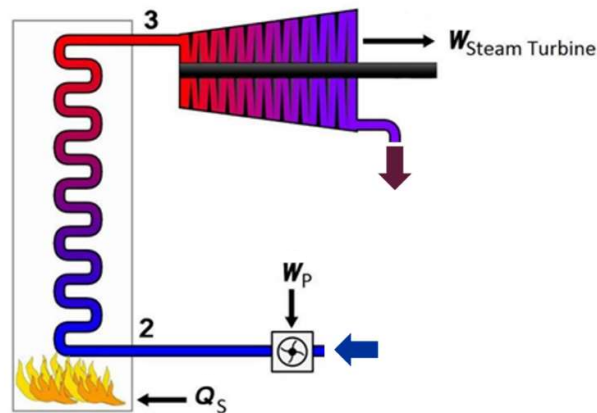
WHR has recently become an important topic for energy use due to energy shortages and increase of carbon dioxide emissions in the world. The organic Rankine cycle system has advantages in recovering the waste heat source to produce useful power. However, there is a pinch point between the working fluid and the heat source in the evaporator due to the constant saturated temperature in the evaporation process.

Compared to the organic Rankine Cycle, the Trans Critical Rankine cycle, TRC, is developed without this minimal temperature difference during heating in the steam generator. Since the steam generating pressure is greater than the critical working fluid pressure in the TRC system, a variable working fluid temperature profile can be obtained to fit the heat source during the heating process. However, an important restriction for the application of the TRC system is that the critical working fluid temperature must be lower than the residual heat source temperature (YANG, 2017).

Rankine cycles require working fluids that matches the system boundary conditions. Parameters such as the temperature of the heat exchanger efficiency in heat sources are indispensable information for choosing working fluids.

Steam locomotives are very good examples to show early experiences with early Rankine Cycles. Using water as working fluid, the locomotives transported water and coal to complete the cycle. In this example, which was not yet a cycle, water was consumed by the system, as shown in Figure 28.

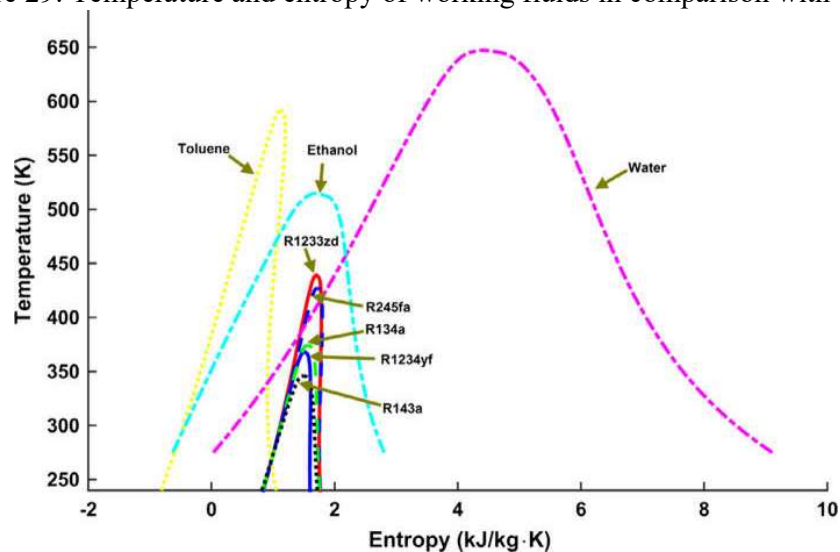
Figure 28: Steam Locomotive principle



Source: Adapted from Ploucha (2019).

All Rankine cycle needs a working fluid to properly generate power. Figure 29 shows some Temperature x Entropy curves of different fluids.

Figure 29: Temperature and entropy of working fluids in comparison with water.

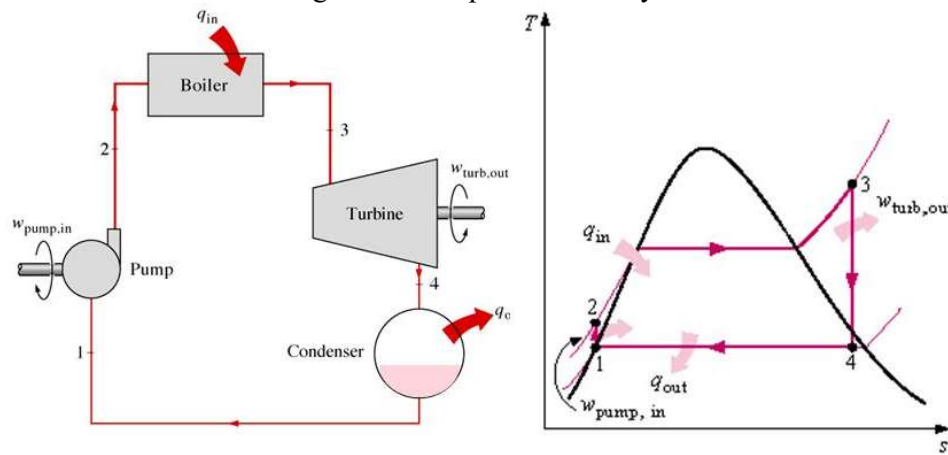


Source: Yang (2017).

As shown in Figure 29, water has a high evaporation point and a wide curve showing its limits of use. For steam locomotives, where steam generation is the machine's main energy source, water does its job, but for heat recovery systems, its use is not recommended. Beyond the technical limits, water is also very corrosive and, for closed cycles, the long-term use is also not recommended.

The use of cycle water also has the drawback of a wide curve, which means high latent heat to be transferred into the boiler (super Heater) and condenser. This feature reinforces the argument that high temperature heat sources are required to use water as working fluids.

Figure 30: Simple Rankine Cycle



Source: Gajjar (2019).

In Figure 30, it can be noted that for proper steam generation, the boiler temperature must be high. At lower temperatures, all efforts to remove heat from the heat source can only be converted to latent heat and not steam generation.

The water is located at one edge of Figure 29 and at the other end conventional refrigerant working fluids can be seen. R1233zd, R245fa, R1234yf, R143a can be easily overheated, but the low condensing temperatures limit their use in heat recovery systems that have a cold sink environment (see Figure 4).

In the middle of this range, ethanol shows good thermal behavior for heat recovery systems, where steam generation is relatively easy and the condensing temperature is not too low with a thin curve. This fluid is used for simulations and for a real OEM application that supports this work.

Ethanol brings many safety concerns and this is the most critical issue that outweighs the thermal benefits, the low costs to market customers, and the low corrosion characteristics compared to water.

## 2.5 WHR SYSTEMS IN COMBUSTION ENGINES

The technology applied to combustion engines has reached a high level of complexity and cost. Thermal deformation studies, new friction materials, new lubricants, new processes, greater cleaning severity and more robust production processes have been developed to decrease fuel consumption and gases emissions. Known in Thermodynamics as cogeneration, waste heat recovery systems are alternative processes that are explored by OEMs and related suppliers.

One option for converting the energy found in the exhaust gas stream consists of Waste Heat Recovery (WHR) systems that convert waste heat to mechanical energy using a Rankine Cycle (RC). In stationary systems, these vapor cycle processes are state of the art. The main challenges when it comes to their mobile use in road traffic are in developing components that are suitable for use in vehicles and achieving a robust operating strategy.

The general requirements are comparable for Passenger Car (PC) and Commercial Vehicle (CV) applications. However, the corresponding operating times as well as route and load profiles are fundamentally different from each other. In addition to contributing to reducing the CO<sub>2</sub> fleet emissions of PC and CV manufacturers, the Total Cost of Ownership (TCO) for customers is vitally important for commercial vehicle applications. In the CV area, using the waste heat from the internal combustion engine is a measure that in the future may contribute to meeting current Europe Union draft legislation for the gradual reduction of CO<sub>2</sub> fleet emissions by 2025 and 2030. In addition, reducing the total TCO with the achievable reduction in fuel consumption is a key focus for commercial vehicles. (CARSTENSEN et al., 2019).

An initial target application for a WHR system within this context consists of vehicles in the cost-driven segment of long-distance traffic. Mileages of 100,000 to 150,000 km per year are absolutely common for these vehicles. Depending on fuel prices, annual fuel costs of several tens of thousands of euros are incurred. The large amount of time that the engine operates in the medium load range at constant speeds above 80 km/h is advantageous. (CARSTENSEN et al., 2019).

The development of a waste heat recovery system leads to increased product cost. However, the commercial vehicle application is increasingly considering the cost in the business case of customers. In other words, it means how expensive it is to have a specific trade in equipment in the fleet. This concept is called Total Cost of Ownership (TCO).

A TCO calculation ideally provides an indication of which equipment would be best for the customer and not just the purchase costs. This calculation includes manpower resources,

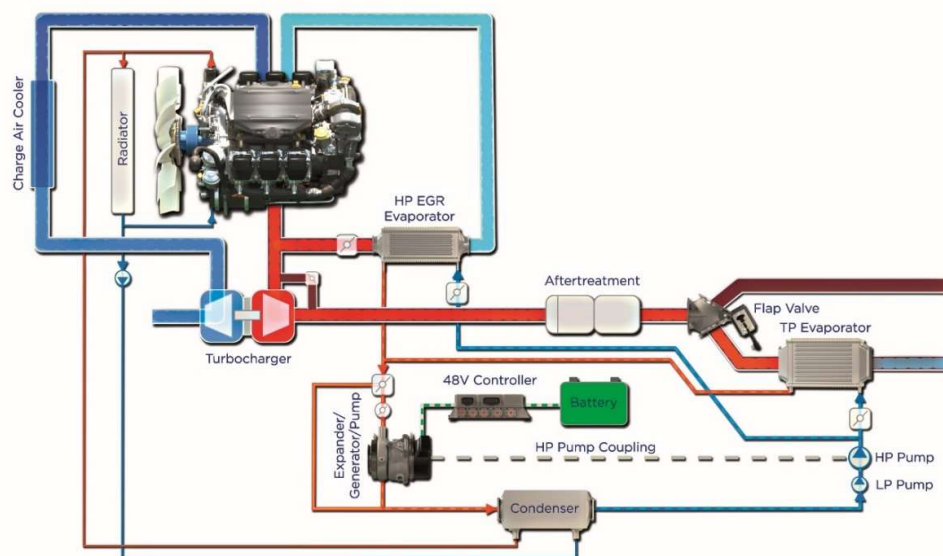
training hours, failure costs (downtime costs), incident preventive maintenance, image cost impacts, non-delivery costs (ongoing contracts), infrastructure costs, and equipment downtime costs.

Some examples of waste heat recovery systems are shown in this chapter, showing their key components that include product costs: Expander, condenser, Super Heater (Evaporator).

### 2.5.1 Borg Warner

Borg Warner Emission Systems (2019) has a Waste Heat Recovery Rankine System in development. The goal is to reduce Diesel Consumption and CO<sub>2</sub> emissions from road trucks to achieve US legislation through the recovery of some Fuel Energy wasted in heat rejection. Their waste heat recovery system generates electrical energy by converting thermal into mechanical energy with an organic Rankine cycle. The system uses fluid phase change similar to air conditioning or refrigeration systems. The cycle is divided into four steps: First, the cool working fluid from the condenser is pumped to high pressure (see section 2.4). Then, waste heat from the engine exhaust gases heats the working fluid to a superheated steam, which drives expander and generates electrical power. The low-pressure vapor is cooled back to a liquid state by the condenser, and the process starts over in a closed cycle (see Figure 31).

Figure 31: Borg warner WHR system with EGR and tail pipe evaporator



Source: Borg Warner emission systems (2019).

Using a 48-volt Generator, Borg Warner ORC turbine expander is well suited for combustion vehicles and supports mild hybrid commercial vehicles, an tendency expected for the next five to ten years. The oil-free bearing and aerospace-inspired turbine allows better efficiency. On the exhaust flow there is a bypass valve that controls the amount of heat entering the waste heat recovery system. The system integrity and durability is also important, which leads to the need of bypassing the tailpipe evaporator at high engine loads.

In short words, the system works with a Rankine Cycle using tail pipe and EGR (Engine Gas Recirculation) as energy sources, which generates mechanical power to feed a generator. The fuel consumption reduction potential ranges from 3% to 5%, depending on the vehicle application (BORG WARNER EMISSION SYTEMS, 2019).

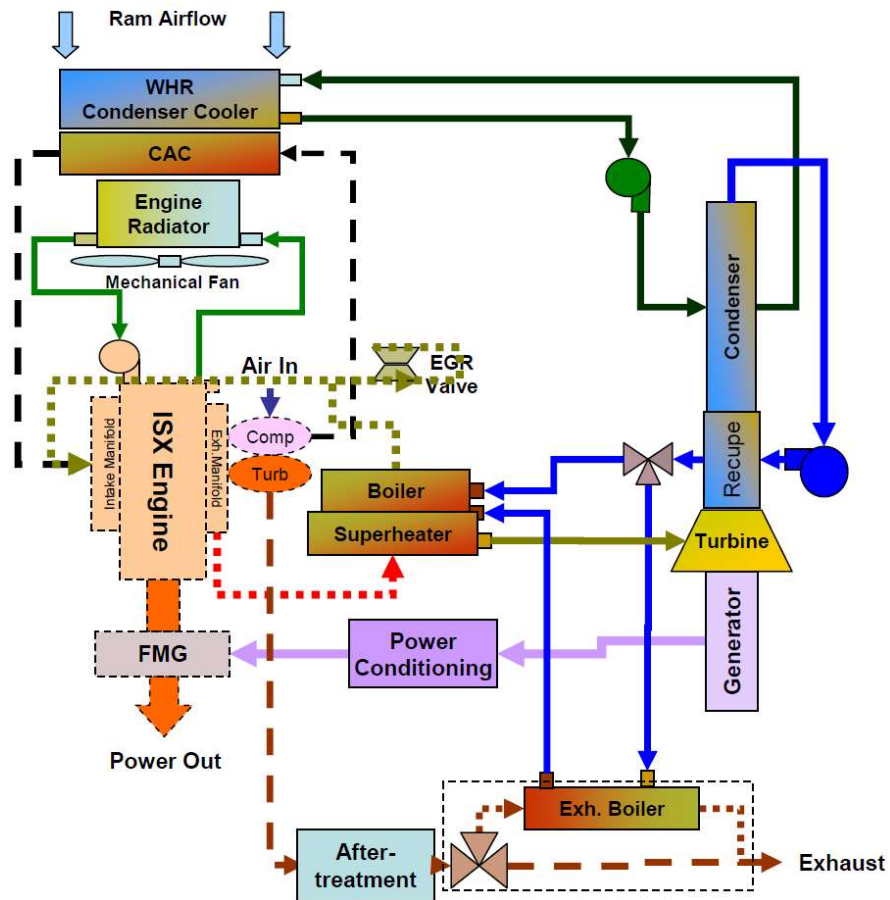
### **2.5.2 Cummins**

Nelson, 2015 shows a “Solutions for a Sustainable Future” in his patent application, offering cogeneration processes. It consists of converting residual thermal heat from the engine exhaust and cooling circuits into additional electrical power or steam for industrial heating applications without additional fuel demand. In addition, with rising fuel prices, waste heat recovery systems are even more attractive today with possible return on capital investment.

The goals are: 10% improvement in fuel efficiency; reduction of the need for increased heat rejection capacity for future heavy duty engines; 10% increase in fuel efficiency that can save, on long haul trucks, over 7,000 liters of fuel per year, covering 200,000 km; reduction of exhaust emissions due to lower fuel use.;

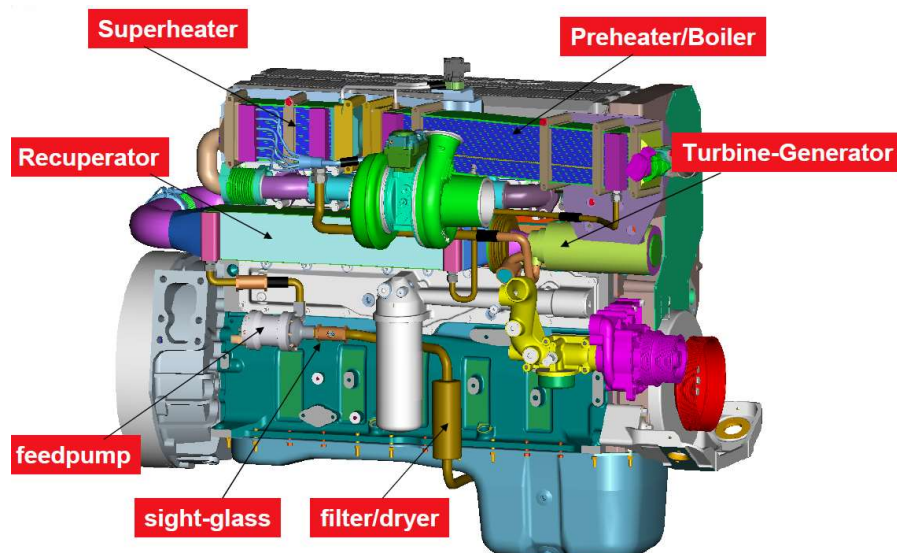
The scheme shown in Figure 32 and the detailed system of Figure 33 show many energy sources that the system intends to use: a boiler receiver preheats the recuperator fluid in the Rankine cycle condenser. A second boiler is installed in the exhaust system. Both preheat the working fluid sent to the super heater. The generated steams are used to run a turbine that runs a generator (NELSON, 2015).

Figure 32: Cummins WHR system Scheme proposal



Source: Nelson (2008).

Figure 33: Cummins WHR engine hardware installation.



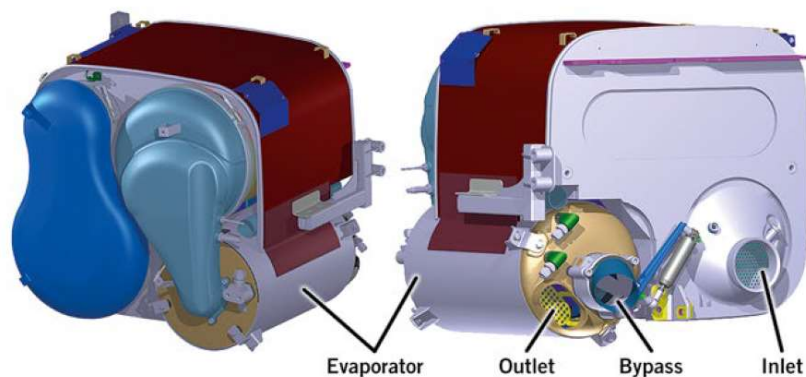
Source: Nelson (2008).

### 2.5.3 MAN WHR system for trucks

This section is a reproduction of the work from Carstensen et al., (2019), property from Volkswagen passenger cars and MAN Truck and Buses. MAN has built a technology demonstrator based on a semi-trailer tractor (TGX 18.480 4×2 BLS) working with an organic RC.

The evaporator, initially designed for passenger cars, is located downstream of the exhaust gas treatment system. The available exhaust gas power in commercial vehicle applications led to the design of a two concentric coil evaporator (superheater), making it possible to further increase heat absorption and improve evaporator efficiency. The evaporator was implemented on an Euro VI exhaust gas treatment box as an additional component (Figure 34). The two coils were arranged concentrically around a centrally located bypass. This bypass uses a production exhaust flap, which makes it possible to modulate or completely bypass the transferred exhaust gas power at any time.

Figure 34: Evaporator integrated on after treatment system box

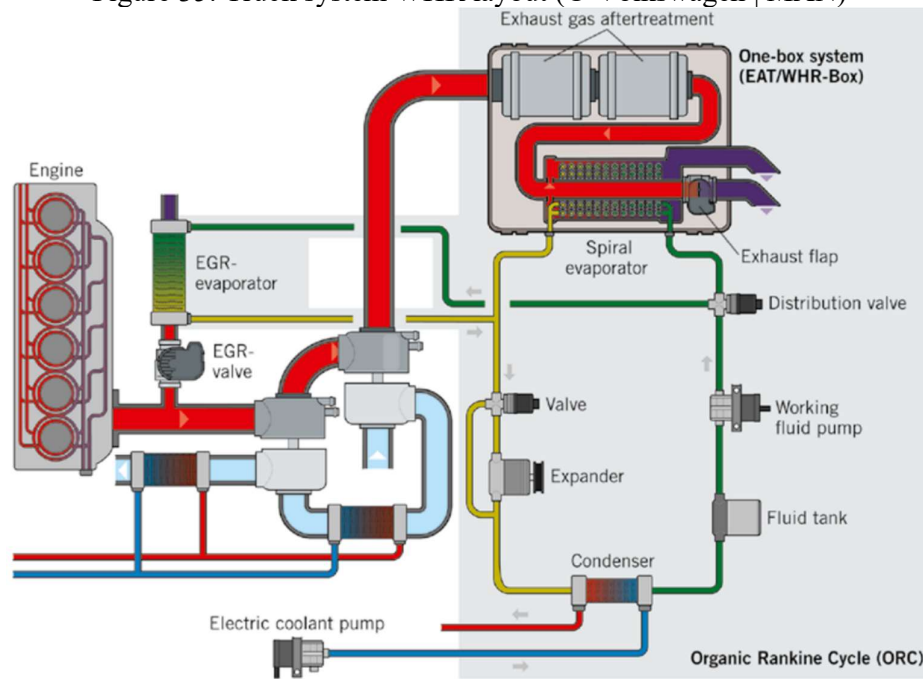


Source: Carstensen et al. (2019).

A replacement system integrated into the box was installed to make it possible to quickly replace the coils directly on the vehicle. This replacement system means that various evaporators can be inserted into the box depending on the engine variant and the available exhaust gas options.

In addition to the main exhaust gas stream, the diesel engine in the demo truck has an cooler for Exhaust Gas Recirculation (EGR) system. Using this heat source for the WHR system makes sense as well (Figure 35).

Figure 35: Truck system WHR layout (© Volkswagen | MAN)



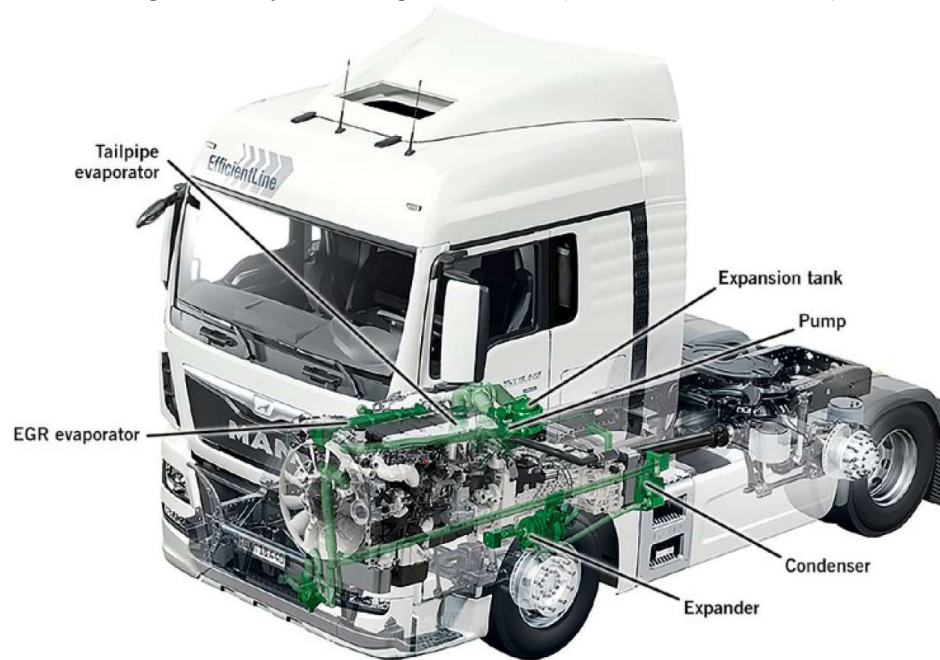
Source: Carstensen et al. (2019).

The system layout with two heat sources requires the ORC working fluid to be distributed between both evaporators with a logic system. The vapor temperature for both mass flow rates is controlled with the pump in combination with a distributor valve. The fluid pump increases the pressure of the liquid to a range between 10 and 30 bar as a function of the operating point. The ethanol evaporates and is heated up to 200 to 220 °C.

At a typical engine operating point, for a loaded commercial vehicle on a freeway, there will be approximately 500 kg/h of exhaust gas at temperatures of around 300 °C available downstream of the exhaust gas treatment system. A mass flow rate of approximately 200 kg/h, at a temperature of over 400 °C, flows through the EGR cooler. At the described operating point, approximately 50 kW of heat output can be transferred from these two heat sources to the WHR system. At high-load operating points, over 100 kW are possible.

See in Figure 36 that the expansion tank, filter, fluid pump, and distributor valve components are located behind the cab. The EGR evaporator replaces the existing EGR cooler, and the main exhaust gas stream evaporator is integrated into the exhaust gas treatment box.

Figure 36: System integration truck (© MAN Truck & Bus)



Source: Carstensen et al. (2019).

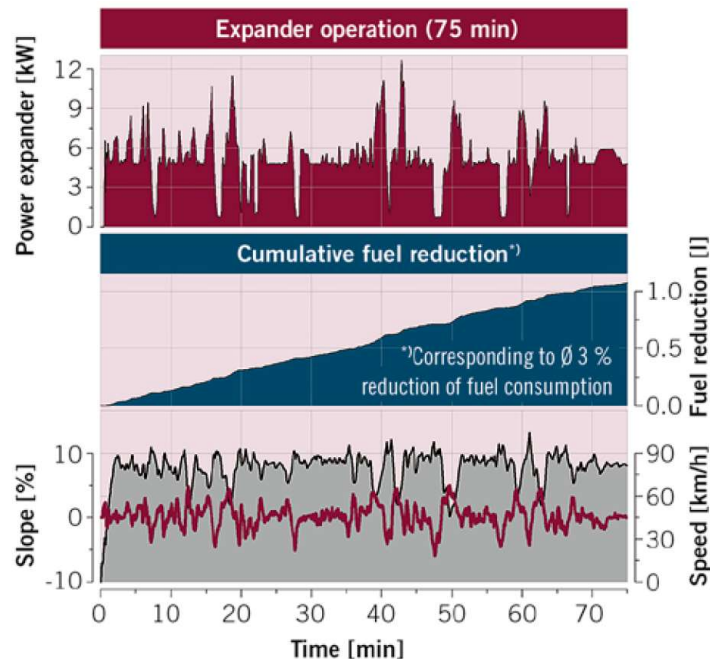
Turbo or positive displacement engines can be used as expansion machines. The former are suitable for coupling to a generator for electrifying the expander power; the latter are the preferred machines for directly transmitting the recovered power to the powertrain mechanically. The direct mechanical power coupling approach was chosen for the WHR truck. Due to the low electrical system demand of 800 to 1000 W in the WHR truck, pure electrical recovery was not an ideal choice. Future solutions with a greater degree of electrification – for example as a result of electrified ancillaries – or hybrid electric vehicles with the corresponding electrical infrastructure will make using the expander power in a purely electrical manner possible.

In order to directly transmit the recovered power mechanically in the WHR truck, an axial piston expander or a scroll expander is used. Both are characterized by a relatively high efficiency over a wide speed range. Within this context, the machine is coupled using a special front end accessory drive (spur gear system) that implements a gear ratio from 2:1 to 3:1 depending on the machine type. The bypass valve for connecting and disconnecting the expansion machine must be installed directly upstream of the expander for a number of reasons, including component protection. The condenser, implemented as a plate heat exchanger, is installed directly downstream of the expander in order to minimize the backpressure at the expander outlet.

The potential of a WHR system when it comes to reducing fuel consumption depends particularly on the vehicle usage profile and the ambient conditions. For example, on a hilly route and due to the thermal inertia of the WHR system, the energy of the WHR system heated uphill may not be available until going downhill and may then not be fully utilizable. The cooling reserve depends not only on the vehicle speed, but also significantly on ambient temperature. When testing the system's potential in the vehicle and on the engine dynamometer using various route profiles with various tonnages of 24, 32 and 40 t as well as the simulated ambient conditions summer and winter, fuel consumption reductions of up to 4 % were measured. The average fuel consumption savings for the demo truck in real-life use are approximately 3 %.

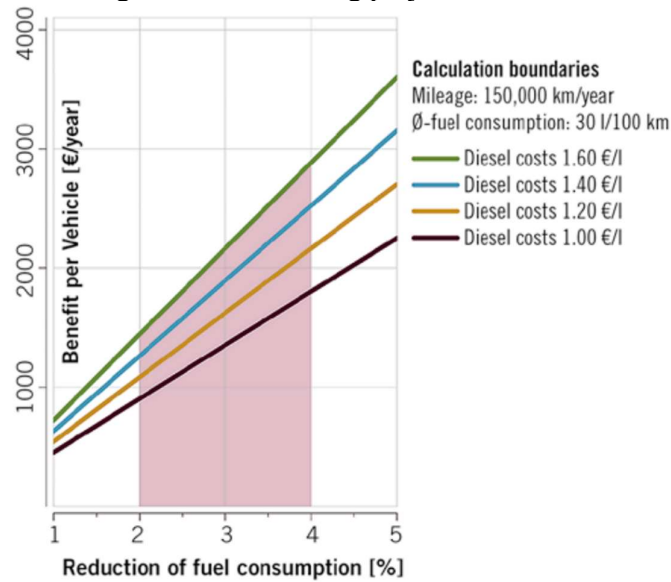
Figure 37 shows the operating parameters for the WHR system on a representative freeway drive. In addition, it shows the annual fuel cost savings resulting from the achieved fuel consumption reduction. The development goal is to amortize the system costs by using the WHR system within a maximum of two years. See fuel saving projection in Figure 38.

Figure 37: Expander energy production and fuel consumption reduction



Source: Carstensen et al. (2019).

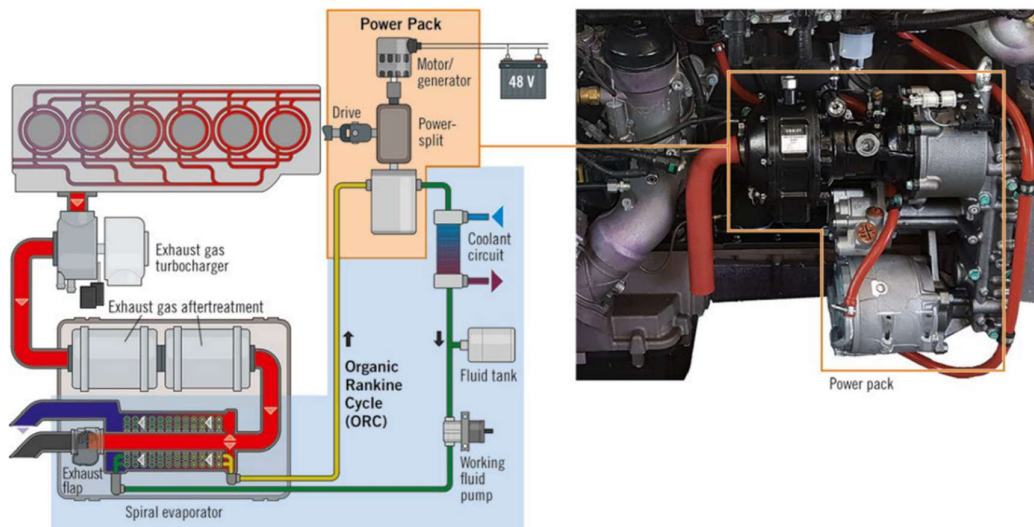
Figure 38: Fuel saving projection in Euro.



Source: Carstensen et al. (2019).

A measure for increasing the potential of the WHR system is to combine it with mechanical power recovery via a 48-V motor / generator that is installed on the same internal combustion engine front end accessory drive.

Figure 39: Combined electrical/mechanical power use (© MAN Truck &amp; Bus)



Source: Carstensen et al. (2019).

As shown in Figure 39 this arrangement makes the combined electrical/mechanical use of expander power possible. This means that the power can be transmitted directly to the powertrain with optimal efficiency or, optionally, fed into the electrical system via the 48-V motor/generator. Driving ancillaries and electric boosting are also possible. The alternating

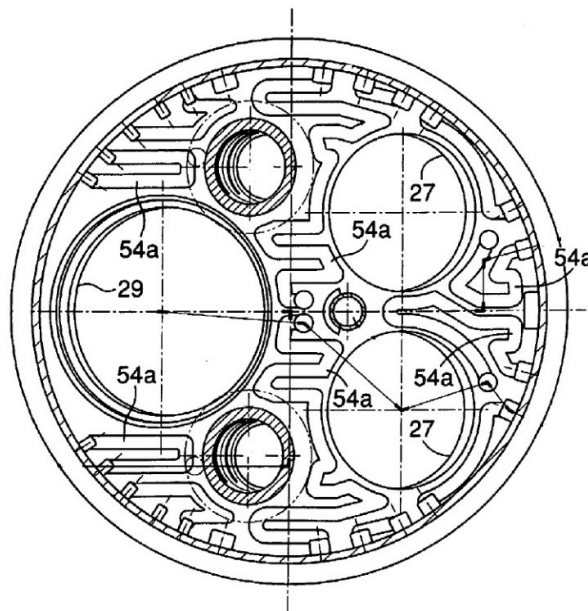
mechanical coupling of the internal combustion engine, WHR expansion machine, and 48-V motor/ generator means that the power flow between them is fully variable. Thus, for example, the maximum expander power can also be recovered regardless of the electrical load required. Excess power will flow directly into the powertrain. Likewise, electric power can be produced through direct coupling between the motor and the generator independently of the expander's operation.

#### 2.5.4 Super heater (boiler) using Coolant Heat rejection

The evaporative cooling system, described in section 2.3.2, is currently a potential alternative for steam generation with consequently applications to Rankine cycles. Researches as Honda (2007), MÜLLER, 2014, Cummins, 2014 and Ford (2017) shows proposals for integrating a Rankine system to the evaporative system. On each from their patent, the concept and ideas are clear, but its feasibility regarding power generation and costs are still not defined. It opens a wide opportunity for further studies.

The proposals are quite similar. The principle is introducing channels around hot spots on engine crankcase and cylinder head. One of the proposals from Honda, 2007 is shown in Figure 40.

Figure 40: Channel on Cylinder Head on above combustion chamber

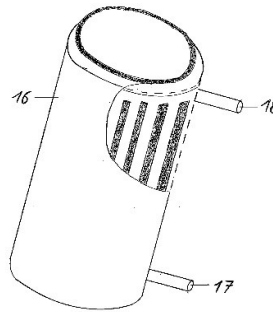


Source: Honda (2007).

The item 29 is the intake valve; 27 are the exhaust valves; 54a represents the channels purposed. All available places are fulfilled by channels, but the cylinder Head structure must be preserved. The labyrinth shape allows water to flow as the working fluid for the Purposed Rankini cycle.

Müller, 2014, with similar proposal, represents its ideas in his patent with sketches like shown in Figure 41.

Figure 41: Longitudinal Super Heater channels around combustion chamber

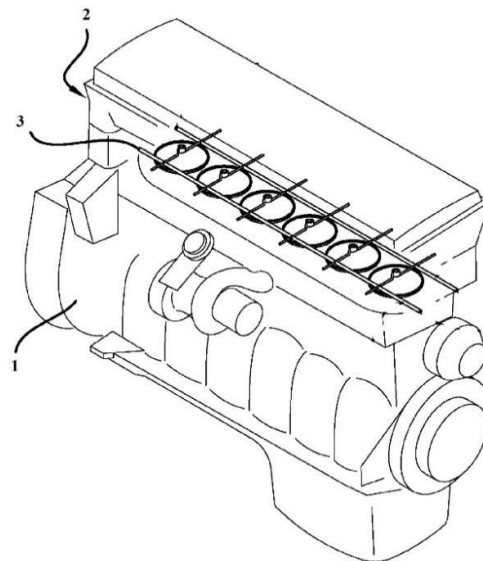


Source: Müller(2014).

The item 16 represents a “double layer” cylinder liner with inlet in 17 and outlet in 18. This proposal also suggests water as working fluid and do not performs deeper thermal analysis.

Cummins, 2014 supported by the companies Engine builders Cummins and Paccar also purposes channels inside engine structure to recovery heat. In Figure 42 part of the channels are showed.

Figure 42: Channel schema between Cylinder head and crankcase



Cummins et al. (2014).

The index 1 represents the engine crankcase, index 2, the Cylinder head and index 3 shows the right channel, which has a mirror channel on the engine left side. Additionally they are linked transversally by 6 auxiliary channels over the combustion chambers. No thermal simulations were performed.

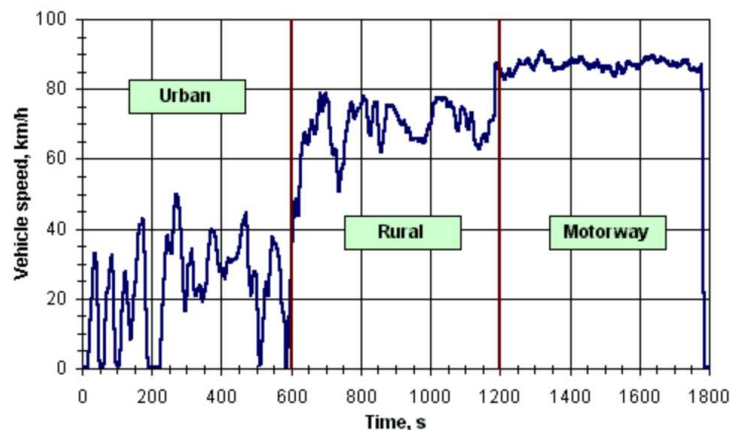
Ford (2017) also purposes the same way of recovering energy from engine coolant systems, but, as well as the other researches, focuses on the construction of its idea. Additionally, the working fluid flow control is also explained, but the main idea keeps the same without showing clear thermal results of the invention.

## 2.6 TEST BENCHES VALIDATION CYCLES

### 2.6.1 ETC (European Transient Cycle)

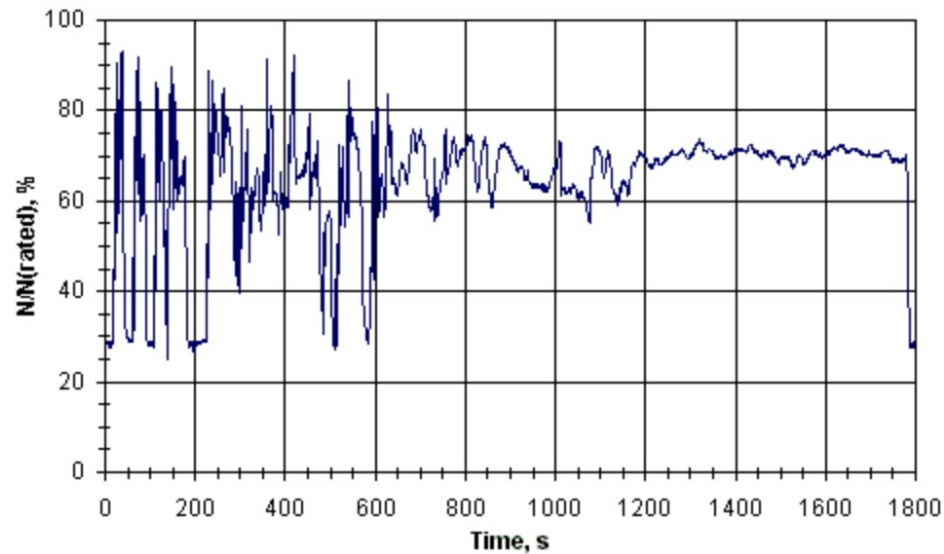
The ETC cycle has been developed by the former FIGE Institute, Aachen, Germany, based on real road cycle measurements of heavy duty vehicles (FIGE Report 104 05 316, January 1994). The final ETC cycle is a shortened and slightly modified version of the original FIGE proposal. Different driving conditions are represented by three parts of the ETC cycle, including urban, rural and motorway driving. The duration of the entire cycle is 1800s. The duration of each part is 600s. The part one represents city driving with a maximum speed of 50 km/h, frequent starts, stops, and idling; part two is rural driving starting with a steep acceleration segment. The average speed is about 72 km/h; part three is motorway driving with average speed of about 88 km/h (DIESELNET, 2019).

Figure 43: Transient Cycle – Vehicle Speed



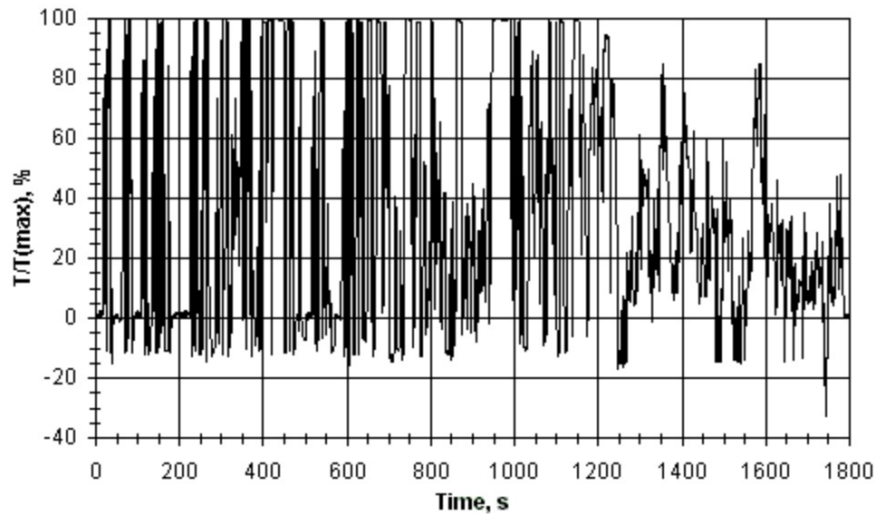
Source: DieselNet (2019).

Figure 44: Transient Cycle – Engine Speed



Source: DieselNet (2019).

Figure 45: Transient Cycle – Engine Torque



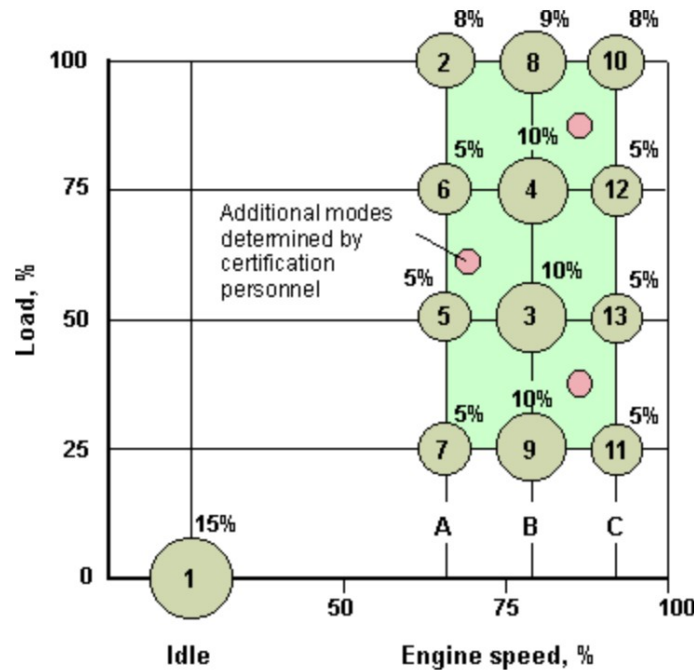
Source: DieselNet (2019).

The cycle was developed in two variants: as a chassis and an engine dynamometer test. Vehicle speed vs time over the duration of the cycle is shown in Figure 43. For the purpose of engine certification, the ETC cycle is performed on an engine dynamometer controlling engine speed (Figure 44) and engine torque (Figure 45) by percentage.

### 2.6.2 ESC (European Stationary Cycle)

The ESC test cycle was introduced together with the ETC (European Transient Cycle) and the ELR (European Load Response) tests by the Euro III emission regulation (Directive 1999/96/EC, effective year 2000) for emission measurement from heavy-duty diesel engines. The ESC is a 13-mode, steady-state procedure. Figure 46 shows the ESC test modes conditions. Except for idle, which lasts 4 minutes, all other 12 points lasts 2 minutes.

Figure 46: ESC test modes

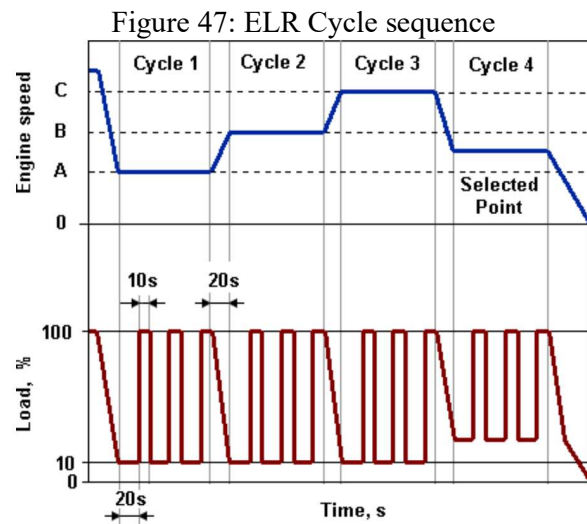


Source: DieselNet (2019).

The engine is tested on an engine dynamometer over a sequence of steady-state modes. The engine must be operated for the prescribed time in each mode, completing engine speed and load changes within 20 seconds. The specified speed shall be held to within  $\pm 50$  rpm and the specified torque shall be held to within  $\pm 2\%$  of the maximum torque at the test speed. Emissions are measured during each mode and averaged over the cycle using a set of weighting factors. Particulate matter emissions are sampled on one filter over the 13 modes. The final emission results are expressed in g/kWh (DIESELNET, 2019).

### 2.6.3 ELR (Engine Load Response)

The European Load Response (ELR) engine test has been introduced for the purpose of smoke opacity measurement from heavy-duty diesel engines. The test consists of a sequence of three load steps at each of the three engine speeds A (cycle 1), B (cycle 2) and C (cycle 3), followed by cycle 4 at a speed between speed A and speed C and a load between 10% and 100%, selected by the certification personnel. Speeds A, B, and C are defined in the ESC cycle. The dynamometer operation on the test engine is shown in Figure 47 (DIESELNET, 2019).



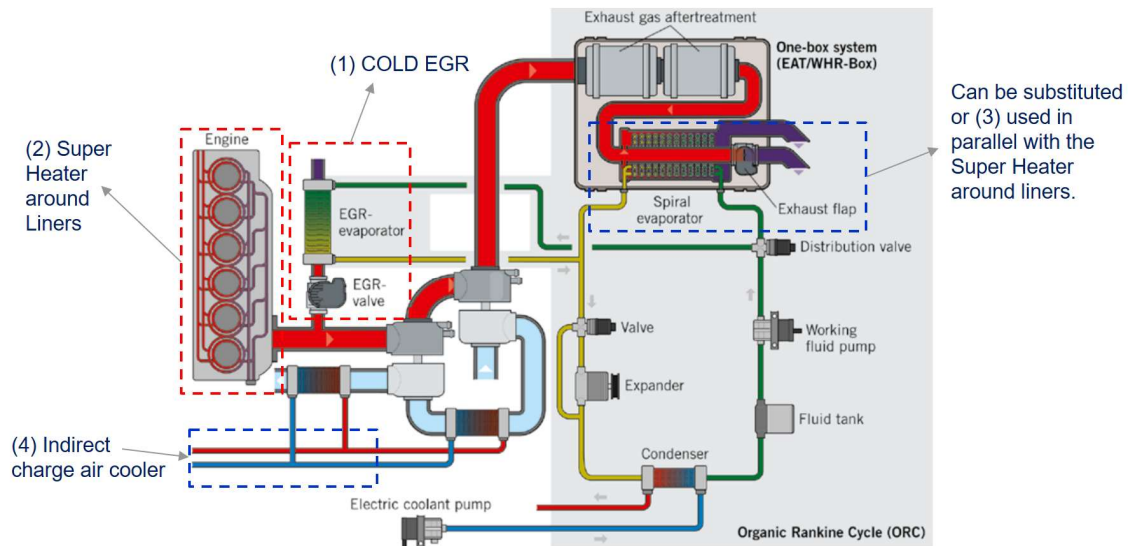
Source: DieselNet (2019).

Smoke measurement values are continuously sampled during the ELR test with a frequency of at least 20 Hz. The smoke traces are then analyzed to determine the final smoke values by calculation. The final smoke value is determined as a weighted average from the mean values at speeds A, B and C (DIESELNET, 2019).

### 3 MATERIALS AND METHODS

The innovative proposals represent possibilities of upgrading the energy use on the existing prototype described in section 2.5.3. In Figure 48, the first two steps represents the innovations and the third and fourth represents the integration possibilities.

Figure 48: Integration of the ideas with an existing prototype



Source: Adapted from Carstensen et al. (2019).

- (1) Cold EGR system;
- (2) Super heater around liners;
- (3) Integration of liners super heater with the current exhaust super heater;
- (4) Indirect charge air cooler.

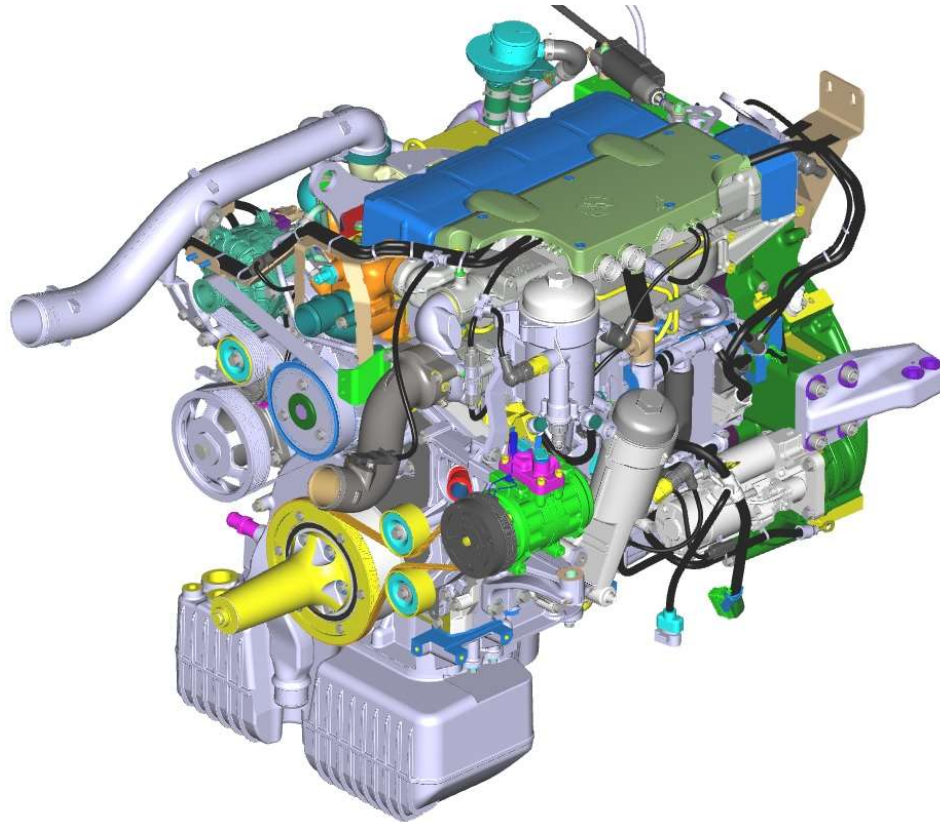
#### 3.1 ENGINE BASELINE FOR STUDY

The engine studied is a 4.5 liter four-stroke, turbocharged, water-cooled, reciprocating four-cylinder diesel engine. It uses overhead valves and a common rail with 1800 bar injection pressure for compression ignition. See Figure 49.

The choice of a four-cylinder engine for this study was due to the high load factor to which this hardware is subjected. As it leads to a high amount of heat rejection, in terms of percentage, more energy recovery is expected to be achieved. In addition, the choice is also

strategically better, as the low packaging available will protect the design of larger engines. Baseline data for this study were obtained from an objective ETC measurement.

Figure 49: Engine baseline for study



Source: My own authorship

An extensive literature review was conducted to map current research on waste heat recovery systems. Some of the connoisseurs of such vendor owners as Cummins, Bosh and Borg Warner were also contacted to grow this portfolio, but the key features of their technologies were kept as internal developments.

The review also features articles on other types of heat recovery, such as absorption, and Peltier - seebeck. Energy sources are explored with reference to boilers and combustion engine heat balance. To use this energy, the conventional and organic Rankine cycles were researched and, to recovery it, the different types of steam turbines were also mentioned. As research has taken combustion chamber temperature as an important boundary condition, heat transfer flux should also be explored, as well as material behavior due to thermal fatigue.

Based on an actual engine measurement submitted for an ETC cycle, it can be seen that a large amount of energy is lost in the cooling process. As soon as a conventional Rankine Cycle shows efficiency of about 30%, and assuming to recover 10% of the energy lost on the

cooling system, the heat balance for the actual proposal is capable of recovering up to 10 kW of liquid power.

The development of this energy recovery system is supported by a well-known commercial vehicle OEM in Brazil. A mathematical analysis has been performed using GT Power software (see section 3.4.2), enabling simulations of different types of channel profiles and dispositions around the combustion chamber. For the EGR system without EGR cooler, a complete ETC cycle was calculated on Excel and VB and are detailed in section 3.3 and APENDIX 1.

It was possible to demonstrate that the use of a supersonic Laval nozzle would be a feasible solution for mixing gases from low pressure line to the high-pressure line on intake clean air flow. The mathematical simulation and calculation provide enough data to build a future prototype that could run into internal test benches or in vehicles from this OEM.

The energy recovered using this system, converted into reduced fuel consumption, needs to outweigh the investments to be worth the customer's TCO (Total Cost of Ownership).

All theoretical studies were done using a 4.5-liter engine, but the potential for a real application is higher in a 13-liter diesel engine capable of delivering 440 HP of net power.

### 3.2 ENGINE BASELINE POWER CONSUMPTION

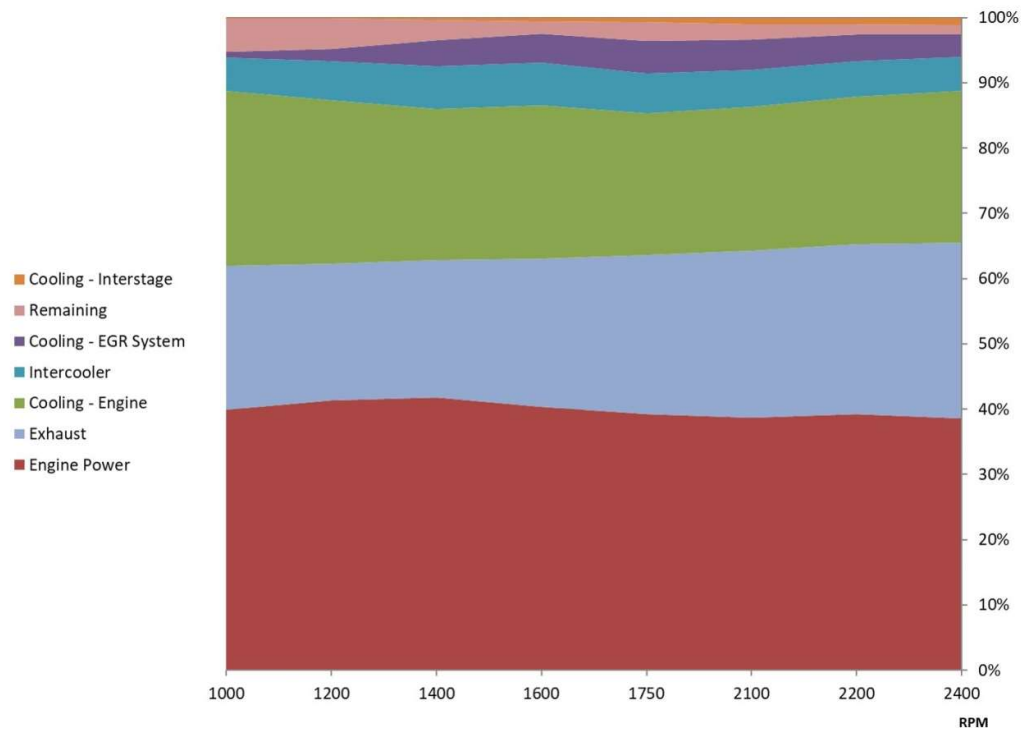
Internal combustion engines are low-efficiency systems. A lot of heat is wasted on the conversion of chemical energy into mechanical energy.

This work shows the feasibility of recovering lost energy from other energy sources: The engine cooling system (coolant). It is little explored today by the scientific community.

Several researches are being conducted for the use of Rankine cycle to recover energy from the exhaust system, as shown in section 2.5. However, an important source of heat has been little explored by the scientific community: The heat rejected by the engine cooling system.

The heat rejection of a diesel engine with gas recirculation system is classified by percentage and shown in Figure 50.

Figure 50: Energy consumption in a diesel engine by percentage



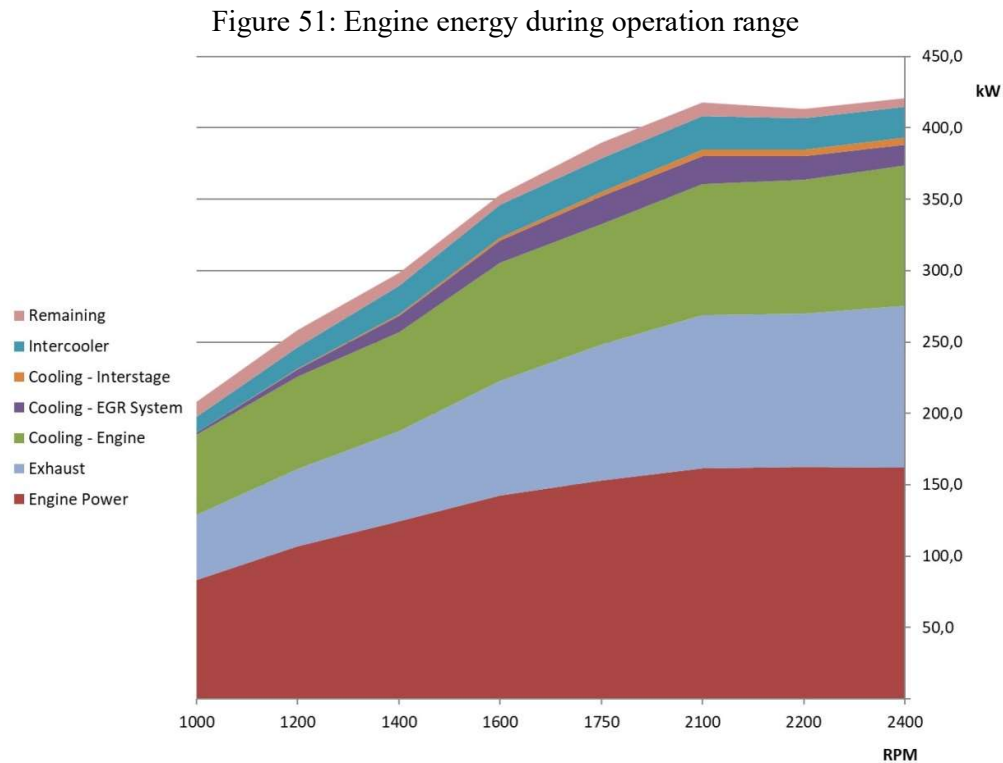
Source: My own authorship

Notice that the amount of energy is not shown, but its contribution to the expenses on all engine range. In “X” axis the engine rotation is represented and in “Y” axis the contribution of each element in percentage. As this engine uses a dual stage turbocharger, there is a cooler between the compressors, which allows the second stage to receive high pressure gases with lower temperature. It avoids the overheats on the second stage. The energy consumption from this cooler is represented on the first item: Interstage. The remaining energy (second line) represents friction and convection losses that are not represented in the other lines. The EGR system represents the energy consumption from the EGR cooler (aimed to be removed); The intercooler represents the energy consumption to decrease the temperature from the turbocharger to the engine intake manifold by using the CAC. The Engine line represents the energy loss to the environment through the cooling system. Exhaust represents the energy loss on the gases expelled from the engine. Notice that there are three lines marked with the term Cooling: Interstage, EGR system and Engine. The use of those three portions for WHR is one of the stated objectives.

The most common waste heat recovery studies are related to the exhaust gases. For example, Borg warner emission systems (2014) used an organic Rankine cycle with tail pipes and engine recirculation systems as the main power source. Nelson (2008) presented a complete

study in the 2008 DEER (Diesel Engine Efficiency Emissions Research), demonstrating through test benches analysis that it would be possible to decrease fuel consumption by approximately 10% using an organic Rankine cycle to recover energy from exhaust gas streams. However, according to the current EURO 5-CONAMAP7 legislation, resolution 403/2008, and the new EURO 6 legislation (current EUROPE), the opportunity to use this energy is diminishing because post-treatment emission processes require high temperatures to allow chemical reactions to decrease emission levels.

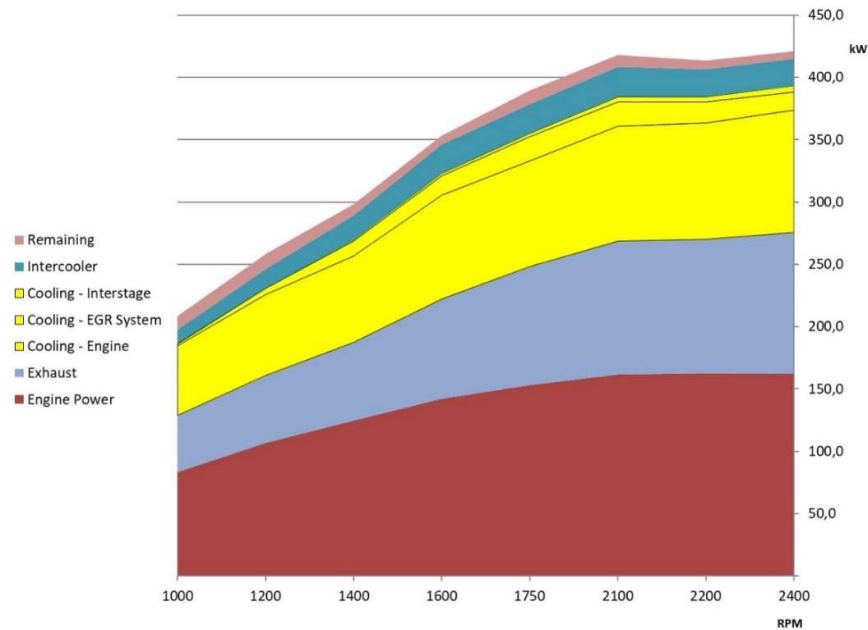
Figure 51 organizes the engine distribution, showing the contribution of the lost energy in proportion to the engine speed.



Source: My own authorship

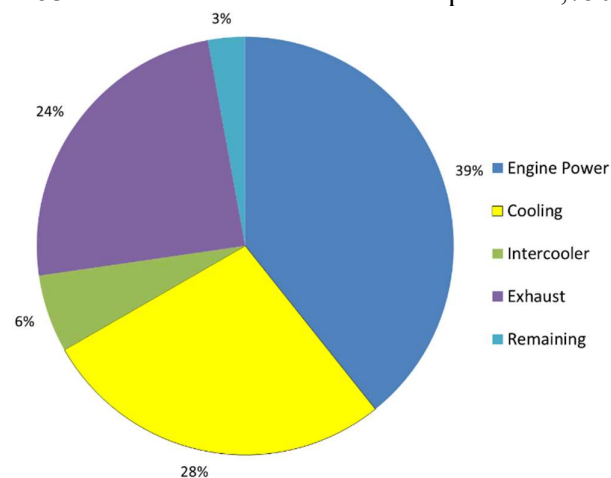
Notice that the fuel combustion increases proportionally with the engine rotation considering the same load through all engine range. This curve, differently from Figure 50 shows the real amount of energy spent on the Engine systems. In Figure 52, all cooling losses contributions were grouped.

Figure 52: Engine energy during operation range grouping coolant contribution



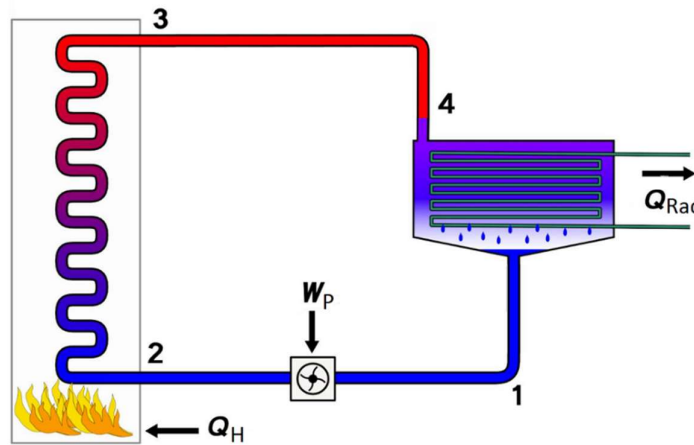
As soon as the Interstage cooler, EGR cooler and Engine cooling system will be used together for the purposed WHR system, this curve represents all cooling contributions grouped and give a qualitative impression of the energy losses only decrease temperatures on the system. The EGR cooler rejects heat to the engine main cooling system, but all measurements are done individually in each equipment. Therefore, the sum of those energies results on the total amount of energy lost for the environment via heat exchangers (radiator + CAC). For each rotation, there is an specific contribution of this energy consumption. By choosing the cruising speed of 1,750 rpm with full load the engine power distribution can be observed shown in Figure 53.

Figure 53: Power Distribution on cruise speed of 1,750 rpm



Notice that 28% from the power generated by the fuel combustion is rejected to the environment via heat exchangers (radiator and CAC), that are represented in Figure 54 by the portion between 4 and 1. The coolant has its temperature dropped down with the removal of  $Q_{\text{Rad}}$ . After 1, the water pump send the “water” to the engines, which heats it again.

Figure 54: Cooling cycle



Adaptation from Ploucha (2019).

Where:

$Q_{\text{Rad}}$  = Energy rejection to environment (27% of the fuel power generation) = 105 kW

105 kW is the result of the calculation based on the input and output temperatures of the Charge Air Cooler (CAC), the cooling heat rejection to the environment corresponds to exactly this value. For the current situation, the ratio  $Q = m \cdot c \cdot \Delta T$  is determined from:

$$Q = Q_H = 105 \text{ kW}$$

$$m = 270 \text{ kg of coolant per minute}$$

$$c = 3.74 \text{ kJ/kg.K (From coolant at working temperature – data provided by supplier)}$$

$$\Delta T = 6.2 \text{ K}$$

This power recovery is barely explored by the scientific community. Even though the temperature drop is low (6.2 K), there is a good opportunity of recovering an important part of it by using a Rankine cycle (section 2.4).

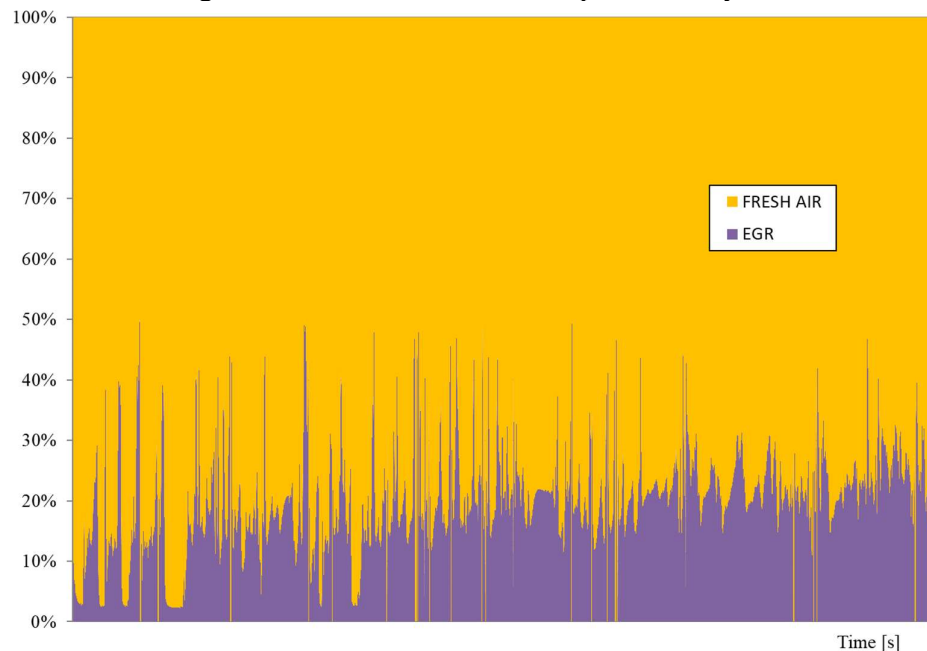


mixing cooled EGR with intake fresh air;  $T_3/P_3$  = temperature and pressure from the exhaust gases;  $T_4\_NSD/P_4NSD$  = Tail pipe temperature and pressure.

Conventional EGR power consumption can be calculated using the exhaust gas parameters combined with the measured temperatures  $T\_EGR\_VK$  and  $T\_EGR\_NK$ . At 1,750 rpm, the engine uses up 19.5 kW to cool down the exhaust before returning to the intake line. As described in section 2.2.2, the HP-EGR drains highly pressurized exhaust before the turbocharger. This deviation itself also causes misuse of energy, considering that the gases that could drive the turbocharger are used to return to the intake line. This side effect is not considered in the simulations and would mean a complete redesign of the turbocharger.

As described in section 2.2.2, EGR systems are used to lower the combustion temperature at specific engine loads, and consequently NOx emissions. Figure 56 shows the EGR demand compared to the engine fresh air intake.

Figure 56: EGR demand in a complete ETC cycle

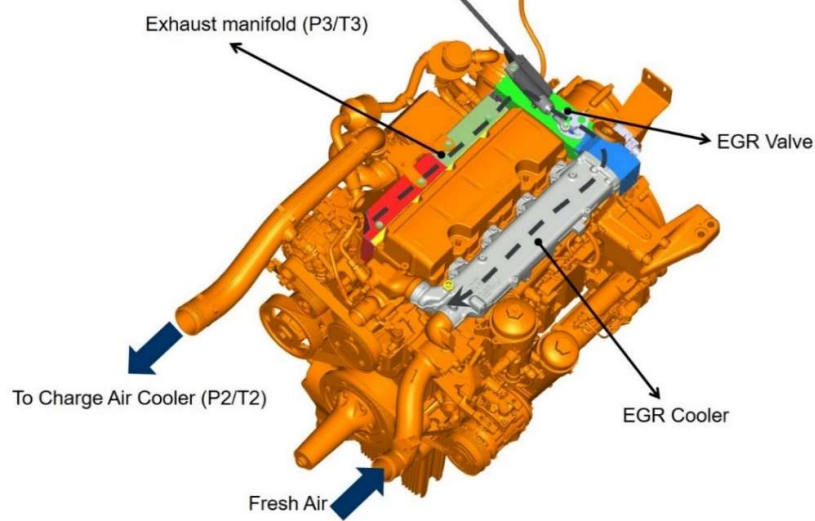


Source: My own authorship

Note that EGR demand peaks at 50% of full intake air, but varies significantly over engine working cycle. The average EGR rate is 25% from the intake (fresh air + EGR).

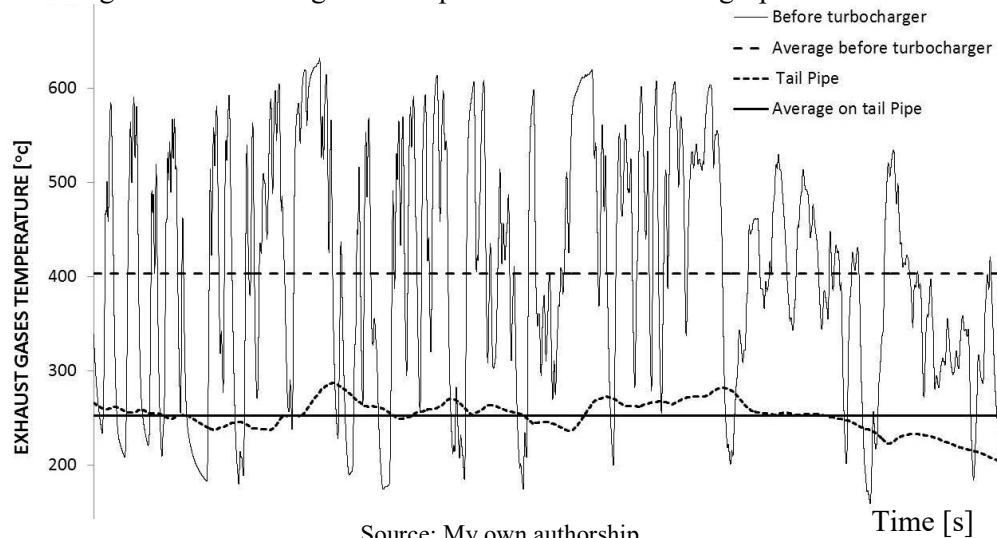
The baseline engine EGR system (Figure 57) uses the inert gases from the exhaust manifold before the turbocharger. This point is directly connected to the intake manifold and on the way, there is a control valve and a heat exchanger that lowers the exhaust temperature before mixing it with the fresh air.

Figure 57: EGR mechanism from baseline engine



Source: My own authorship

Figure 58: Exhaust gases Temperature on low- and high-pressure lines



Source: My own authorship

Based on the data in Figure 58, it is possible to calculate that the average temperature before the turbocharger is 403 °C, reaching peaks greater than 600 °C. Since energy is necessary to cool these gases, an obvious solution would be to capture the exhaust gases from a lower temperature point. In a combustion engine vehicle, this point is known as tail pipe, which, in this case, reaches temperatures below 300 °C, maintaining an average of 253 °C, as shown in Figure 58.

Mixing different pressure line flows has always been a challenge for engineers. The EGR system is especially complex because of the energy waste involved. In a conventional 4-

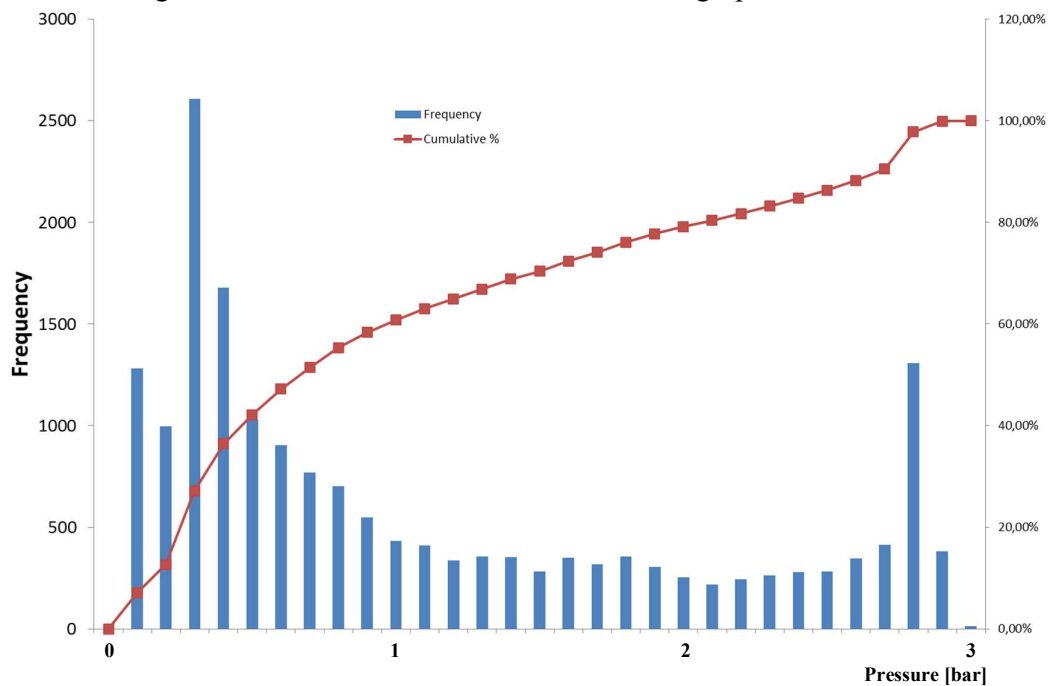
cylinder diesel engine, decreasing the exhaust gas temperature from 600 °C to 100 °C can mean a significant loss of energy. Some concepts have been developed to lower the inlet temperature, but most of them stop in the presence of particles on the turbocharger compressor wheels. See LP-EGR in section 2.2.3.

The present work proposes a high flow variable valve to mix exhaust gas from vehicle tail pipe (lowest possible temperature) with pressurized fresh air in the intake line (after turbocharger) before or after Charge Air Cooler (CAC).

The removal of inert gases from the tail pipe would directly mean a 40% decrease in heat rejection from the EGR system (403 °C compared to 253 °C). Even the estimation of reduced fuel consumption is not simple, if preventing 6kW heat rejection could be converted directly to useful energy, that would mean 3.5% of the 164kW power of the engine under study.

Unfortunately, removing inert gases from the tail pipe is not an easy task. In section 2.2.3, low pressure EGR was presented as a possible solution, but particulate contact with the turbocharger compressor wheel is not desirable. The technical requirements for durable systems are very expensive and it is not worth removing the EGR cooler from the system. Therefore, the proposal of this work is to take inert gases from the tail pipe and mix it with fresh air after the turbocharger compressor.

Figure 59:  $\Delta P$  distribution between low- and high-pressure lines.

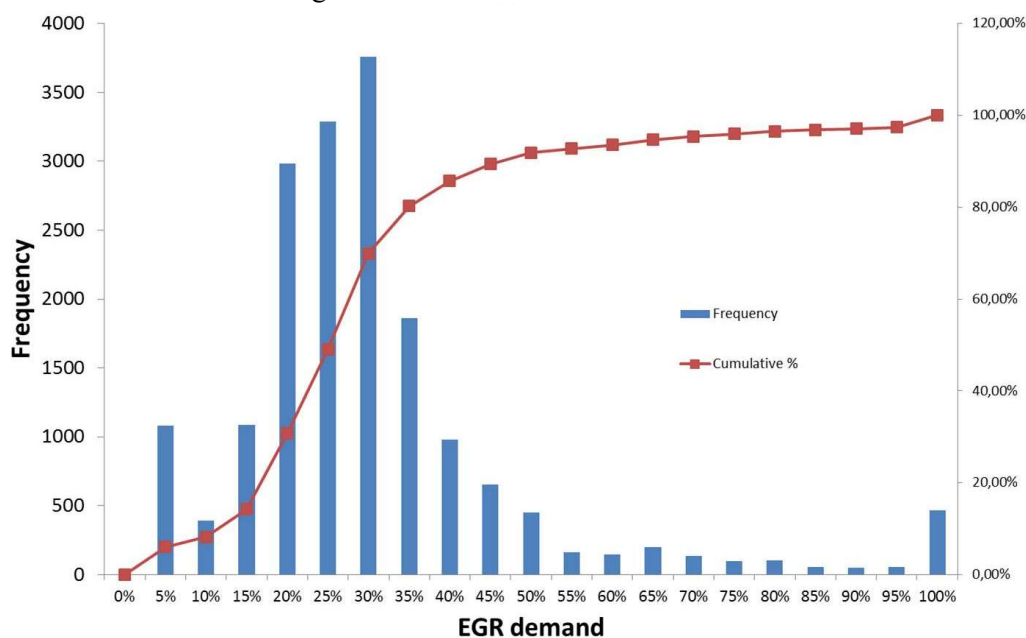


Source: My own authorship

In Figure 59, it is possible to understand how different pressures are from the tail pipe line and the fresh air line after turbocharger compression. Note that over 60% of the  $\Delta P$  measured in this ETC cycle is under 1 bar. The remaining measured points work at a difference of 2bar or 3bar, 20% each.

In “x” axis it is possible to observe different  $\Delta P$  ranges, showing its distribution on the ETC cycle run. Notice that the map isn’t a normal distribution, showing highest frequency on values under 1,5 bar. It means that 75% of the occurrence. The remaining 25% of the occurrences are located over 1,5 bar with a peak frequency at 2,8 bar.

Figure 60: EGR demand distribution



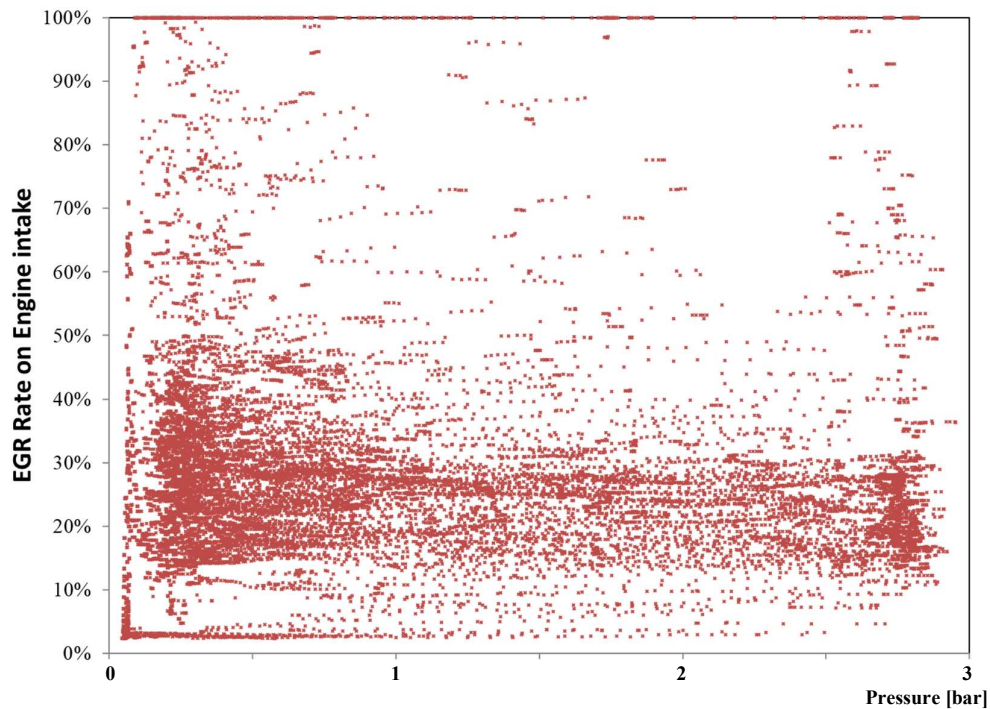
Source: My own authorship

While  $\Delta P$  is not constant, so is the demand for EGR. In Figure 60, 100% EGR demand means 30% of full engine input (fresh air + recirculated gases).

The EGR demand has a normal distribution floating around 25%. Notice on the cumulative curve that circa of 96% from EGR use are not over 50% of the flow and 70% do not overtake the limit of 30% of EGR demand. The values showed under 5% and over 55% EGR demand are considered measurement errors during transient conditions and are not considered as real demands.

The combination of  $\Delta P$  and EGR demand is shown in Figure 61. A mechanism for mixing these pressure lines must be developed to mix the EGR demand.

Figure 61: EGR rate and  $\Delta P$  between tail pipe and charged fresh air.



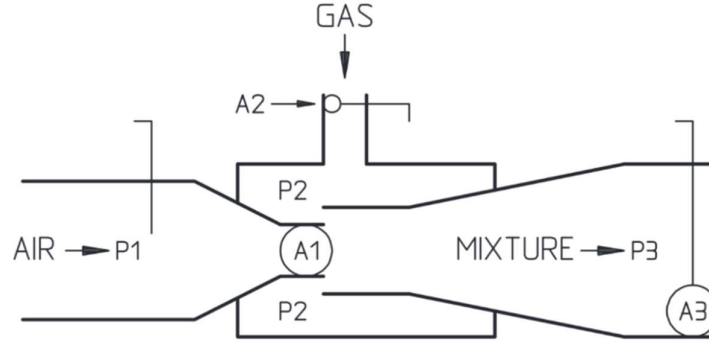
Source: My own authorship

The Figure 61 is a mixture from the data showed in Figure 59 and Figure 60. Each point represents one of the 18000 measured points. In “y” axis the EGR rate is shown and in “x” the  $\Delta P$  between intake fresh air line and tail pipe line are shown. It is possible to see part of the map with values over 50%, which represents 8,5% of the measured points. For the purposed action (Cold EGR system without cooler), two main regions are defined in this curve: 1 – until  $\Delta P = 1$ , where EGR demands are relevant until 50% and 2 – between  $\Delta P = 1$  and  $\Delta P = 3$ , where EGR demands are relevant until 30% of mixture. Measurements over those limits were taken out of the simulation to be considered errors (see Figure 60 for pressure distribution).

### 3.3.1 Conventional VENTURI concept

A conventional VENTURI system uses the Bernoulli principle to increase flow velocity in order to decrease flow pressure and consequently mix Low Pressure (LP) with High Pressure (HP) lines.

Figure 62: Conventional Venturi concept



Source: Esapyronics (2019).

Based on the Bernoulli principle, White (2018) shows the calculation of the delta point pressure accessible by point in equation 5:

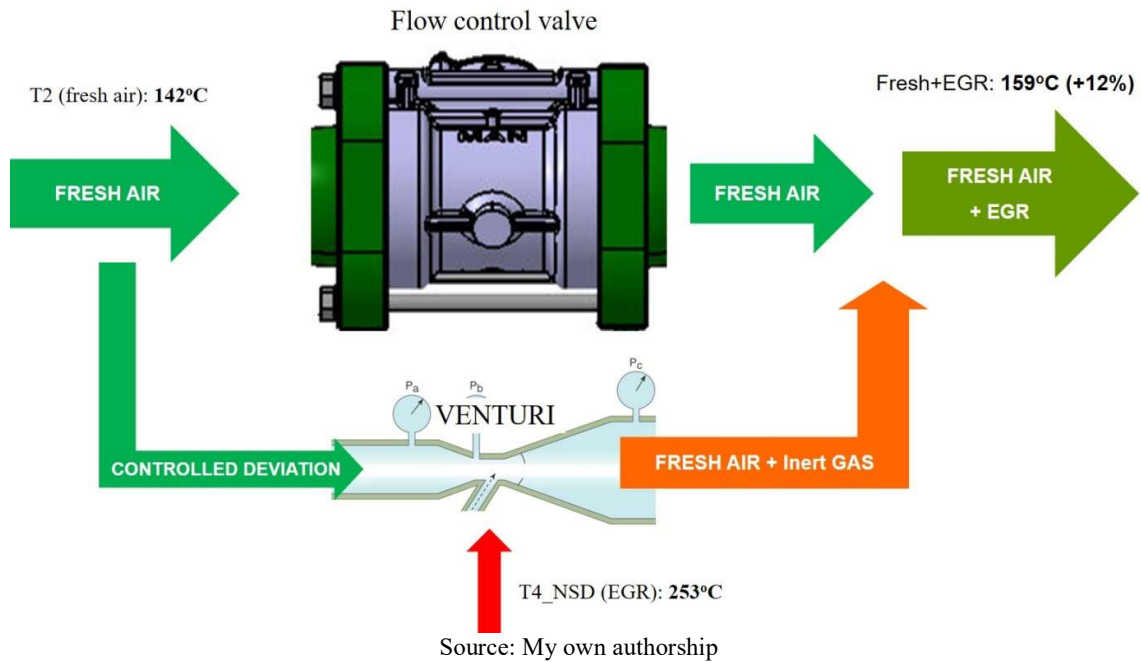
$$(p_1 - p_2) = \frac{\rho}{2} \left( 1 - \left( \frac{A_2}{A_1} \right)^2 \right) \left( \frac{Q}{A_2} \right)^2 \quad (1)$$

As can be seen in Figure 61, EGR demand is not necessarily coincident with high  $\Delta P$  demands. This means that there are high EGR demands for low  $\Delta P$  and vice versa. Therefore, determining the throat required to achieve all pressure drops combined with mass flow, temperature and pressure is no simple task.

The required 81.7% reduction in area leads to the conclusion that a conventional inlet line installation would cause unnecessary pressure drops in the air inlet line throughout the engine map.

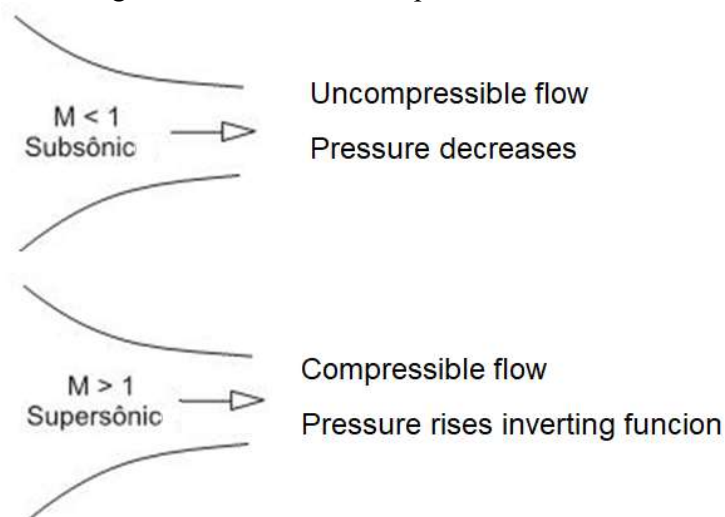
To avoid this condition, the idea would be to create a “variable Venturi” concept, as shown in Figure 63. There would be a control valve in the fresh air track, which controls the back pressure in order to derivate flow to the line where the Venturi is located. With this action, when EGR is not required, the control valve would open and fresh air would flow without restriction.

Figure 63: Variable Venturi



Unfortunately, during the development of this concept, a flaw was found: the compressible flow could not be modelled with Bernoulli equations. Incropera (2006) shows that the Venturi equations are representative only for incompressible flow. The identification of this transition is determined by the flow velocity. In Figure 64 both situations are shown.

Figure 64: Subsonic and supersonic flow behavior



Source: Adapted from Incropera (2006).

In other words, even if there is a flow deviation to avoid unnecessary pressure drop, there will still be conditions where the required flow velocity in the Venturi exceeds the limit of incompressible flow modeling.

Using the reduction area obtained, mass flow was converted into flow velocity and compared with the speed of sound. Due to the sensitivity of the calculation, all present parameters were considered and the speed of sound was also calculated for each measurement point based on fresh air temperature and density. Speed of sound is determined by equation 6 and can be compared with the flow speed to determine Mach.

$$a_1 = (kRT_1)^{\frac{1}{2}} \quad (2)$$

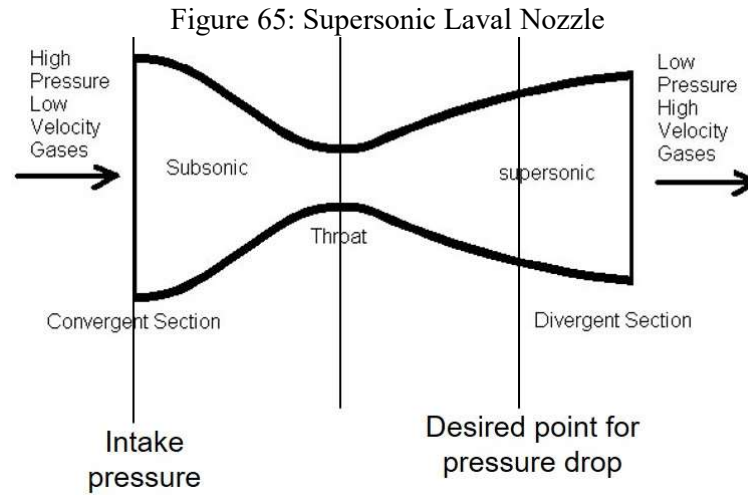
Considering aerodynamics, White, 2018 shows some 5 ranges that can classify Mach:

|                  |                                                                          |
|------------------|--------------------------------------------------------------------------|
| $Ma < 0,3$       | Non-compressible flow: Density effects are negligible                    |
| $0,3 < Ma < 0,8$ | Subsonic flow: Considers density effects, but no shock waves are present |
| $0,8 < Ma < 1,2$ | Transonic flow. Difficult region for modeling due to mix behavior        |
| $1,2 < Ma < 3,0$ | Supersonic flow. Subsonic regions are not present                        |
| $3,0 < Ma$       | Hypersonic flow. Shock waves and flow variations are strong              |

For internal flow, the most important question is whether Mach is above or below 1.

### 3.3.2 Application of a Laval nozzle

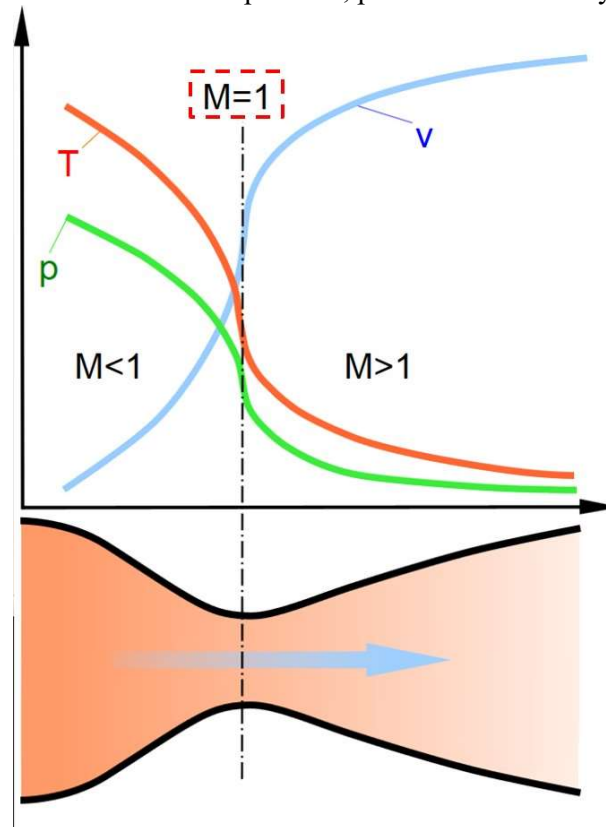
A Laval nozzle consists of a system that reduces pressure and increases flow velocity. This system is commonly used on aircraft jet engines to increase speed. When fluid initiates a sonic flow in the throat, the pressure drop appears just after the throat with a supersonic flow. The behavior of this system is the opposite of Figure 64.



Source: IOK (2018).

This system is able to decrease temperature and pressure by increasing flow velocity just past the throat, as shown in Figure 66.

Figure 66: Laval nozzle temperature, pressure and velocity behavior.



Source: IOK (2018).

To calculate the pressure drop for a specific fluid flow, temperature, pressure and flow must be available. It is demonstrated that to achieve a pressure drop after the throat, a sonic flow must be present in the throat. This means that the speed in the throat reaches the speed of sound. Equation 7 relates the areas at different positions of the nozzle and (A) to the area of the throat ( $A^*$ ) that is not yet known (WHITE, 2018):

$$\frac{A}{A^*} = \frac{(1 + 0.2Ma_1^2)^3}{1.728Ma_1} \quad (3)$$

Just as fluid can be modeled by a subsonic flow for a conventional Venturi, the Laval nozzle also has its limits. An important one is the blocking flow, which means there is a maximum flow physically allowed through the throat.

Stagnation temperature ( $T_0$ ) is shown in equation 8:

$$T_0 = T_1 + \frac{v_1^2}{2c_p} \quad (4)$$

Where  $T_1$  is the inlet temperature and  $V_1$  is the inlet velocity.

To obtain stagnation pressure (equation 9), Mach needs also to be calculated.

$$p_0 = p_1(1 + 0.2Ma_1^2)^{3.5} \quad (5)$$

Where  $p_1$  is the inlet pressure and Ma is Mach (flow speed compared with local sound speed)

The blocking flow is given by equation 10:

$$\dot{m} = 0.6847 \frac{p_0 A^*}{\sqrt{RT_0}} \quad (6)$$

Assuming a control valve installed as in Figure 67, there will be an interface to communicate the low pressurized fresh air with the inert gases that will be admitted from the Tail pipe line, which is under atmosphere pressure. The Laval Nozzle cross section will need to count on exhaust inlet channels in the position indicated as the desired pressure drop point

in Figure 65. This point is used on the simulations on section 4.1 and has direct impact on the system performance to reach the desired pressure drop.

Figure 67: Variable supersonic Laval nozzle

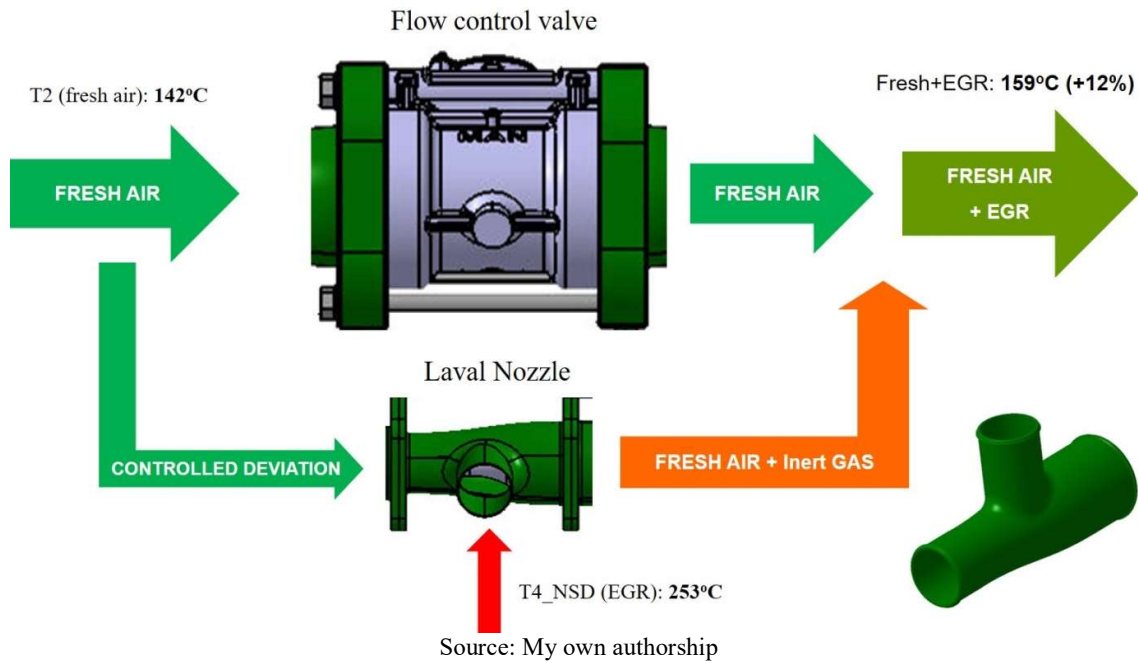
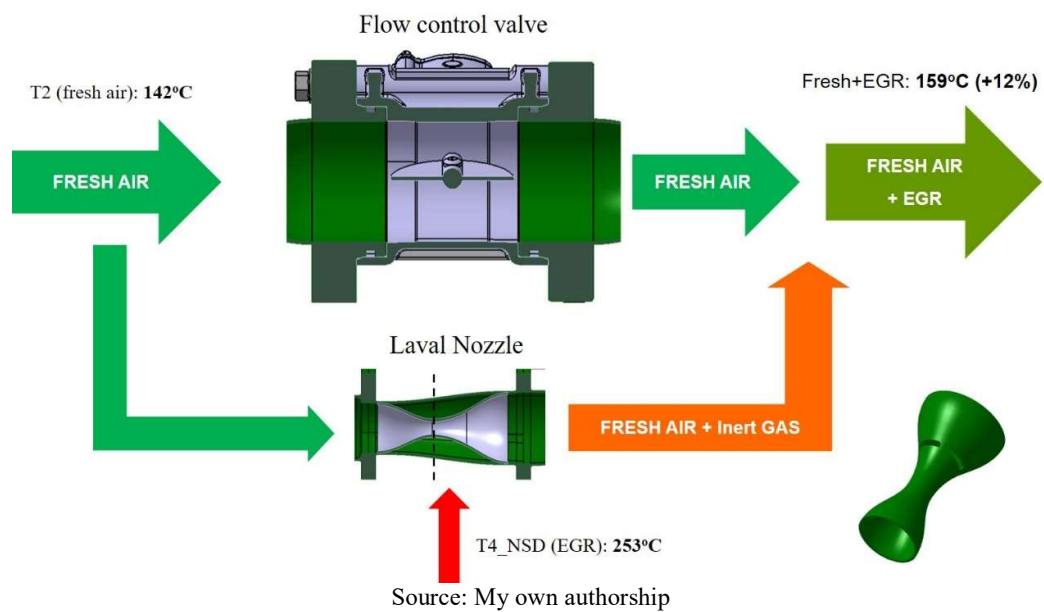


Figure 68: Variable supersonic Laval nozzle cross section



The main difference between Figure 67 and Figure 68 are the views without and with a cross section respectively. Additionally, on the last, it is possible to see the channels that are planned on the Laval Nozzle profile to allow the inert gases to be introduced on the inlet flow. The former image shows the housing designed to make the interface between the fresh air and exhaust gases from the tail pipe.

A controlled fresh air and the Laval Nozzle could always be configured to receive blockage flow to ensure sonic throat and supersonic flow of pressure drop after the throat. Once the  $A/A^*$  ratio is known, equation 11 shows how Mach can be determined.

$$\left\{ \begin{array}{ll} \frac{1+0.27(A/A^*)^{-2}}{1.728A/A^*} \quad (a) & 1.34 < \frac{A}{A^*} < \infty \\ 1 - 0.88 \left( \ln \frac{A}{A^*} \right)^{0.45} \quad (b) & 1.0 < \frac{A}{A^*} < 1.34 \end{array} \right\} \text{subsonic flow} \quad (7)$$

$$\left\{ \begin{array}{ll} 1 + 1.2 \left( \frac{A}{A^*} - 1 \right)^{1/2} \quad (c) & 1 < \frac{A}{A^*} < 2.9 \\ \left[ 216 \frac{A}{A^*} - 254 \left( \frac{A}{A^*} \right)^{2/3} \right]^{1/5} \quad (d) & 2.9 < \frac{A}{A^*} < \infty \end{array} \right\} \text{supersonic flow}$$

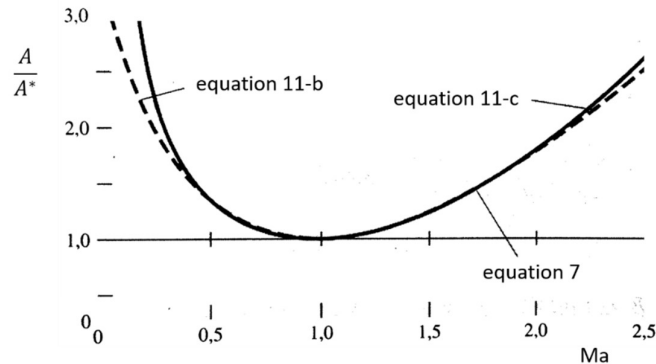
Mach (Ma) at the outlet can be calculated by any diameters and distances in “x”, but in this case, 25mm was assumed as the pressure drop collection point. Therefore, the relationship between the sonic throat and output areas is 2.16, which leads to the Mach calculation with equation 11- c and points to Mach = 2.39 (supersonic flow).

Using the obtained Mach, the pressure drop can be calculated with equation 12, which is a rearranged version of the stagnation pressure (equation 9), but now, assuming it is known and the outlet pressure to be calculated:

$$p_2 = \frac{p_0}{(1+0.2Ma_1^2)^{3.5}} \quad (8)$$

The relation between the throat and other nozzle inlet diameters is shown in Figure 69.

Figure 69: Relation between areas compared with Mach for isentropic flow



Source: Adapted from White (2018).

In APENDIX 1 it is possible to see all columns used for the calculation of the conventional venture and also for the Laval nozzle. All the graphics presented in the figures from section 4.1 were plotted using the data calculated with this table in Excel®. The mixing concept detailed in this section is documented at MAN Latim America (2019).

### 3.4 HOW TO RECOVERY ENERGY FROM LINERS

A well-known OEM for commercial vehicles has always been an investor in new technologies to reduce fuel consumption and increase customers' TCO. Thus, this company is supporting and expects to test the first prototypes and their real viability for serial production.

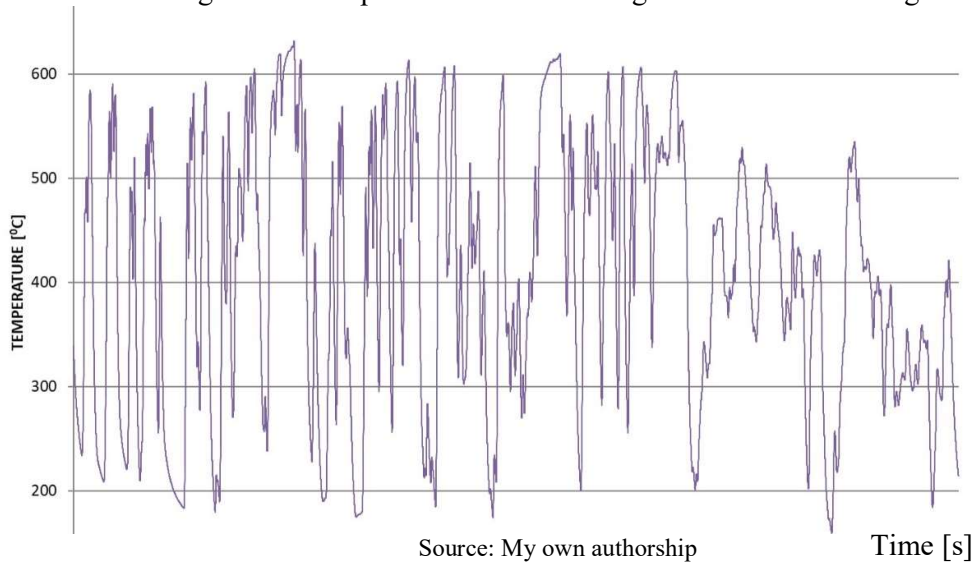
An auxiliary system that recovers coolant heat using the same engine packaging is presented. With this system, it is possible to generate work using the engine liners as a steam generator for a Rankine cycle. A side effect will also be noticed: It is possible to at least reduce the size of the original cooling system, or even suppress it.

As observed in section 3.2, 24% is lost for the exhaust system, and 27% is consumed by the cooling system. The coolant itself is responsible for cooling down all metallic parts in contact with combustion gases inside the engine. Those areas are high potential steam generators, but lack hardware to accomplish this function.

#### 3.4.1 Heat transfer in the cooling system

Figure 70 shows that the average temperature from exhaust gases are approximately 405 °C, reaching peaks of 630 °C. The exhaust manifold (casted iron part) itself reaches temperatures above 500 °C. Note that the temperature varies during the measurements proportionally to the ETC cycle applied to the engine (see section 2.6.1).

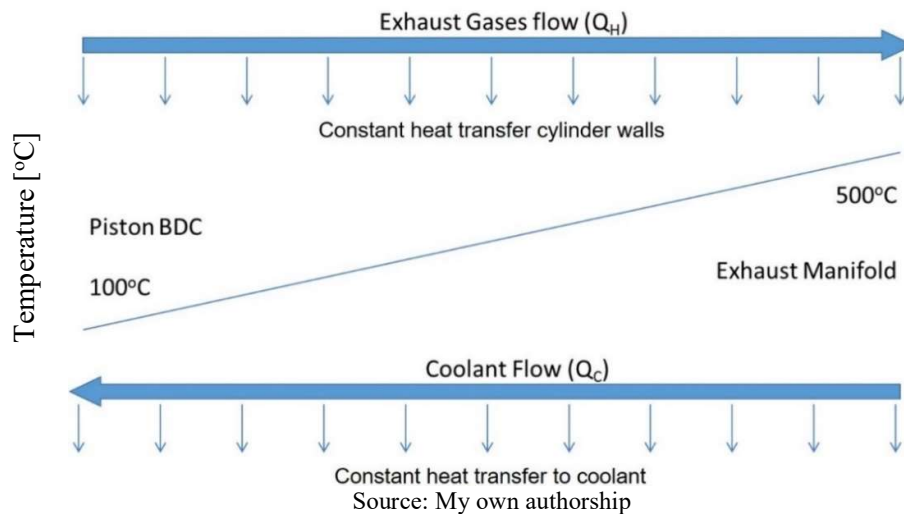
Figure 70: Temperature from exhaust gases before turbocharger



This data comes from the objective measurement done in a test bench. The measured points are shown in Figure 55.

Looking at the cylinder walls, the exhaust gas path and its heat transfer to the coolant are shown in Figure 71. The BDC wall temperature is considered 100 °C, and the highest temperature of 500 °C is the contact between exhaust manifold and engine head. This path means the complete flow of exhaust gases inside the engine (crankcase and cylinder head). Notice that there isn't a linear model to simulate forced convection from exhaust gases to the cylinder wall, and from the cylinder wall to the coolant. The cylinder temperature gradient needs to be considered on the simulation.

Figure 71: Cast Iron temperature around exhaust ducts inside engine



Notice in Figure 72 that the combustion chamber walls face a permanent energy flow that comes from the chamber to the coolant galleries.

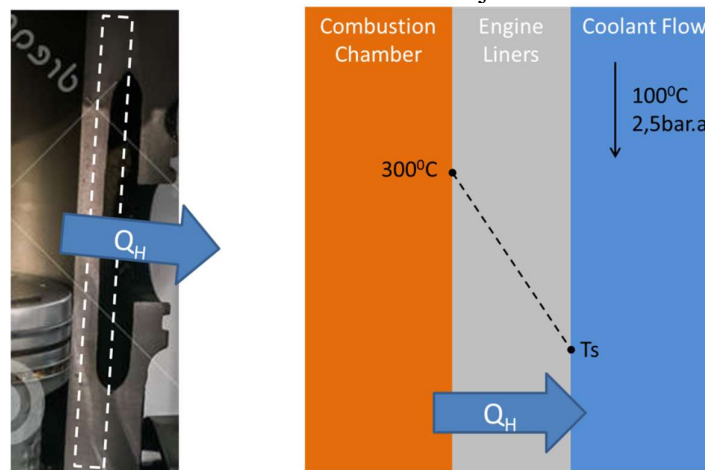
Figure 72: Combustion Chamber and Heat rejection flow to the coolant



Source: My own authorship

The combustion chamber walls can reach temperatures of up to 300 °C. As the Coolant evaporation temperature is around 110 °C on the max system pressure, the external chamber wall cannot be over this temperature ( $T_s$ ). The estimated temperature distribution through the liner walls in a specific height position is shown in Figure 73.

Figure 73: Combustion Chamber and Heat rejection flow to the coolant

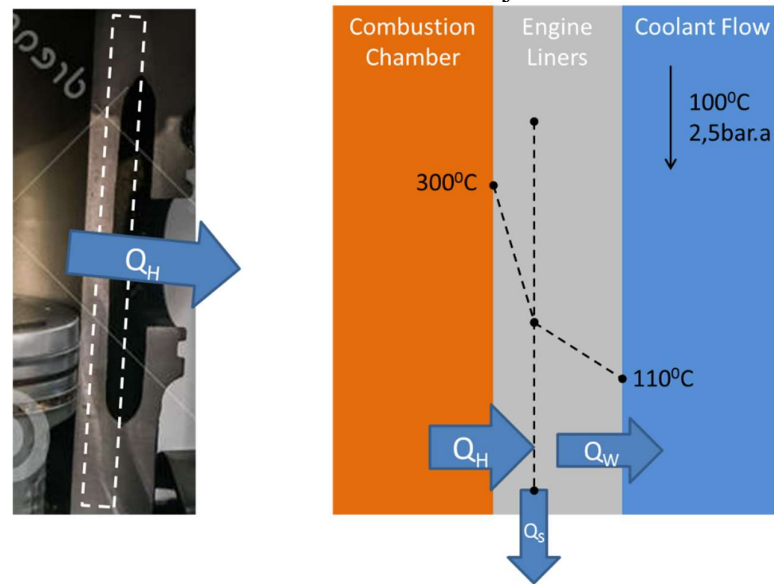


Source: My own authorship

Assuming that the engine combustion chamber temperatures cannot be impacted from the design changes due to risk of oil cooking, the chamber heat extraction must always be kept

independently of the way it is extracted from the walls. Alternative extractions must obey the flow shown in Figure 74, where  $Q_H$  is the heat transferred by convection from the combustion chamber to the Cylinder walls (Liners),  $Q_S$  is the potential heat recovered by a working fluid flowing inside the liners and  $Q_W$  is the remaining heat transferred to the “water”.

Figure 74: Combustion Chamber and Heat rejection flow with extraction

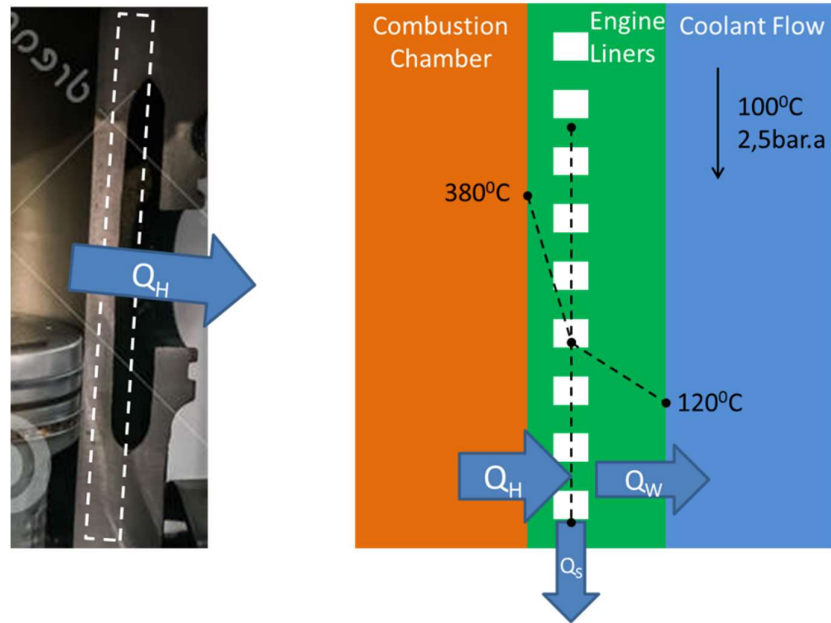


Source: My own authorship

If  $Q_S$  could eventually assume the value of  $Q_H$ , it would mean completely suppressing the conventional cooling system. This is part of the entire project completion, and can be seen in Figure 101 identified as Super Heater.

Heat extraction from engine liners will be limited by heat rejection of the material and channel profile used around the liners (heat exchanger). The super heater for the Rankine cycle assumes some of the dimensional boundary conditions from the available engine liners. As the conduction area will be decreased on the liner walls, as shown in the model from Figure 75, the heat transfer area “lost” for the forced convection cavity will need to be compensated by a material that enables heat transfer to the coolant. This concept is necessary to keep the internal combustion chamber temperature. The concept of an internal channel liner is shown.

Figure 75: Proposal for combustion Chamber and Heat rejection flow



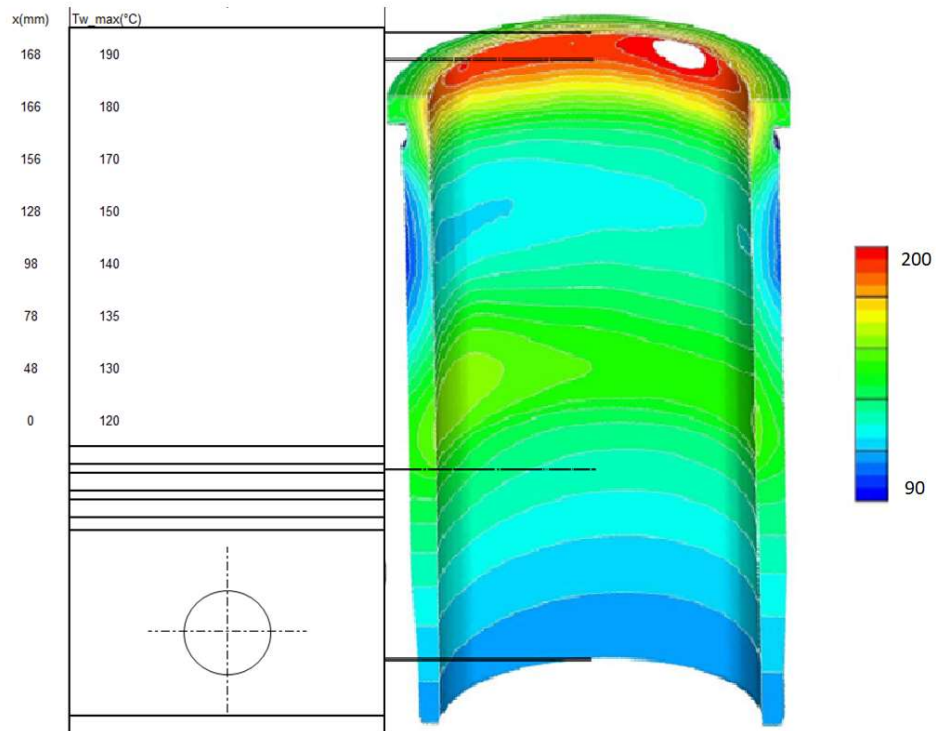
Source: My own authorship

To assess whether an alternative heat extraction could eventually replace the complete cooling system, an analysis of the cooling galleries was performed.

### 3.4.2 Simulation

The temperature distribution shown in Figure 76 is the result of previous simulations available on the OEM databank and enhance the behavior described on section 2.1.2, Figure 7. Notice that the higher temperatures are on the TDC, where the combustion starts and reaches its higher energy (pressure and temperature), expanding the combustion chamber and moving the piston in direction to the BDC, where the exhaust phase starts at low pressure rates.

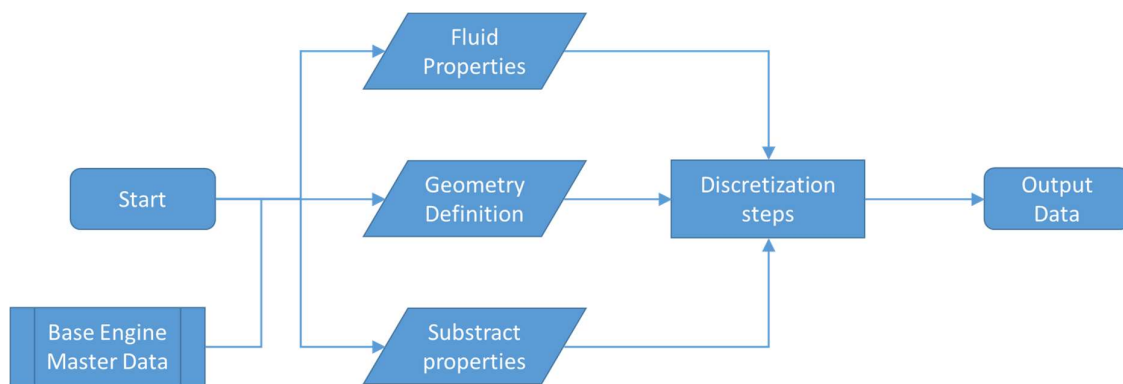
Figure 76: Temperature profile of the liner



Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>2</sup>

The Base Engine Master Data, like shown in Figure 77, is one of the parameter necessary to run the calculation in 400 discretization steps starting from the liquid working fluid entering the channels at 10 bar pressure. Each calculation step determines the fluid condition until the outlet point on the combustion chamber TDC.

Figure 77: Simulation diagram



Source: My own authorship

<sup>2</sup> Stanzer, S, Klammer, J, from MAN Truck and Buses (Austria) and Santos, W.M from Volkswagen Caminhões e Ônibus (Brasil) formed an working group to simulate the purposes idea using GT-Power software.

The simulation starts using:

- Existing engine boundaries conditions, like temperatures along the liners;
- Channels geometry:  $dz$  = liner diameter;  $R$  = radius from the spiral channel;  $d$  = radius from the section of the spiral channel;  $P$  = spiral step; Channel Length;
- Substrate thermal properties, such thermal conductivity and friction factor;
- Fluid properties like thermal conductivity, mass flow, input temperature and pressure and inlet condition (liquid);

By knowing flowrate and boundary conditions, the heat transfer conductivity “ $h$ ” can be determined and the equations 9 and 10 are further used to determine the other fluid properties, which gives the output for the simulation in each from the 400 steps of discretization: Temperature along liners, fluid rate between liquid and steam, pressure drop along the spiral and effective heat transferred ( $Q_p$  = latent + sensible);

For the simulation, round cross-section channels (Figure 78) around liners with thermal conductivity equal to the current crankcase cast iron were considered. The heat transfer with phase change inside a circular tube is defined by Kandlikar, (1990) and can be calculated either with equation 9 or equation 10. The calculation confidence level is high until fluid rate 80% (vapor/liquid).

$$\frac{h}{h_{sp}} = 0.6683 \left( \frac{\rho_l}{\rho_v} \right)^{0.1} \bar{X}^{0.16} (1 - \bar{X})^{0.64} f(Fr) + 1058 \left( \frac{q_s''}{\dot{m}'' h_{fg}} \right)^{0.7} (1 - \bar{X})^{0.8} G_{s,f} \quad (9)$$

$$\frac{h}{h_{sp}} = 1.136 \left( \frac{\rho_l}{\rho_v} \right)^{0.45} \bar{X}^{0.72} (1 - \bar{X})^{0.08} f(Fr) + 667.2 \left( \frac{q_s''}{\dot{m}'' h_{fg}} \right)^{0.7} (1 - \bar{X})^{0.8} G_{s,f} \quad (10)$$

$$0 < \bar{X} < 0.8$$

Where:

$$h_{sp} = \frac{Nu_D \cdot k}{D} \quad (11)$$

$h_{sp}$ ...heat transfer coefficient

$Nu_D$ ...Nusselt number

$k$ ...thermal conductivity (Fluid)

$D$ ...tube diameter

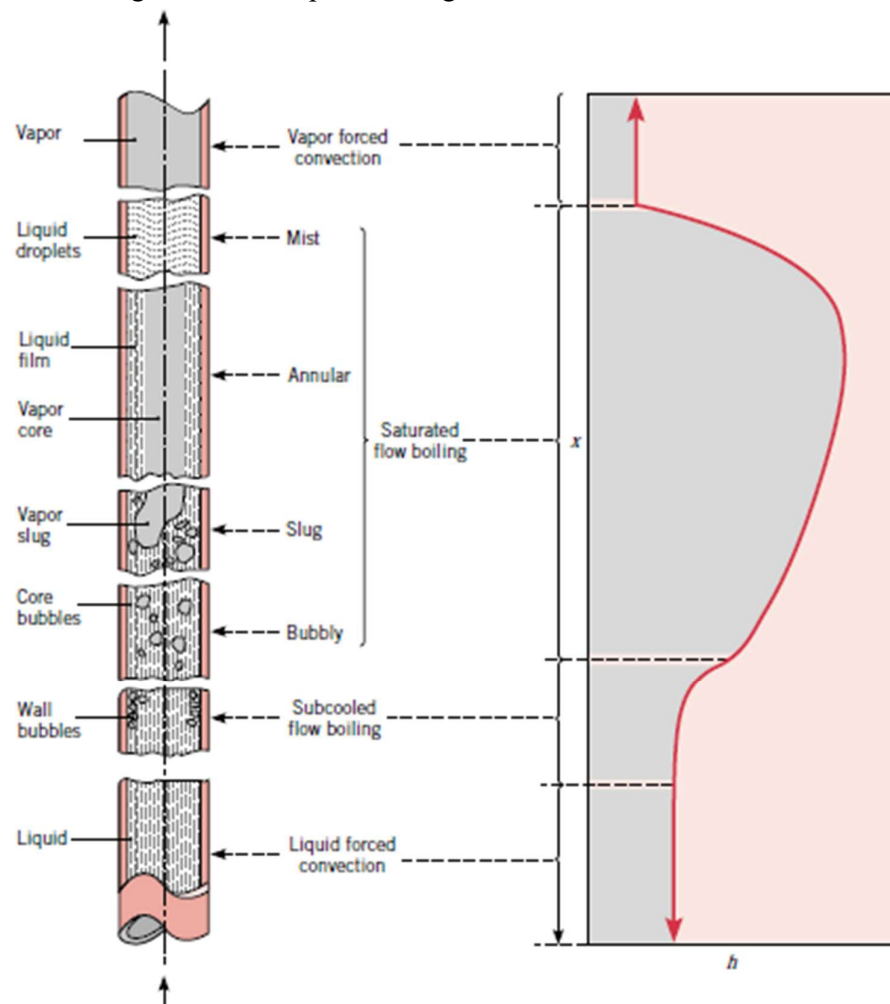
$$f \equiv \frac{-(dp/dx)D}{\rho u_m^2/2} \quad (12)$$

$f$ ...friction factor

$u_m$ ...mean velocity

The expected phase change behavior is shown in Figure 78. The available energy throughout the channel must be enough to change the liquid forced convection into overheated (vapor) forced convection. This model is used on the spiral channel purposed around liners, starting on BDC and ending on TDC. This track is divided in 400 steps of calculation.

Figure 78: Fluid phase change inside a heated circular tube



Source: Kandlikar (1990).

Using the GT Power software with application of the chosen geometry and premises, seven different scenarios were simulated trying to find a way to get superheated fluid on the liners output. All cases were analyzed considering:

- Cast iron thermal conductivity;
- Ethanol as working fluid;
- Same distance between the channels;
- 400 discretization steps;
- Same liner diameter:
- Channel geometry is a channel composition or diameter with distance to the chamber wall;

GT, or “Gamma Technologies” is a company that provides CAE Platforms for Multi-Physics System Simulations. GT-POWER one of the applications and is considered today the industry standard engine performance simulation, in use major engine manufacturers and vehicle OEMs. GT-POWER is used to simulate engine performance considering minor or major software and hardware changes. Quantities such as power, torque, airflow, volumetric efficiency, fuel consumption, turbocharger performance and matching, pumping losses, are some of the examples given. GT-POWER incorporates physical models that allows it to simulate from the combustion chamber to tailpipe-out emissions, intake and exhaust system, including acoustic characteristics (level and quality). Hardware structure, temperature, pressure analysis, and control system modeling are also possible features from the software. The software offers Standard engine models that are easily converted to real-time capable models (GT,2019).

The software is applicable to any size engine, from smallest utility engine to largest marine application; is flexible to allow studies of innovations; it includes combustion and after treatment models; is flexible turbocharger modeling (wastegated, VGT, supercharging, two-stage, turbo compounding and twin-entry turbine); includes complete controls library for dynamic system controls studies and all thermo and fluid dynamics simulations. The software also includes advanced features and applications that includes chemical kinetics library, vehicle models for integrating engine and vehicle simulations. It can be linked with Simulink, Star CD, Converge, Fluent and other codes; finite element solution of cylinder structural temperatures, useful for part load and dynamic operation.

## 4 RESULTS

### 4.1 COLD ENGINE GAS RECIRCULATION SYSTEM

Since the Bernoulli principle works for all situations where non-compressible air flow increases, a system was designed to allow pressure drop, increasing air velocity only when exhaust gas is required. This design allows the system to decrease heat rejection.

Figure 79 shows the temperature that would result from mixing fresh air directly with exhaust gas from the tail pipe. Note that this setting does not consider recovering any power stations after the treatment system. If this system is considered in parallel, the exhaust gases would have even lower temperatures.

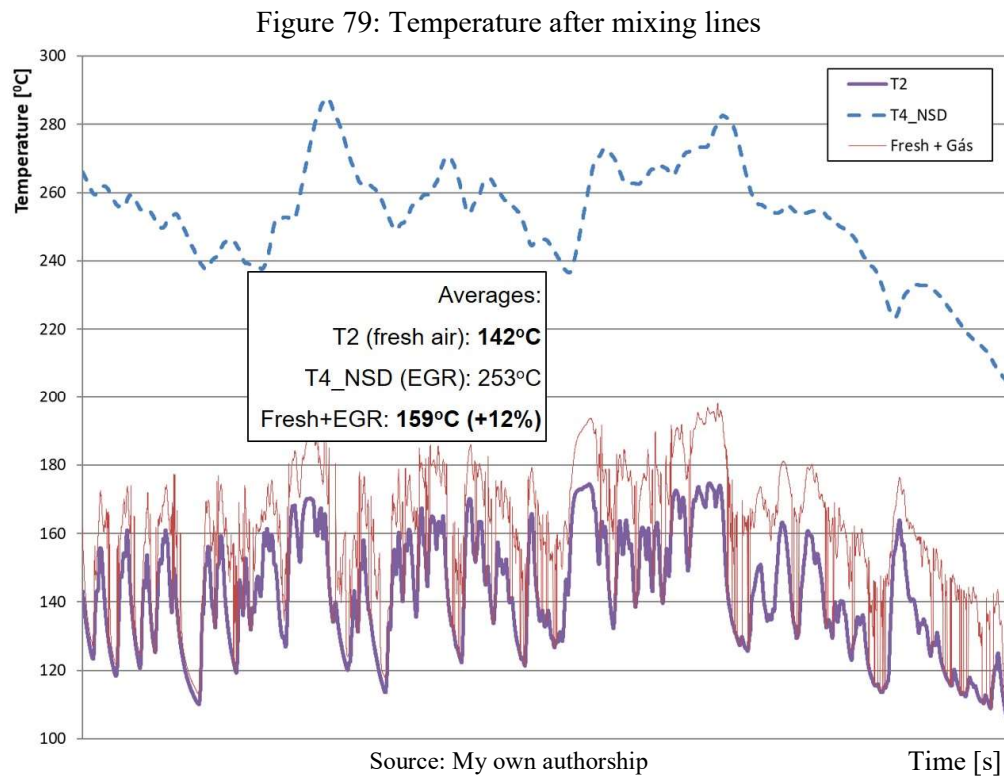
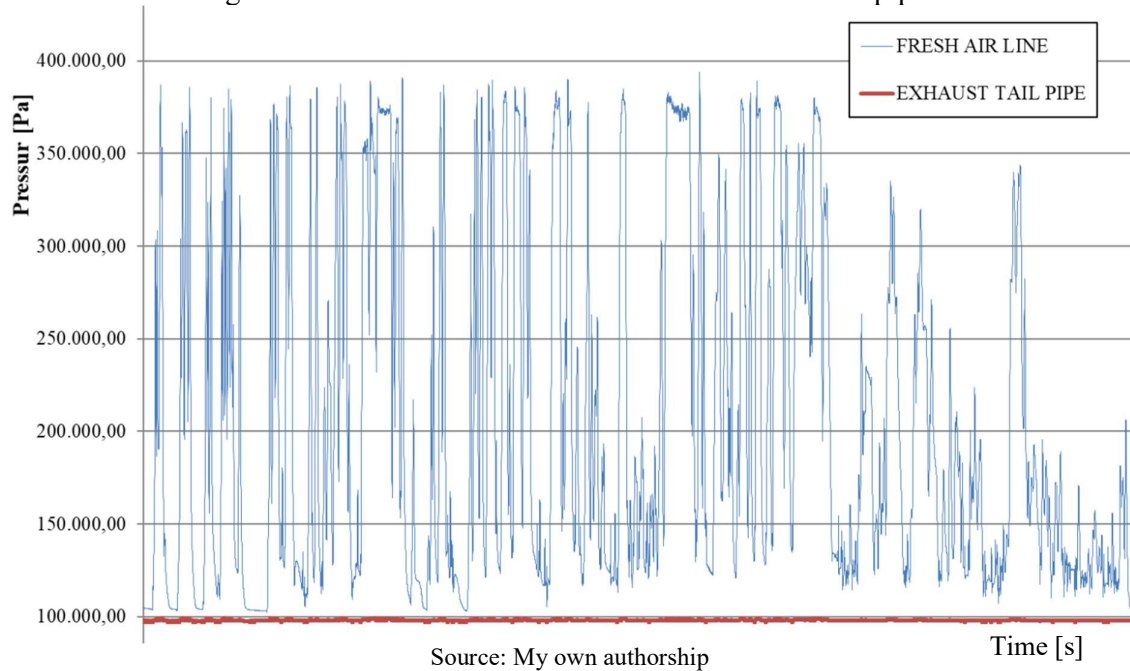
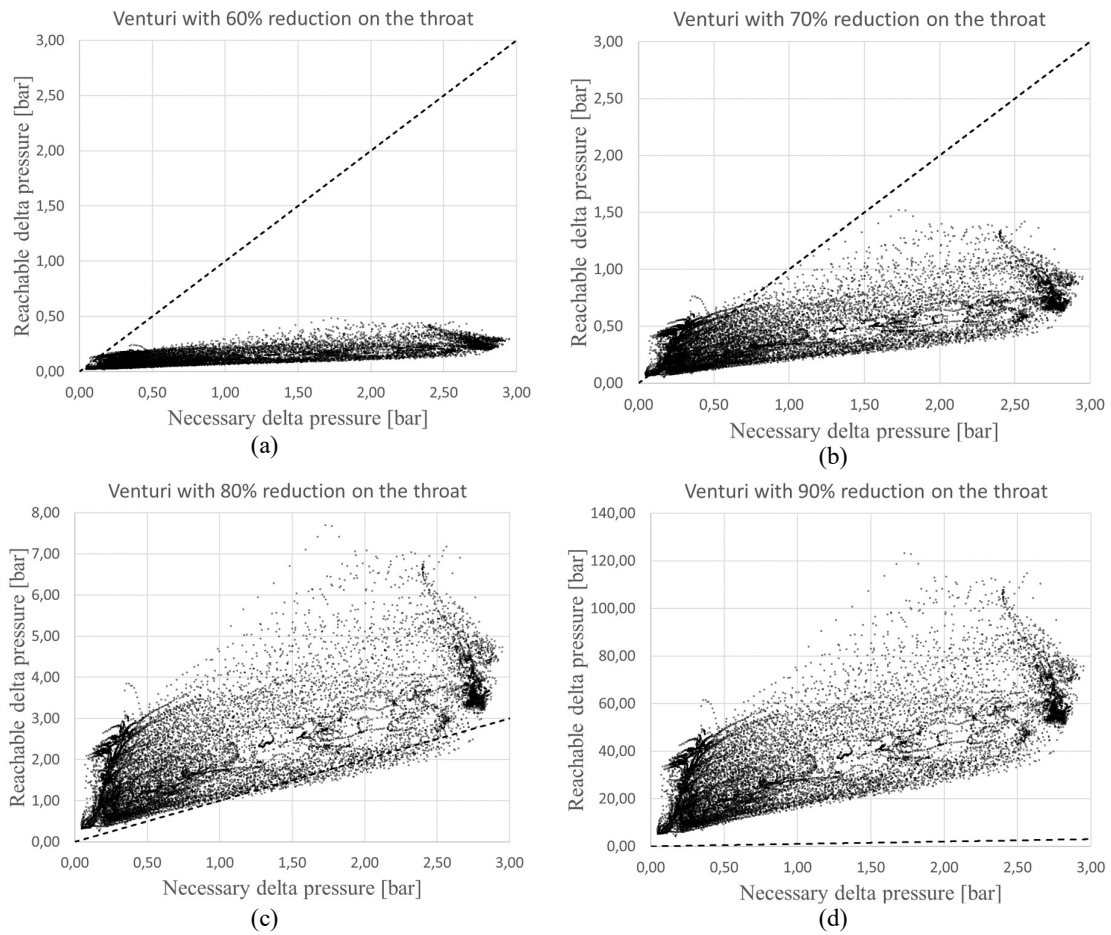


Figure 80 shows the fresh air pressure and the exhaust gas pressure in the tail pipe throughout the measurement. This is also data collected from the testbench from Figure 55, as well as the data from Figure 80.

Figure 80:  $\Delta P$  between Fresh air line and Exhaust “Tail pipe”

To overcome this  $\Delta P$ , it is necessary to know the venturi throat and inlet diameter. To reach reasonable simulations, four different venturi throat reductions were considered: 60%, 70%, 80% and 90%.

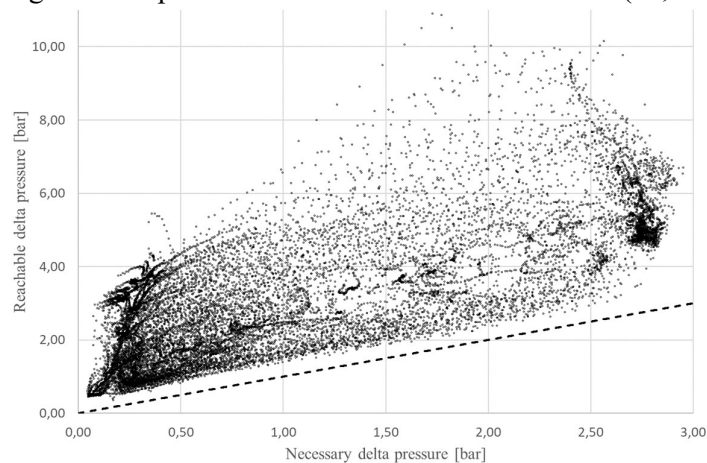
Figure 81 shows the impacts of changing the Venturi throat. The curves are plotted to compare the required  $\Delta P$  with the theoretical attainable  $\Delta P$  obtained from equation 5. Note that this parameter has been calculated for all points measured during the ETC cycle. The dotted line is a curve plotted based on the required pressure drop (maximum 3 bar) and is maintained on curves (a), (b), (c), and (d). For curve (a), it is noted that the section reduction was hardly acceptable to create an attainable  $\Delta P$  necessary to overcome the required  $\Delta P$ . Curve (b) shows more points exceeding the required pressure drop. In curve (c), almost all necessary  $\Delta P$ s are reached, and in curve (d), all points are exceeded, even more than the necessary.

Figure 81: Throat reduction impacts on  $\Delta P$ 

Source: My own authorship

In order not to unnecessarily reduce the cross section, the mathematical model was defined to find the ideal throat diameter, i.e. that the necessary  $\Delta P$  would be attained at all the point of the ETC cycle (see Figure 82).

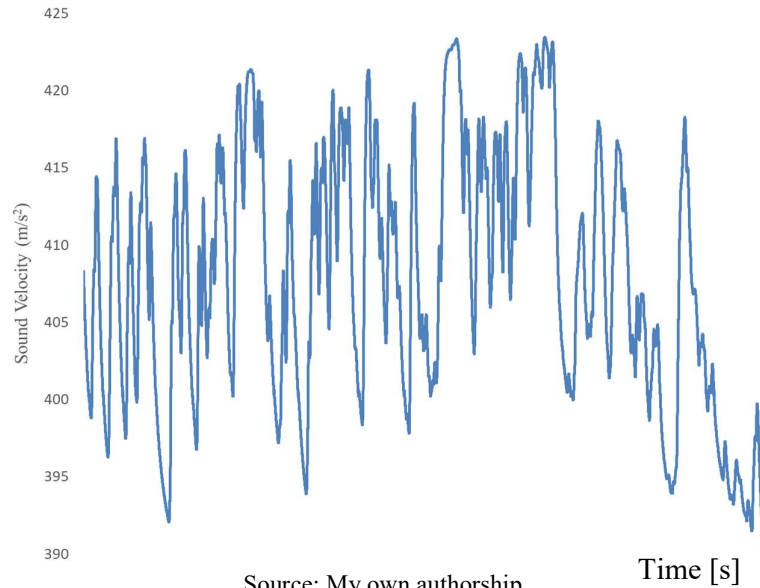
Figure 82: Optimal Venturi reduction on the throat (81,7%)



Source: My own authorship

Notice that in this condition, all points are above the dotted line. I means that the reachable delta pressure is always higher than the necessary delta pressure. With the necessary throat reduction, the Venturi local speed of sound was calculated and is observed in Figure 83.

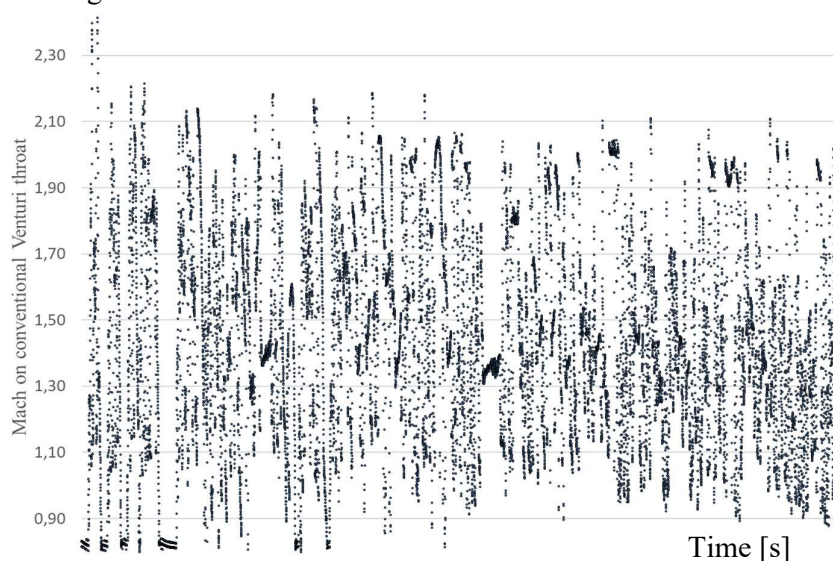
Figure 83: Venturi throat local speed of sound considering temperature and density



Source: My own authorship

Comparing the velocity flow of the Venturi throat with the calculated speed of sound, Mach was obtained and plotted in Figure 84, where Mach (Ma) is a dimensionless parameter that express the relative importance between inertial and compressibility forces. Its representability for internal flow determines whether it is compressible or not.

Figure 84: Mach in Venturi throat in line with fresh air flow



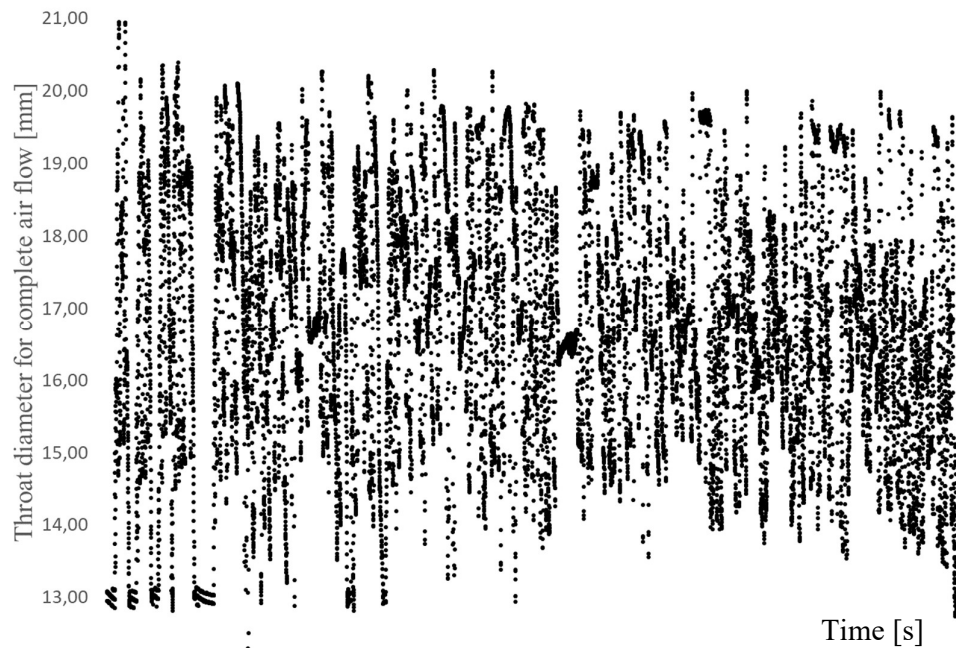
Source: My own authorship

For internal flow (ducts), the most important question is whether Mach is above or below one. Since most of the flow through the throat exceeds  $\text{Mach} = 1$  (Figure 84), it can be concluded that incompressible flow simulation is not feasible to achieve the pressure drop required to mix fresh air with EGR from the tail pipe.

To overcome the technical constrain imposed by incompressible flow simulations, it was necessary that the first proposed system had its conventional Venturi concept replaced by a supersonic Laval nozzle.

With a fixed inlet area of 60mm, the sonic throat was calculated for each measurement point in the ETC cycle. The supposedly required values are shown in Figure 85.

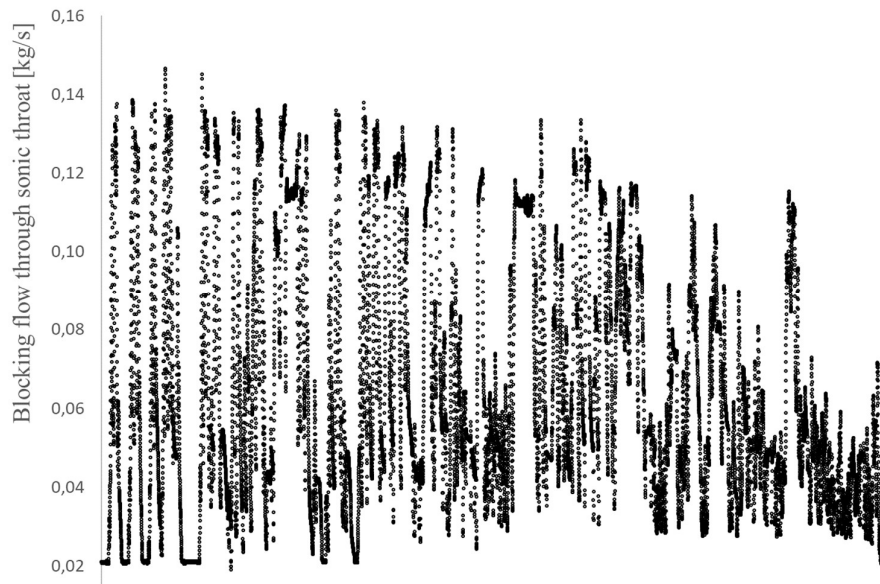
Figure 85: Laval nozzle sonic throat diameter



Source: My own authorship

The hypothetical sonic throats obtained in Figure 85 leads to specific blocking flows that are shown in Figure 86.

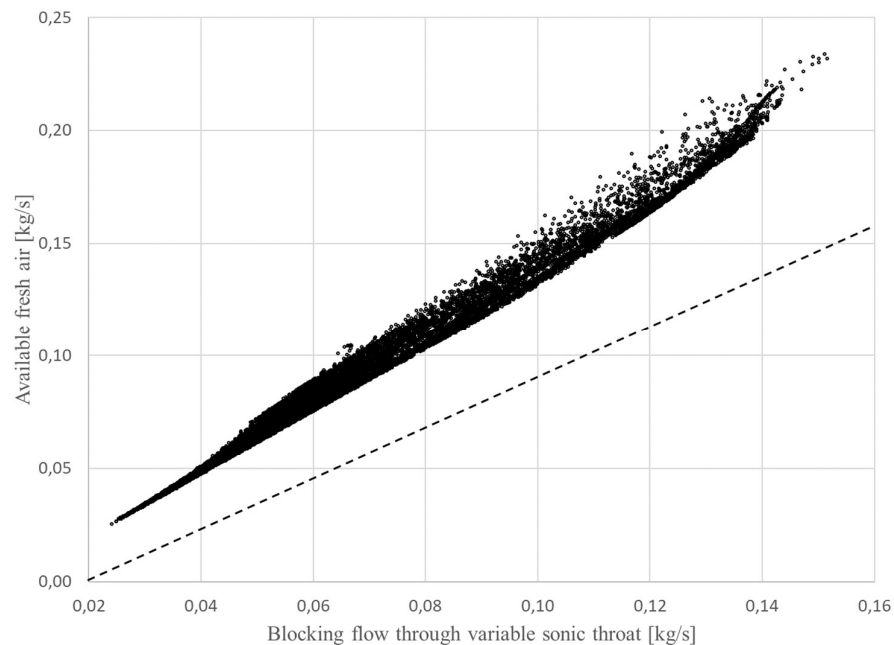
Figure 86: For different sonic throats



Source: My own authorship

The analysis shown so far considers a hypothetical in-line Variable Throat assembly, which would allow a maximum air flow of 0.15 kg/s. This construction, besides its low efficiency, would not allow the complete and necessary air flow to the engine, reaching peaks of 0.23 kg/s of fresh air. Figure 87 shows that all measured points would have more fresh air flow than the calculated blocked throat.

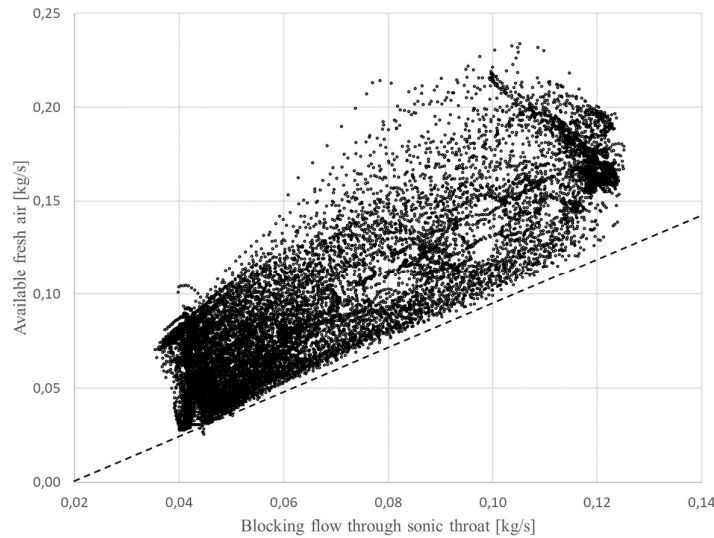
Figure 87: Hypothetical in-line Variable Throat



Source: My own authorship

To allow for fresh air flow available, a fixed throat was estimated based on Figure 88. Using a fix throat (around 17mm) would mean reducing 92% of the section. Figure 87 shows what the blocking flow behavior would be like compared to the fresh air available in this situation. Modeling this curve uses the same parameters shown, but instead of determining the throat by the  $A/A^*$  ratio, the throat is fixed.

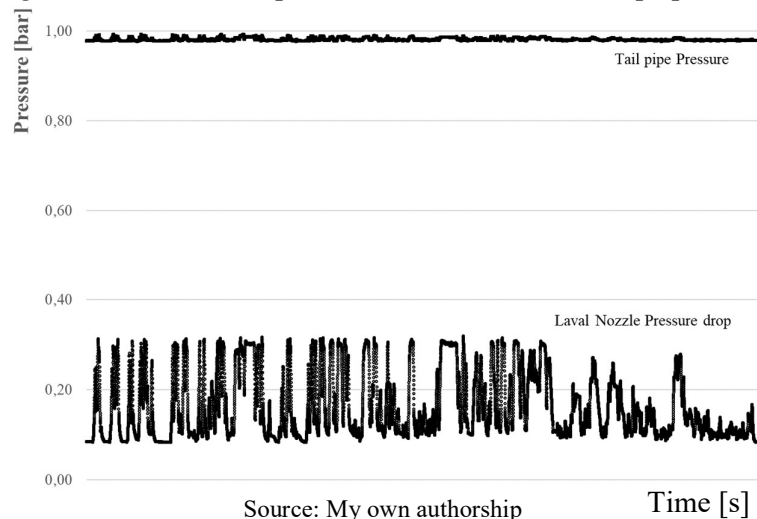
Figure 88: For fix sonic throat



Source: My own authorship

Figure 88 shows that fresh air demands above 0.14kg/s would not function according to any nozzle valve because its value is greater than the blocking flow. Therefore, the solution to divert air flow to the Venturi with a flow deviation valve would now not only be desirable, but mandatory (see Figure 68)

Figure 89: Pressure drop from Laval Nozzle x Tail Pipe pressure



Source: My own authorship

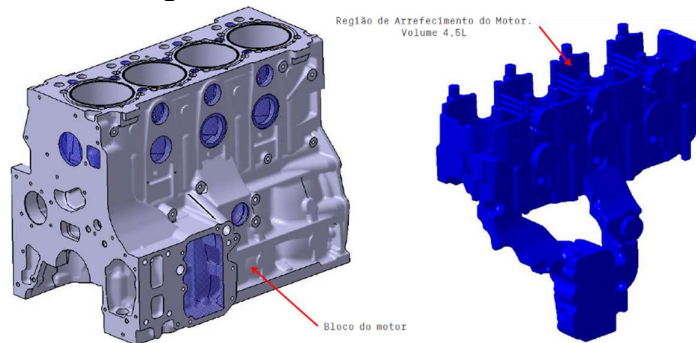
Note that for all measured points using a 17mm throat, there will be increased air demand as needed to make the Laval nozzle work. The first step would be to divert the remaining air out of the Laval Nozzle flow. Considering sonic flow for the lowest fresh air flow through the Laval Nozzle, the reachable pressure drop is plotted in Figure 89.

#### 4.2 HEAT RECOVERY FROM LINERS

The 4.5-liter engine shown in Figure 49 is in contact with the coolant in the following areas:

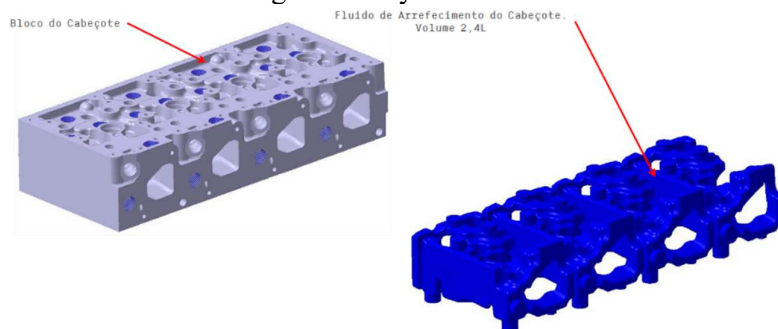
- Crankcase (Figure 90): 4.5L of coolant and  $0,5\text{m}^2$  of surface;
- Cylinder head (Figure 91): 2.4L of coolant and  $0.28\text{m}^2$  of surface;
- EGR cooler (Figure 92): 1.2L of coolant and  $0.14\text{m}^2$  of surface;
- Thermostatic valve housing (Figure 93): 0.3L of coolant and  $0.4\text{m}^2$  of surface;

Figure 90: Crankcase and oil cooler



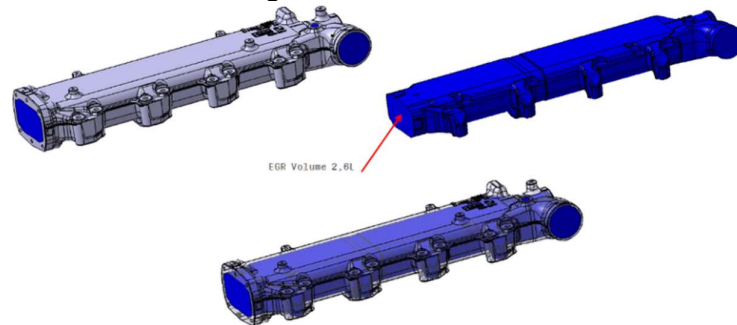
Source: My own authorship

Figure 91: Cylinder Head



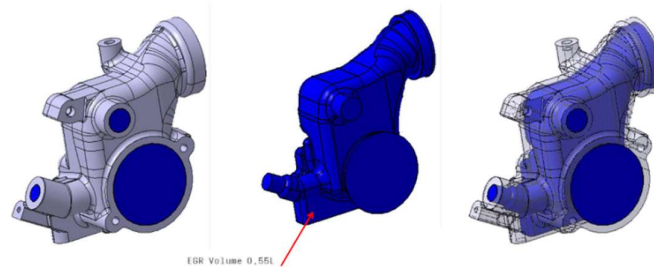
Source: My own authorship

Figure 92: EGR Cooler



Source: My own authorship

Figure 93: Thermostat Valve Housing:



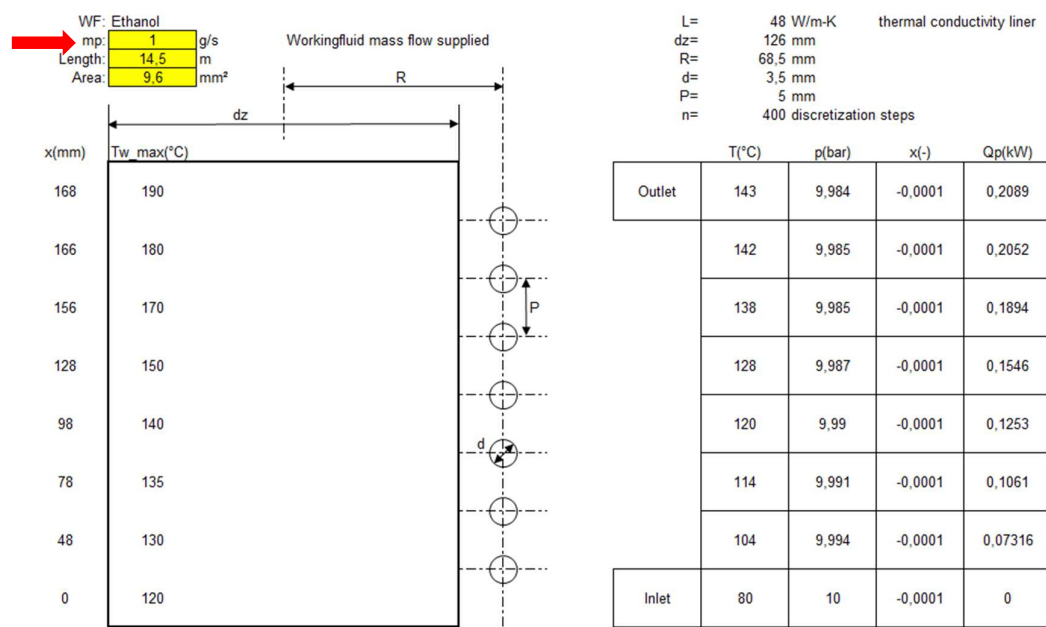
Source: My own authorship

Assuming the engine as a control volume, the mass flow (in/out) is 270kg per min. It flows constantly through the galleries with a complete heat exchange surface of  $4.4 \text{ m}^2$ . Despite the losses by convection around the engine and periphery, the heat transferred to the coolant in the engine in the study is 105kW, which would mean roughly  $23.86 \text{ kW per m}^2$ . It is clear that not the entire surface is in contact with a high heat source, but considering the turbulence and high-speed flow, this estimate is a truthful value to the objective under discussion.

In order to avoid oil cooking, any system developed to substitute the existing “water cooled” galleries should be able to keep the combustion chamber walls temperature at the same level. In this way, seven cases were simulated for the proposed superheater around the liner (from Figure 94 to Figure 100). It is possible to notice, on the left side, a schematic half section of the combustion chamber, where  $x(\text{mm})$  represents the vertical displacement, and  $T_{w\_max}(\text{°C})$  represents the internal wall chamber temperature. There is a schematic plot of the channels in the middle, which surrounds the combustion chamber and its main discrete dimensions. On the right side, a table represents the behavior of the working fluid (ethanol) flowing inside the channels (Figure 78). It is represented by the temperature (T) increase, pressure (p) drop, and transferred Heat (Qp) from the engine channels to the working fluid. The inlet condition was assumed as  $80 \text{ °C}$  and 10bar. The GT-Power module receives the mathematical model with the geometrical specifications. All input data, since geometry data

until temperature, pressure, discretization steps, thermal conductivity, friction and working fluid are inputs to the model, which is able to simulate all desired situations. The boundary conditions and simulations were done from a OEM working group (STANZER, 2019).

Figure 94: Liner evaporator: Case 1.



Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>.

Figure 94 shows case 1, which considers a channel geometry called “1” and low mass flow = 1 g/s (see an arrow on the left of the value “mp”). It resulted in a laminar flow and low-pressure loss, but a low heat transfer and no evaporation.

<sup>3</sup> Stanzer, S, Klammer, J, from MAN Truck and Buses (Austria) and Santos, W.M from Volkswagen Caminhões e Ônibus (Brasil) formed an working group to simulate the purposes idea using GT-Power software.

Figure 95: Liner evaporator: Case 2.

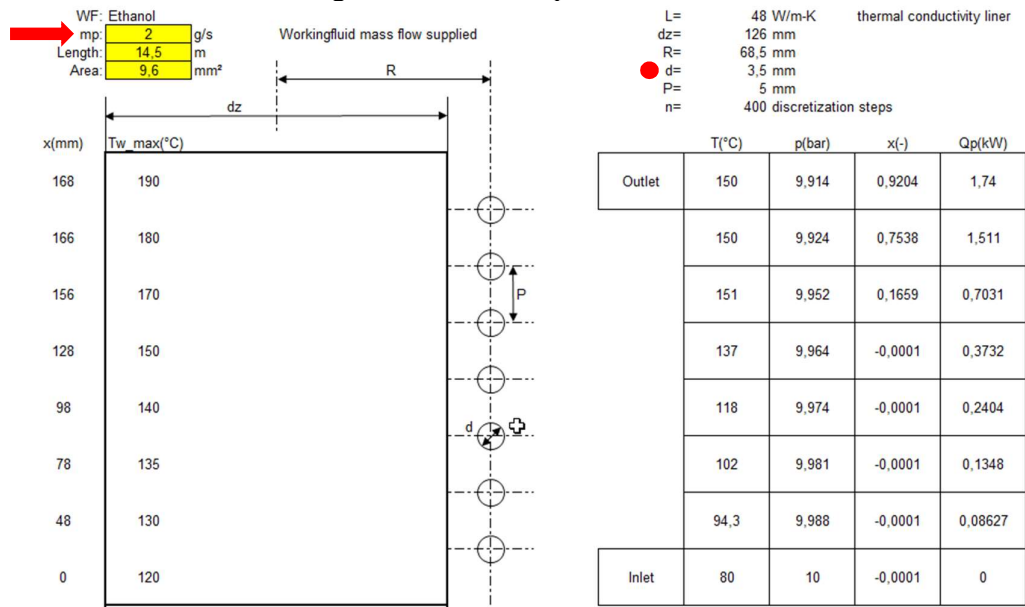
Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>.

Figure 95 shows case 2, which uses the same channel geometry from case 1, but a higher mass flow (see an arrow on the left of the value “mp”– base for comparison with case 1). It resulted in an almost completely evaporated output.

Figure 96: Liner evaporator: Case 3.

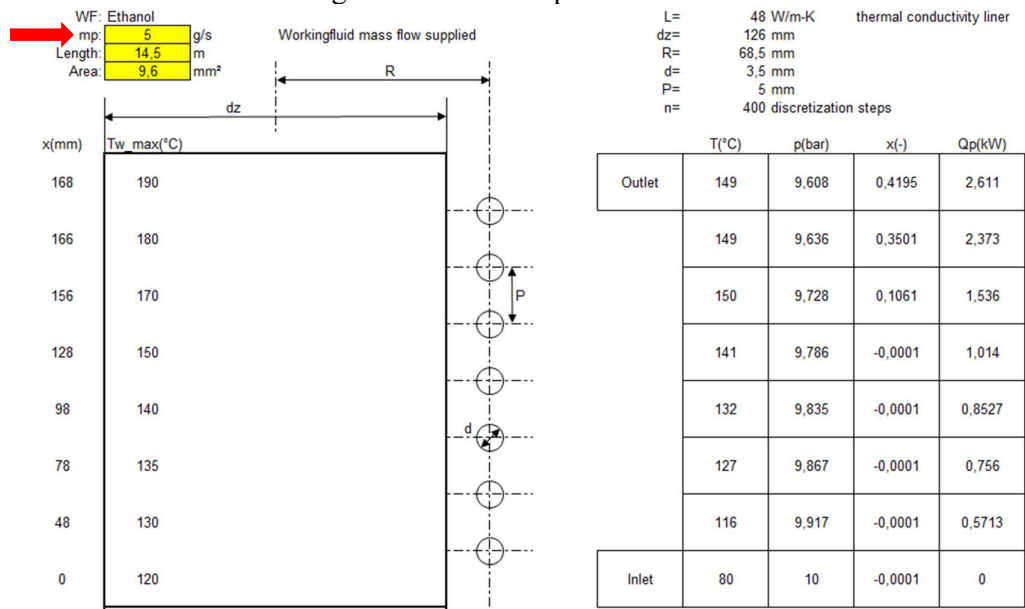
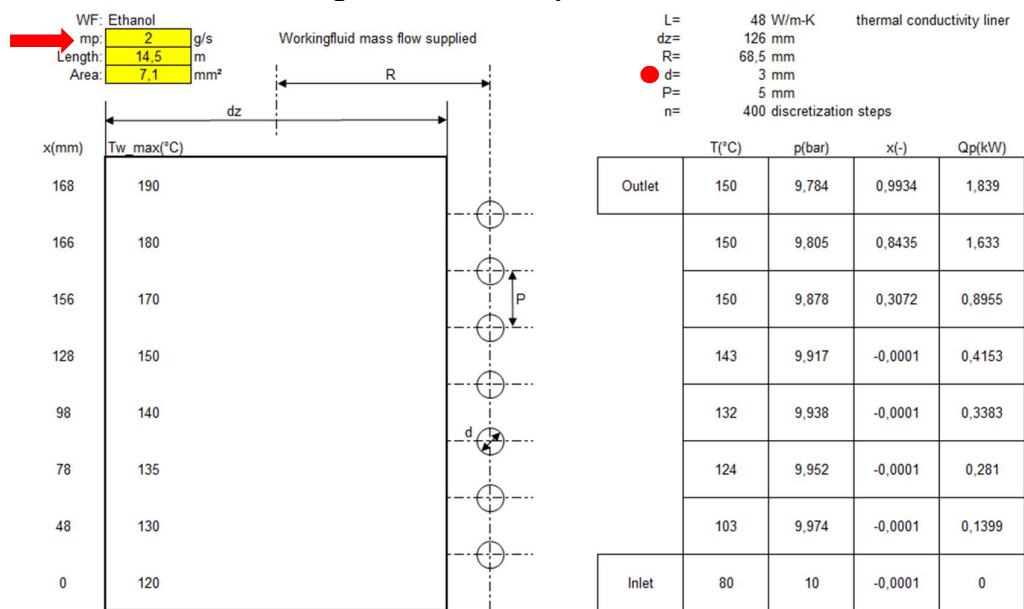
Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>

Figure 96 shows case 3, which uses the same geometry from case 2, but increases mass flow to 5g/s (see an arrow on the left of the value “mp”– base for comparison with case 2). It resulted in approximately 42% evaporation (less than case 2), but had a high heat output absorbed by the working fluid. It would be an interesting condition to work in series, downstream from the next evaporator. For example, post after treatment system.

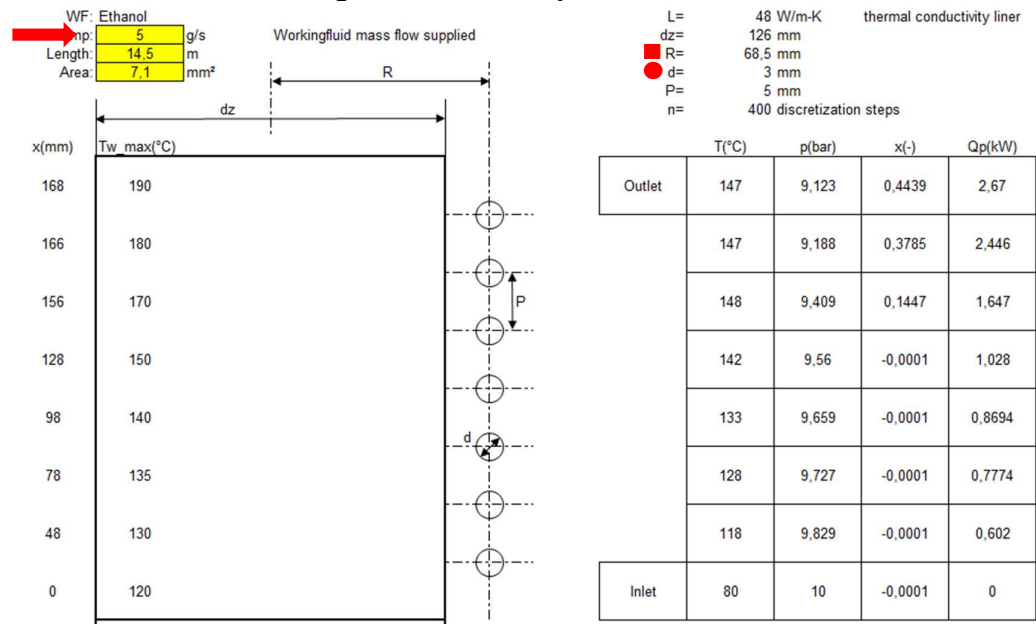
Figure 97: Liner evaporator: Case 4.



Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>.

Figure 97 shows case 4, that, when compared to case 2, has a smaller channel diameter = 3 mm in comparison with 3,5 mm (see a round mark on the left of the value “d” on cases 2 ad 4). Resulted in a slightly better heat exchange, but increased pressure drop.

Figure 98: Liner evaporator: Case 5.

Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>.

Keeping the Geometry parameter “d” = 3mm and increasing mass flow “mp” from 2g/s (case 4) to 5g/s, Figure 98 shows case 5, which has low improvement of heat exchange, with a higher pressure drop.

Figure 99: Liner evaporator: Case 6

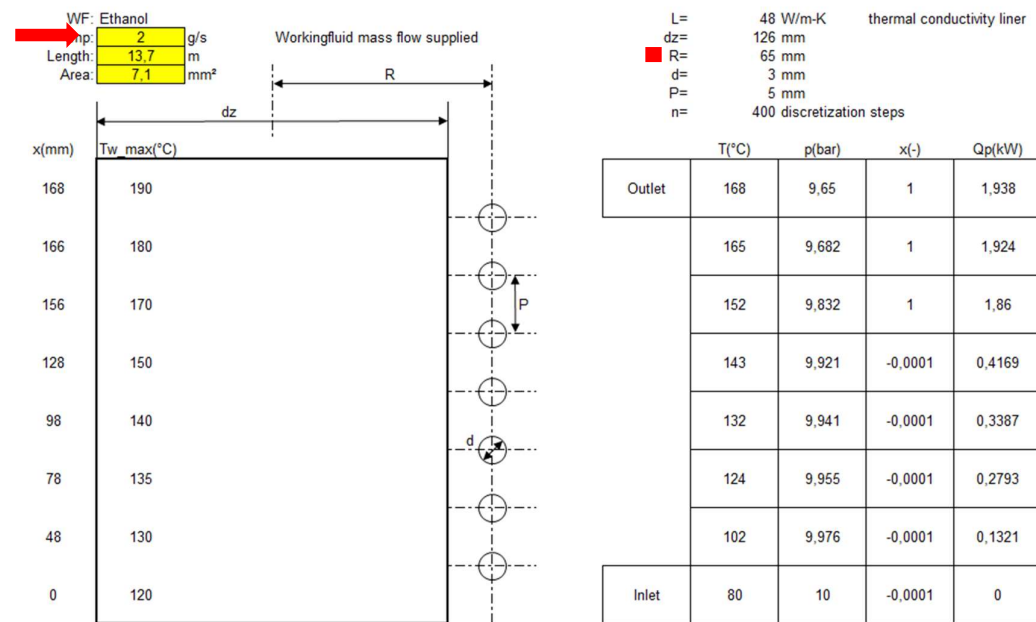
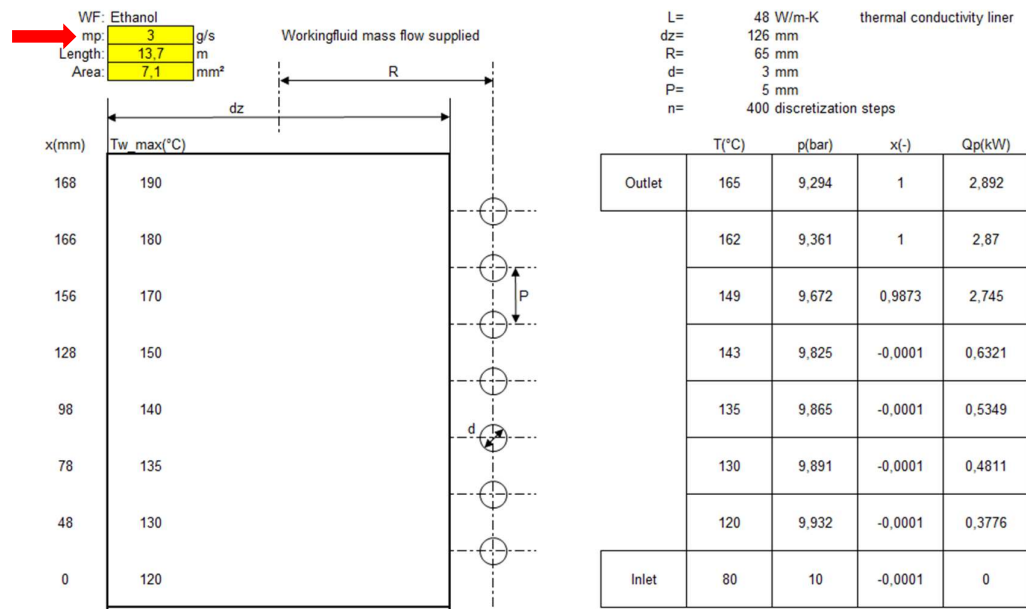
Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>

Figure 99 shows case 6, which also changes geometry, but not anymore the channel diameter. Now the updated parameter is the wall thickness to chamber that is the result of decreasing the parameter “R” from 68,5mm to 65mm (see a square mark on the left of this parameter. This is the channel geometry 3. There is a sensible improvement in heat transfer and complete phase change with overheated output. This condition is recommended for the installation of expander in series upstream evaporator.

Figure 100: Liner evaporator: Case 7



Source: Stanzer, S, Klammer, J. and Santos, W. work group <sup>3</sup>

Figure 100 shows case 7, where channel geometry 3 is kept in relation to case 6, and mass flow “mp” is increased to 3g/s. There was a sensible improvement of heat transfer.

A compromise between cases 6 and 7 shall be found. In one hand, the case 6 reaches the overheated condition faster than case 7, but removes less energy from the cylinder walls. On the other hand, case 7 removes more energy, but may represent a problem to the expander, which works with superheated Steam to generate power. Only objective measurements will be able to find this operation point.

## 5 DISCUSSION

Thermal simulations using GT Power and fluid mechanic simulations with compressible flows already consolidate the knowledge and do not represent the main contribution. The key findings are demonstrations that it is possible to use wasted energy from the cooling system and that it is possible to have an EGR system completely without a heat exchanger flowing through outside the turbocharger.

These two solutions were able to demonstrate that a conventional “water-cooled” system is completely replaceable with a system that uses ethanol as a working fluid and functions as a Waste Heat Recovery station.

The baseline engine uses an HP-EGR. Charged air (after turbocharger) is cooled by a Charge Air Cooler (CAC). The cooling system uses additive water, which exchanges engine heat for liquid-air heat exchanges (radiator) using the principle of forced convection with liquid flow powered by a water pump placed on the engine. To transfer this system into the proposed enhanced system, some logical steps must be taken (see Figure 48).

### 5.1 (1) COLD EGR (WITHOUT COOLER);

The first step in achieving this work objective was modeled in section 3.3. The result of this action is a system that can reject less energy to the radiator increasing engine efficiency. There are both economic and technical side effects with this action: it would be cheaper due to cooler-free EGR hardware, due to cheaper control valves, applicable on lower temperatures than the initial HP-EGR and due to lower capacity radiator to be installed on the vehicle; besides a more efficient thermal machine (lower heat rejection), the engine can also benefit in performance or fuel consumption because the hot gases would not anymore be deviated to the EGR cooler, being 100% available to the Turbocharger, but this result depends on further steps in this work.

Since the recirculation gas temperature is still not low enough for the intake, the Laval Nozzle must be installed before the charge air cooler (air-air). At the rated speed (1,750rpm), on the one hand, the lack of EGR would decrease the radiator heat exchange capacity by 19.5kW (Figure 51) with a delta temperature of 323K (403-80). As the system reduces the delta to 173k (253-80), the energy spent to cool the gases would be 10.4kW. On the other hand, the charge air cooler should increase its heat exchange capacity by 9.1kW (19.5-10.4).

The purposed system has potential to decrease even more the EGR temperature in combination with the existing CAC. Hountalasa, et al., 2008 shows that the effect of lower EGR temperature is stronger at high EGR rates. It has a positive effect on Brake Specific Fuel Consumption (BSFC) and lower soot (a black powdery or flaky substance consisting largely of amorphous carbon, produced by the incomplete burning of organic matter).

The Laval Nozzle clearly differs this system from the Low pressure EGR presented in 2.2.3 affirming its innovative application. The use of a Laval Nozzle to admit the inert gases from a low pressure line after the turbo charger, and eventually after CAC, brings thermal and durability advantage in comparison with the known systems and may mean one of the reasons for increasing life of ICE.

## 5.2 (2) SUPER HEATER AROUND LINERS;

The superheater around engine liners, proposed in section 3.4, has been shown to be able to completely replace the “water” cooling system with a working fluid that can recover some of the wasted energy. As presented in section 2.4, simple Rankine cycles are capable of recovering up to 30% of the overheated working fluid. Differently from the researches showed in section 2.5.4, it is shown that the heat transfer ability progressively increases on the seven simulation steps. Channel geometry and mass flow ratio are clearly significant variants to be considered.

Based on the data from case 7, the spring profile around the liners presents a 13.7m long tube with a 3mm diameter, and a total shell area of 0.13m<sup>2</sup>. Considering the total heat transfer ability of 2,892W, case 7 shown in Figure 58, the heat extraction is 22.2kW/m<sup>2</sup>.

The mathematical model has been simplified to a round section to use existing empirical data, but improvements in the profile and use of better conductivity materials may increase potential transferability.

The evaporator to perform a Rankine cycle must meet the dimensional boundary conditions of the available engine liners. As the conduction area will be decreased in the liners walls, as shown in the model of Figure 75, the “lost” heat transfer area to the forced convection cavity needs to be sized to compensate for heat extraction from the combustion chamber. This situation could potentially mean  $Q_H = Q_S$  (Figure 75), and is necessary to maintain the internal combustion chamber temperature.

The working fluid for the purposed Rankine cycle should ideally be the same for the cooling system, which should be easily available on the market. However, the fluid that meets this requirement (ethanol) raises many concerns about system security.

The use of ethanol in an OEM Waste Heat Recovery (WHR) system shows corrosion and performance effectiveness to support the decision of using it in an OCR cycle. A system, currently not integrated with cooling, is now running tests, but unfortunately the specifications and safety features are not disclosed at this time. An article about this test is referred to in item 2.5.3.

With cycle fluid definition, a positive displacement pump must be specified to control the working fluid flow, and an expander can be defined based on the achieved steam generation obtained from the mathematical model.

Applying a system capable of recovering power from the cooling system will be a potential for replacing or at least reducing the conventional cooling system. The concept of this superheater allows the application of a WHR without the need for an extra installation package, which is one of the biggest challenges in the existing WHR vehicle proposals.

Smaller channels diameter improves heat transfer at low and medium mass flows due to turbulent flow. The low wall thickness improves heat absorption of the working fluid by limiting material resistance. The liner evaporator is suitable for steam generation or mixed steam for subsequent overheating, e.g. EG-evaporator. Considerable heat absorption into the working fluid.

The smaller diameter of the channels increases pressure loss, especially in the overheated area. Limiting the permissible wall temperature prevents overheating at higher pressures.

### 5.3 (3) INTEGRATION WITH EXHAUST SUPER HEATER;

The exhaust gas superheater purposed from Carstensen et al., 2019 decreases the Exhaust gases temperature on the Tail pipe. By integrating this system to the purpose cold EGR, the recirculation gases will decrease its temperature even further. The first step temperature reduction was taking the inert gases out of Tail pipe. Now that Tail pipe temperature are lower, the overall system efficiency will increase.

Carstensen et al., 2019 plays an important role on this context. As a demonstration vehicle is available for tests, all energy improvements solutions created inside the group can be

used to improve the WHR capability and overall better energy use. The demonstration vehicle shows an average decrease of 60% on the exhaust gas temperature after the spiral superheater. It would mean lowering the EGR delta temperature from 173k (253-80) to 71.8k (253\*60%-80), representing only 4kW of heat rejection compared to the initial 19.5kW (delta temperature of 403-80 = 323k) . Beyond the energy benefits, the hardware packaging requirement would lower and cheaper by the elimination of the EGR cooler. Additionally the inert gas in the tail pipe would have already been treated and would present lower particulate.

This configuration can allow the Laval nozzle to be installed even after the CAC if the intake air temperature is properly decreased.

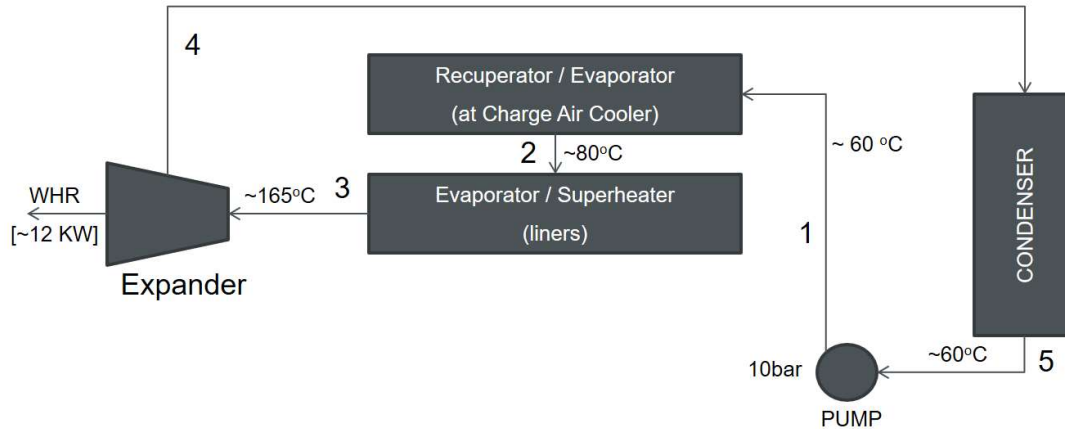
#### 5.4 (4) INDIRECT CHARGE AIR COOLER.

In order to increase the power source of the super heater, it is proposed to use a recuperator capable of recovering part of the energy from the charged air-cooling system. It would be possible to install an indirect CAC (liquid-air), which would also use ethanol as a working fluid in series with the liner's solution.

The i-CAC reduces pressure drop in charge air routing by up to 50% when compared to a direct CAC. The compact packaging reduces the overall volume of the charge air cooler in the intake system by up to two thirds. The increased heat capacity of the indirect cooling system reduces temperature spikes in transient operation, thus helping to limit transient NO<sub>x</sub> emissions. The low temperature radiator (in this case, the condenser in Figure 101) needs less packaging and accelerates heating, reducing cold start emissions (FLIK et al., 2009).

Liquid ethanol would be preheated in this recuperator (i-CAC) and would flow around the liners to achieve complete evaporation. Figure 101 shows the achievable Rankine cycle based on the simulations from section 3.4.2.

Figure 101: Heat rejection avoidance system (complete)



Source: My own authorship

Ethanol would replace conventional coolant (water and additives) as a working fluid. A CONDENSER would be present occupying the coolant radiator installation space. The RECUPERATOR / EVAPORATOR would be a heat exchanger (liquid x air) that would replace the conventional direct charge air cooler (air x air). The EVAPORATOR/SUPERHEATER around the engine heat sources would replace the complete engine coolant galleries.

Combining these features would lead to a more efficient system with a WHR potential shown in equation 13:

**Cold EGR + Liners superheater + Recuperator (old CAC) Liquid/air:**

$$\text{WHR potential} = [105\text{kW (radiator)} - 19.5\text{kW(EGR)}] + [23.4(\text{CAC}) + 10.4\text{kW(Cold\_EGR)}] = 119.3\text{kW} \quad (13)$$

Considering only 10% Rankine efficiency (normally 30 for conventional systems), this would mean 11.93 kW recuperation + 9.1kW (19.5-10.4) heat rejection avoidance, resulting in 21.4kW of extra power available for the system.

The integration of an indirect charge air cooler, could decrease even more the intake air temperature, what would allow the installation of the Laval Nozzle after the CAC as described in section 5.3.

**Cold EGR + Liners superheater + Recuperator (old CAC) Liquid/air + Exhaust WHR:**

$$\text{WHR potential} = [105\text{kW (radiator)} - 19.5\text{kW(EGR)}] + [23.4(\text{CAC}) + 4\text{kW(Cold\_EGR)}] = 104.9\text{kW} \quad (14)$$

Considering only 10% Rankine efficiency (normally 30 for conventional systems), this would mean a 10.49kW recuperation + 15.5kW (19.5-4) heat rejection avoidance, resulting in 26kW of extra available power to the system.

The condenser would be required to reject 83.97kW  $[105\text{kW}-19.5\text{kW}+10.4-(119.3*10\%)]$  with the cooling system WHR. If linked to a WHR exhaust system, only 79kW  $[105\text{kW}-19.5\text{kW}+4-(104.9*10\%)]$  would have to be rejected. Any improvement in the Rankine Cycle can improve energy savings and decrease condenser size.

## 6 CONCLUSION

As a conventional coolant system rejects  $23.86\text{kW/m}^2$  with water and coolant and the purposed evaporator is capable of extracting  $21.5\text{kW/m}^2$ , it has been demonstrated that the cooling system can be completely replaced by evaporators / superheaters surrounding the “hot spots” from an ICE.

Note that the proposed system and implementation steps may be cheaper than the old system. Even the expander cannot have a clear price because up until now there was no large-scale production and constructively, not much different from an air conditioning compressor, should not cost more than 50% of the known part.

The use of cleaner, lower temperature exhaust gases has also proved feasible and avoids significant heat rejections. Like the proposed WHR system, the implementation of the so-called “Cold EGR” shows potential cost savings when compared to HP-EGR.

These findings support further analysis and ensure that there are still many steps before stop using internal combustion engines. They can be the missing gap to reach the feasible TCO for the customer, for example. Even the objective tests depends on OEM investments, the presented simulations are enough to kick off further activities and represents important scientific contribution.

## 7 NEXT STEPS

- **Model improvements including cylinder head heat transfer;**

The simulation showed in section 3.4.2 represents the heat transferability from the Combustion chamber gases to the Engine crankcase. This heat transfer represents only part of the total energy transferred to the cooling system. The combustion gases flows also through the cylinder head channels before being expelled to the exhaust manifold. As both Crankcase and cylinder head are made from the same material (cast iron), the heat transferability can be extrapolated to the cylinder head with the same potential showed for the crankcase. Therefore, the heat transferability per square meter on the crankcase (see Figure 90 for reference) can be also considered for the Cylinder head (see Figure 91) and other sub components that eventually needs contact either with the coolant or with the working fluid, as purposed;

- **CFD simulation for the cold EGR using Laval Nozzle;**

All simulations done for the Laval Nozzle are based on compressible fluid stated theory. The calculations uses real data from 18.000 points in a original engine condition. The presence of a Laval nozzle is a calculated condition. If the resources are available, a CFD simulation could bring even more details from the system behavior. Different situations and Nozzle geometries could be simulated in addition to the throat, inlet, outlet and drain diameters.

- **Test bench update with an cold EGR and new engine calibration.**

After a complete CFD analysis, a prototype from the Laval nozzle can be built and installed in test bench as an add on component. The analysis of this single idea can bring the exact benefit of this development. The EGR cooler partially drains the exhaust gases. Since it is now available, The real benefit of it could be measured: in one hand, the Turbocharger has more energy available for recovery, but in the other hand, the Laval Nozzle would also bring some pressure drop to the fresh air line. This balance could not be simulated theoretically.

- **Simulation using both systems from section 2.5.3 and purposed system;**

Using data from the systems separately brought already many conclusions and proved feasibility of integration. Nevertheless, a complete simulation with all involved system could identify new opportunities and avoid errors before including the new features on the prototype vehicle.

- **Prototype update and vehicle measurements.**

Vehicle tests brings all side effects observed neither in simulations nor in test benches.

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## APENDIX A – Calculation sheet for laval nozzle simulation

The parameter described in Figure 55 (Measurement points from the ETC cycle) were measured 18.000 times during the complete cycle. The transient conditions represents a real operation. It means that the main conditions that the engine will be submitted in vehicle are simulated and considered on the simulations, which increases the representativeness of the calculations and boundary conditions. The image in this appendix highlights the excel used for the Laval Nozzle simulations. The Measured parameters are marked with M and the Calculated parameters are marked with C.

|                                   |     |                                   |     |                              |     |                             |     |                               |     |                              |     |
|-----------------------------------|-----|-----------------------------------|-----|------------------------------|-----|-----------------------------|-----|-------------------------------|-----|------------------------------|-----|
| %                                 | (M) | %                                 | (C) | kg/h                         | (M) | kg/s                        | (C) | kg/m <sup>3</sup>             | (C) | m/s                          | (C) |
| AGR Sollwert dynamisch            |     | AGR-Rate                          |     | Abgasmassenstrom             |     | Abgasmassenstrom            |     | Densidade                     |     | Velocidade                   |     |
| AGR_DYN                           |     | AGR_RATE                          |     | GA                           |     |                             |     |                               |     |                              |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Pa                                | (C) | kg/h                              | (M) | kg/s                         | (C) | kg/h                        | (M) | kg/s                          | (M) | kg/s                         | (M) |
| "Pressão dinâmica"                |     | AGR-Massflow                      |     | AGR-Massflow                 |     | Luftverbrauch               |     | Luftverbrauch                 |     | Luftverbrauch                |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Nm                                | (M) | bar                               | (M) | mbar                         | (M) | Pa                          | (C) | mbar                          | (M) | Pa                           | (C) |
| Istmoment (Welle gefiltert)       |     | Druck nach Verdichter             |     | Druck nach Turbine           |     | Druck nach Turbine          |     | Druck nach Schalldämpfer      |     | Druck nach Schalldämpfer     |     |
| MD                                |     | P2                                |     | P4                           |     | P4 A                        |     | P4_NSD                        |     | P4_NSD A                     |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| mbar                              | (M) | l/h                               | (M) | °C                           | (M) | °C                          | (M) | °C                            | (M) | °C                           | (M) |
| P_BARO                            |     | Q                                 |     | T3 AVERAGE                   |     | T3                          |     | T4                            |     | T4_NSD                       |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Em P2 (Ideal Gas)                 | (C) | °C                                | (M) | Pa                           | (C) | m <sup>3</sup> /s           | (M) | kg/s                          | (C) | m/s                          | (C) |
| [kg/m <sup>3</sup> ]              |     | Temperatur nach Verdichter        |     | Druck nach Verdichter        |     | Luftverbrauch               |     | Luftverbrauch                 |     | c <sub>2</sub>               |     |
| Densidade do Fresh Air            |     | T2                                |     | P2                           |     | Q <sub>2</sub>              |     | Q <sub>2</sub>                |     | Vel <sub>2</sub>             |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| P <sub>estagnação</sub> constante | (C) | T <sub>estagnação</sub> constante | (C) | h <sub>estagnação</sub>      | (C) | 10,90                       | (C) | ...                           |     | ...                          |     |
| Mach <sub>in</sub>                |     | P0 2                              |     | T0 2                         |     | H2                          |     | Q6                            |     | Delta P possível (Pascal)    |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| 11                                | (C) | Necessária p/ deltaP real         | (C) | 0,000166574                  | (C) | 0,04                        | (C) | ...                           |     | ...                          |     |
| Mach redução vent                 |     | Q2_5 VENTURI convencional         |     | UMITEY                       |     | delta p necessária          |     | Ain/A*                        |     | A*                           |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Vazão possível ponto a ponto      | (C) | Diâmetro da garganta necessário   | (C) | Vazão possível com escolhido | (C) | Pressão à jusante SUBSONICO | (C) | Pressão à jusante SUPERSONICO | (C) | ...                          |     |
| kg/s                              |     | mm                                |     | kg/s                         |     | P5 [bar]                    |     | P5 [bar]                      |     | ...                          |     |
| bloqueio individual               |     | Diâmetro crítico da garganta      |     | bloqueio escolhido           |     | fresh-block                 |     | ...                           |     | ...                          |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Temp_out                          | (C) | vel som local                     | (C) | vel local                    | (C) | delta P                     | (C) | Excesso de vazão              | (C) | Vazão na garganta            | (C) |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Abertura da válvula               | (C) | 1,01179931                        | (C) | Vazão NSD                    | (C) | REAL                        | (C) | ...                           |     | ...                          |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Tailpipe (Ideal Gas)              | (C) | Necessário em volume              | (C) | Em volume                    | (C) | ...                         |     | ...                           |     | ...                          |     |
| [kg/m <sup>3</sup> ]              |     | m <sup>3</sup> /s                 |     | m <sup>3</sup> /h            |     | ...                         |     | ...                           |     | ...                          |     |
| Densidade NSD                     |     | Vazão EGR no NSD                  |     | Proporção de NSD x Fresh air |     | ...                         |     | ...                           |     | ...                          |     |
| ...                               |     | ...                               |     | ...                          |     | ...                         |     | ...                           |     | ...                          |     |
| Em massa                          | (C) | Fresh + Gás                       | (C) | 1014,895799                  | (C) | Densidade Mistura           | (C) | 0,118551405                   | (C) | Volume                       | (C) |
| Kg/h                              |     |                                   |     | kg/h                         |     | [kg/m <sup>3</sup> ]        |     | m <sup>3</sup> /s             |     | %                            |     |
| Proporção de NSD x Fresh air      |     | T4_NSD_5                          |     | Vazão Fresh+EGR              |     | 4_NSD_5                     |     | Vazão Fresh+EGR               |     | aumento de volume no venturi |     |