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Tecnologie per il recupero energetico da motori a combustione interna

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Abstract

Modern diesel engines used for agricultural applications have peak brake thermal efficiencies in the range of 40-42% for high-load operation with substantially lower efficiencies at road-load conditions. Some energy and exergy analysis can be found in literature revealing that the largest losses from these engines are due to heat loss and combustion irreversibility, and any improvement in engine efficiency requires reducing or recovering these losses. Unfortunately, much of the energy losses either occurs at low temperatures resulting in large entropy generation (such as in the radiator), are transferred to low-temperature flows (such as the oil and engine coolant), or are directly wasted to environment through radiation or convection. While the opportunity of recovering part of these losses for heavy-duty applications have already been demonstrated, the potential efficiency improvements deriving from such a strategy for light-duty Diesel applications are unknown because of transient operation, the low thermal quality of exhaust gases at typical driving conditions, and the encumbrance of the added recovery system. We have used an experimental setup to perform a series of tests in order to investigate the potential for efficiency improvement through waste-heat recovery from the exhaust gas and the engine oil of a tractor Diesel engine. Results from steady-state and transient tests are presented, and the issues concerning waste heat recovery potential and its transient behavior are discussed for both engine oil and exhaust gas. An additional experimental setup, consisting of two heat exchangers placed respectively on the engine oil and exhaust gas, allowed to make some reflections about issues concerning the heat exchange, its inertia, and its influence over transient operations.

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Waste heat recovery for Diesel engines

1.1 Introduction

Climate changing due to greenhouse effect and pollution, both caused by human activities, are subjects that concern the human kind more and more. The agricultural sector is responsible of a wide variety of emissions causing environmental problems: CITEPA report (3) pointed out that 14.2% of nitrogen oxides, 14.5% of VOCs (Volatile Organic Compounds) and 50.3% of PM10 are connected to agricultural activities, and in particular to agricultural machinery. Greenhouse effect emissions of CO_2 have also to be taken into account: this contribution, even if small (2.8%), has been growing over the entire 20th century and has raised up by 167% since 1960[1].

However, environmental concerns are not the only reason for which further research on Diesel engines for agricultural machinery is needed. The raise in oil prices, for example, is directly connected to petrol and Diesel fuel cost to consumers: as figure 1.1 shows, oil price can change dramatically in a phew years because of political and economical circumstances, and its tendency has been constantly increasing since 1998. It is clear that, if prices are not going down in a phew years, fuel consumption too will become a major economic output in the economic balance of agricultural activities.

[1]Unfortunately, a part from the case of NO_x (62%), CITEPA report do not specify in what part Diesel engines are responsible of these emissions. Still, chosen data refers to the typical pollution coming from agricultural machinery. In the case of greenhouse emissions, in fact, only CO_2 was brought as example, since the largest contribution (methane emissions) almost entirely comes from husbandry

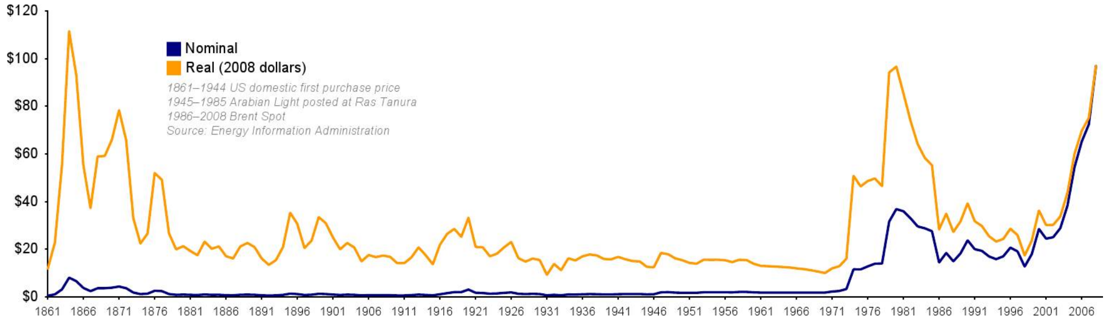

Figure 1.1: Oil prices tendance during recent history: in blue the official price, in orange the price updated with reference to inflation (1)

These facts drive to increase efforts toward pollutant emissions and fuel consumption reduction for agricultural machinery. Pollution-based issues can be solved by the use of technological solutions specially conceived to reduce emissions, such as NO_x catalyzed or non catalyzed reactors (SCR or SNCR), or filters for particulate matter. However, both pollution and fuel consumption can be reduced by increasing energetic performances of agricultural machineries, in particular with respect to Diesel engines.

1.2 Diesel engine

Invented at the end of XIXth century by Rudolf Diesel, a German engineer, the Diesel engine is an internal combustion engine.

Fuel ignition is the main feature distinguishing it from its petrol-fueled “brother”. Diesel engines, in fact, make use of a compression ignition rather than a spark one. In a gasoline-engine, air and fuel enter the cylinder together during the intake stroke. This prevent a large increase in pression, since high temperatures attained in the cylinder would cause fuel auto-ignition, thus leading to incomplete combustion. In Diesel engines, instead, air enters the cylinder alone during intake stroke, which means that no matter what pressure is attained during compression, there will not be auto-ignition. This leads to the possibility to attain much higher pressures, which directly implies a higher efficiency. Fuel is injected when the compression phase has almost completed and the air has attained a temperatures of about 700-900°C, enabling fuel to go through a rapid vaporisation followed by an almost instant combustion. The possibility of decreasing vaporisation time and enhancing air-fuel mixing in combustion chamber have been, since Diesel engine invention, the most researched fields, and results of engineers efforts led to a large increase in Diesel performances.

As said, Diesel engine has a higher efficiency than its gasoline brother, even using a lower quality fuel. However, higher pressures imply a thicker structure and, as a consequence, heavier components and lower rotating speed. These features, typical of old Diesel engines, restricted their use first to marine and railway applications, and than to large vehicles such as trucks and tractors. Furthermore, pollution connected to particulate matter emissions caused by incomplete combustion has always been a primary issue for Diesel engines, and its solution is coming to light only in last years.

Turbocharger is maybe the most important innovation in Diesel engine technology. It consists of a gas turbine recovering exhaust gas thermal and pressure energy content, and of a compressor bringing inlet air to a higher pressure level, thus increasing engine power–to–weight ratio. This invention, together with the increase in atomisation in fuel injection, allowed a strong enhancement in Diesel engine performances. Engine weight reduction and a faster response to user's instructions thus allowed to spread its application to the enormous market of private cars.

1.3 Waste heat recovery

Still, Diesel engine is far from perfection. Researchers are still working hard in order to increase its performances.

In the context of the reduction of fuel consumption, waste heat recovery (WHR) is considered to be a possible strategy to achieve this result.

The second principle of thermodynamics states that every engine or thermodynamic cycle converting the thermal energy of a heat source into work must waste a certain amount of heat to its environment. Once the hot source and the renvironment temperatures are fixed, the Carnot efficiency allows to estimate the maximum fraction of work that can be extracted from a given quantity of heat.

However, nowadays, engines and, more generally, devices for energy conversion performances are far away from Carnot efficiency. This means that, very simply, a lot of the available heat that could be converted into useful work is instead wasted.

In a Diesel engine there are several sources of waste heat: exhaust gases, engine oil, transmission fluid, cooling water, intercooler, EGR, etc; several authors, such as

Bourhis et al. (4), Edwards et al.(2), Junhong et al. (5), Gopal et al. (6), Srinisvan et al. (7), have already studied the possibilty of recovering a part of the wasted heat, concluding that WHR for Diesel engines is a possibility that is worthed for further inspection.

Before starting the project of a WHR system it is important to estimate the recovery potential of each of the available sources, in order to understand if some of them are more energetically and economically convenient to be exploited than others. But before starting with measures and calculations, a small summary should be made about the WHR sources that are to be taken into account.

Exhaust gas is normally the most valuable source of waste heat. In a Diesel engine combustion takes place inside the cylinder, and energy released pushes the piston up to the end of the cylinder. When the engine movement pulls back the piston, reducing combustion chamber dimensions, it drives exhaust gas through the exhaust valve out of the cylinder.

In a basic Diesel engine, the gas runs directly toward the exit of the exhaust pipe. However, in modern applications, a phew systems can be placed between the cylinder and the exit, such as pollution-reducing equipments. Anyway, no matter what is placed before its exit (in conventional engines), exhaust gas is still far from a state of thermal and mechanical equilibrium with environment, which means that a certain quantity of its energy content could still be recovered.

Exhaust gas outlet temperature, pressure and mass flow can widely change as a function of engine type, equipments and load factor. Just as a general consideration, the higher the engine efficiency, the lower its exhaust gas temperature.

As well as every industrial and technological applications, Diesel engines are provided with several secondary system, whose nature and properties depend on engine type. However, for what concern this study, every Diesel engine is provided with a lubricating oil circuit.[1]

Engine oil is stocked in a vessel called “carter”, placed under the engine. A pump takes the oil from the carter and drives it throughout a circuit eventually spraying it over moving parts, in order to lubricate and cool them. Engine oil temperature can

[1]In standard applications, cooling liquid is not pure water but a mix of water and glycol, the objective of the latter addition being to give antifreeze capabilities to the fluid. However, from now, water-glycol mixture will be referred to simply as water

reach very different values depending on engine type and operation, but generally it does not exceed 130°C, while its normal temperature stays around 110°C at full load operations. Oil cooling is assured by a heat exchanger placed on the water circuit.

1.4 Specific features of agricultural applications

As every technical application, Diesel engines for agricultural machinery have their own typical features, which have to be well known in order to understand the issues they are connected to.

The most typical feature of a tractor is the fact that, in several applications, power consumption is not strictly related to tractor displacement. During working operation, in fact, most of power absorption is related to the working equipment pulled by the tractor. This is possible thanks to a second shaft, directly connected to the engine and coaxial with tractor shaft, that transfers mechanical energy from the engine to the agricultural equipment. The two shaft movements are normally independent, which means that it is possible for the tractor to move while the equipment is not working, as well as the latter to do his job while the tractor is not moving. The shaft connection between the tractor and its equipment is generally placed beside the vehicle and it is called "power take-off" (PTO).

This feature is important in order to understand engine's transient behavior.

One of tractors specific features is, in fact, their particular kind of transient operations. Tractors are generally used alternatively for two different goals, each having its specific features:

- On the field, for pulling and giving energy to agricultural equipments
- On the road or on the field, for transporting goods

The main difference lays clearly in load factor variation with time. In fact, normal operations of agricultural machines on the field involve a periodical variation for engine load:

- Full-load operation, when tractor is moving with its equipment working
- Partial-load operation, typically in maneuvering operations, when equipment is not working

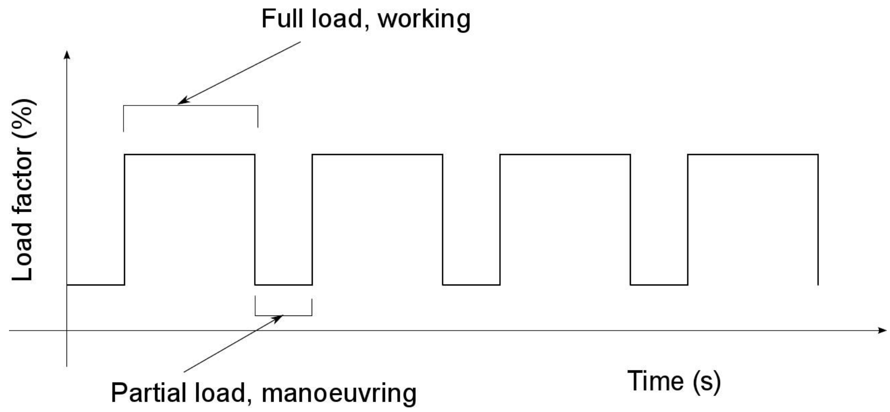

Figure 1.2: Sample of load factor evolution over a simulated on-field operation cycle

This is schematically shown in figure 1.2. However, this representation is not completely veritable. In fact, even if full load conditions are well represented by a constant-load behavior, the same cannot be said for maneuvering operations, where evidence is shown of a very unsteady and unpredictable behavior. Concerning road transportation instead, load behavior is much more variable, more similar to that of cars. For such cases an example of engine speed changing with time, taken from Edwards' paper (2), is shown in figure 1.3

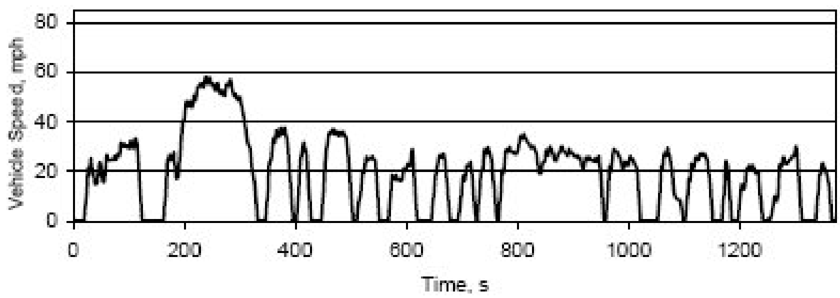

Figure 1.3: Vehicle speed over a simulated warm UDDS drive cycle (2)

The two different components of standard real-life operations can mix in very different ways, depending on several variables such as field dimensions, kind of crop, and many more.

These features make agricultural machines behavior in transient operation very different from other applications employing Diesel engines, such as cars, trucks, ships and trains. This reasoning leads to the idea driving this study; Diesel steady-state behavior do not change dramatically between different applications, being mainly related to some general features that could normally make possible to compare car, truck and tractor engines. The typical features in real-life operations are what really makes the difference, involving variable load conditions and a strong influence of transient behavior

It is therefore important to know what is the transients impact on WHR potential, in order to make a correct choice of all system parameters[1].

1.5 Influence of heat exchange properties on recovery potential

Heat recovery potential has been defined, until now, as the energy content of some sources of heat dispersion to the environment for an engine. However, even if the technological side of this matter has still to be treated, there is one first and important issue that need to be considered in order to give a correct evaluation of the real recovery potential of a heat source. Most of WHR strategies in fact need to transfer the energy content of a heat source to a new fluid, specifically meant for working in a recovery cycle. The matter of waste heat recovery is therefore directly connected to the subject of heat exchange properties of different fluids and materials.

In the case of WHR recovery for a Diesel engine, this matter comes strongly to the point. It is well known in fact (and it will be proved in the following chapters of this work) that exhaust gas is the most valuable source of waste energy, both for quality and quantity, and its high temperature makes it a much stronger candidate than engine oil for waste heat recovery[2]. However, it is likewise well known that heat transfer properties are not the same for liquid and gas flows, the latter having a much lower attitude in heat exchanging. Exhaust gas, therefore, generally require higher exchange surfaces in order to recovery their energy content.

This reasoning lead to the need of studying the subject of heat exchange with deeper interest. In static applications, in fact, the only constraint is generally related to economic-related issues, since the larger the heat exchanger the more expensive it

[1] a strong influence of transient duration could, for example, be solved using a high thermal capacity fluid for an intermediary cycle in order to increase system inertia

[2] As will be further explained in the following chapter, in fact, higher temperatures directly lead to a higher potential in work production

is.[1] For moving applications, instead, other issues become important, mainly related to exchanger's weight and size. On a vehicle, in fact, room available is limited, and increasing the total weight has a direct influence on its performances.

This is why two heat exchangers (whose features will better described further ahead) have been inserted, the hot flow being either the engine oil or the exhaust gas circuits, and the cold flow being water coming from aqueduct. For both of them the surface and the main features of the exchanger were kept, in order to limit the analysis to each flow heat exchange properties. Measuring the temperature difference over the heat exchanger and the mass flow on both sides allows to have an experimental knowledge of heat exchange properties of both exhaust gas and engine oil, thus being able to insert these data to the rest of the analysis.

[1]This is a general statement that largely simplifies the complex subject of heat exchanger choice. Lots of other parameters, such as the need of a pumping system, the attitude of fouling and, in general, lowering heat exchange properties with use, and many more, have an influence on the choice and the price of a heat exchanger

2

Exergy

2.1 First and second law of thermodynamics

2.1.1 First law of thermodynamics: energy efficiency

Since their first invention, systems for energy conversion have gone through an enormous growth in their use across all the industrialized countries. From steam engines to modern gas turbines, the possibility to produce useful work by burning specific substances has dramatically spread, both in terms of quantity and variety of applications. This discovery is among the basis of the modern society in which we all live.

Nevertheless, especially during last years, an increasing concern regarding climate changes, fossil fuel depletion, and, in general, sustainable development, led to spend more and more resources in order to improve these systems energy performances, thus cutting down their fuel consumption.

Historically, energy analysis has always been the most used strategy for estimating engines performances. It leads to the energy efficiency, a value that allows to estimate the amount of useful energy produced by a system compared to its primary energy consumption.

Energy analysis comes from a balance between inlet and outlet energy flow in a well-defined system. The first law of thermodynamics says in fact that energy cannot be created or destructed, but only exchanged or transformed. The following is an example of a balance equation for an open system:

$$(e_p + e_k + h)_i \dot{m}_i + (e_p + e_k + h)_o \dot{m}_o + \sum_j \dot{Q}_j - \dot{W} = 0 \tag{2.1}$$

where e_p is gravity potential energy, e_k is kinetic energy end h is enthalpy[1] [2].

A first possible definition for energy efficiency can be obtained by considering every inlet quantity as an energy source (putting them at denominator) and every outlet quantity as an energy product (putting them at numerator). We can in this case get to the following equation:

$$\eta_{en} = \frac{\dot{W} + (e_p + e_k + h)_o \dot{m}_o}{(e_p + e_k + h)_i \dot{m}_i + \sum_j \dot{Q}_j} \tag{2.2}$$

In most part of technical applications for energy production, though, these hypothesis can be made:

- Mechanical work is the only useful product. In most of application, in fact, the outlet flow has only a small amount of energy left, it is just rejected to the environment. and it is not considered to be a useful product[3]
- Variations in gravity potential energy and kinetic energy can normally be neglected [4]
- The system is supposed to be adiabatic, since this is often a good approximation that allows to simplify calculations

These hypothesis lead to the following expression for energy efficiency:

$$\eta_{en} = \frac{\dot{W}}{h_i \dot{m}_i} \tag{2.3}$$

Equation 2.3 is used when a system is being considered in his integrity, where the inlet flow is a fluid with high energy potential, such as a fuel or hot steam. However, if we want to focus on cycle efficiency and there is no need to give a definition of the

[1] Every thermodynamic property mentioned above is written in minuscule letters in order to represent specific properties, as conventionally used

[2] In this equation the thermodynamic convention was adopted by considering as positive quantities the work produced by the system and the heat received by the system

[3] This is the case, for exemple, of exhaust gas of internal combustion engines, where combustion products leaving the cylinders are of no use and rejected in atmosphere. Some particular engines, such as plane gas turbine, do not match with this statement since, in this case, the exhaust gas kinetic energy is the useful product

[4] They become relevant only in some specific applications, such as in hydraulic turbines or in turbogas jet engines

heat source, a generic energy inlet (usually in the form of heat) can be considered. In this case energy efficiency is given by:

$$\eta_{en} = \frac{\dot{W}}{\dot{Q}_1} \tag{2.4}$$

where $\dot{Q}_1$ is the amount of heat pumped into the system per time unit.

Finally, energy efficiency gives very valuable information, quantifying what part of available energy can be transformed into useful energy [1].

2.1.2 Second law of thermodynamics: entropy

The energy analysis is interesting but, often, incomplete. The first law of thermodynamics gives in fact information about quantities in energy exchanges, but does not show in which direction they naturally take place. As Mago says, "*thermodynamics explain us that different kinds of energy are not the same*"(9). This statement drives to look for other means to more effectively evaluate energy systems performances. What is really needed, in fact, is a way for taking into account of the difference in quality between different energy forms.

As said before, in fact, energy cannot be neither created nor destructed, but it can be exchanged and transformed among its different forms. For example, it is possible to get work from potential energy through an hydraulic turbine, or to have thermal energy from kinetic energy through a fan, etc.; it is only necessary to use the correct conversion system.

However, a form of energy exist that is different from others: thermal energy. Like every other form of energy, such as mechanical, kinetic, electric etc., thermal energy can be associated, at microscopic level, to particles movement. But unlike any other form, thermal energy movement is disorganized.

Thus, transforming thermal energy into any other energy form means converting a disorganized movement into an organized one (that is what take place, for example, in a turbine). This process lead to significant losses, that are directly connected

[1] Generally speaking, the most correct way to write the equation for energy efficiency is to put at numerator what is considered to be a valuable product, and at denominator what is considered to be a valuable resource. Lior (8) makes reference to a more specific definition, where he considers as a product everything that has an economic value and as a resource everything which has to by paid for. However, this definition is very useful for an economic estimation of a system, but do not give enough information when we are interested in thermodynamic performances

to the conversion process, which means that there is no way to get rid of them. A 100% conversion from thermal energy to, for example, mechanical energy is simply not achievable, even with ideal reversible engines.

This feature of thermal energy is resumed by the second law of thermodynamics; the Kelvin statement says that "*No process is possible in which the sole result is the absorption of heat from a reservoir and its complete conversion into work*".[1].

The equation defining the second law involves the use of a new thermodynamic property: entropy, which is a measure of disorder at molecular level in a system:

$$dS = \sum_j \frac{\delta Q_j}{T_j} + \delta S \tag{2.5}$$

It is easy to notice that entropy, unlike energy, is not conserved. Every process leads to a creation of entropy (that is an increase in disorder of the system) that can be equal to 0 in the best and ideal case of reversible processes. This can lead to other statements of the second law, which say that "*in an isolated system entropy can only increase*" or, more dramatically "*entropy of universe is increasing*".

As can be seen in equation 2.5, entropy is a quantity strictly connected to heat exchanges. There are no modifications in entropy when heat is not involved. Work, for example, involves only an organized particle movement, that as no influence on the disorder of a system and, consequently, on its entropy.

The second law of thermodynamics establish a difference in quality between thermal energy and other energy forms, explaining why some processes can take place naturally while others cannot. Temperature is the thermodynamic property of matter that is of particular interest in this context. It measures thermal equilibrium, that is the ability to determine which of two or more systems will experience an increase in disorder level if they come to the condition of being able to exchange energy.

Being capable to effectively use entropy could be very helpful in energy system analysis, since it is able to quantify the irreversible effects causing efficiency losses. However, entropy concept is quite difficult to manipulate in practical applications, since it is a rather abstract property and, what is more important, it is not directly connected to energy, that is actually what interest us the most.

[1] The other original statement, the Clausius one, says that "*No process is possible whose sole result is the transfer of heat from a body of lower temperature to a body of higher temperature*". The equivalence of the two statement can be demonstrated(10). In this study the Kelvin statement was chosen because it is directly related to the subject

2.2 Combining First and second law together: exergy

2.2.1 Exergy definition

In previous paragraphs, the two laws of thermodynamics have been introduced. Their content can be summarized as following:

- The first law says that "*You can't get something for nothing* (11) (Conservation of energy)
- The second law says that "*You can't even get all you pay for* (11) (Entropy creation)

A question arises: if you cannot get all you pay for, how much can you actually get?

What we are looking for is a thermodynamic property, such as entropy or enthalpy, allowing to estimate the potential to produce work from a given quantity of energy in the light of the second law.

This property is called **exergy**, defined as "*The maximum theoretical work that can be extracted from a given entity when it is brought to equilibrium with its environment*"

One of most important exergy features that arises from this definition is that the exergy content of a certain entity is not only connected to entity itself, but also to its environment. In his paper about exergy, Scott (12) gives a very simple but clear example: taking a cylinder full of air at atmospheric pressure and putting it anywhere on earth, its exergy content will be equal to 0. But if the same cylinder is brought on the moon or deep under the sea, even if its energy has not changed, its exergy has dramatically increased. Others examples, related to temperature, can be a glass of hot water or an ice cube: they both have way different exergy contents in summer and winter, which means depending on the environment state the system is surrounded by.

Therefore, exergy can also be seen as a measure of departure from environment. The higher the difference of temperature, pressure, altitude, velocity etc. between the system and its environment, the higher its exergy content.

Exergy can be very useful to estimate the performances of several industrial systems, since it allows to compare different forms of energy, and to measure system's efficiency with reference to the maximum obtainable value. Several authors, such as

Rosen (13),(14),(15), Dincer (14), and Bejan (16), developed and used exergy analysis for this goal.

Time as come to pass to equations. Rosen (13) and Bejan (16) give a very proper demonstration of exergy concepts, that will be here reported.

Starting from balance equations for energy and entropy:

$$h_i\dot{m}_i - h_o\dot{m}_o + \sum_i \dot{Q}_j - \dot{W} = 0 \tag{2.6}$$

$$s_i\dot{m}_i - s_o\dot{m}_o + \sum_j \frac{\dot{Q}_j}{T_j} + \dot{S} = 0 \tag{2.7}$$

the contribution of heat exchange with environment Q_0 can be obtained from equation 2.7:

$$\dot{Q}_0 = T_0[s_o\dot{m}_o - s_i\dot{m}_i - \sum_{j\neq 0} \frac{\dot{Q}_j}{T_j} - \dot{S}]$$

This expression can be inserted into the energy balance equation:

$$\begin{aligned}\dot{W} &= h_i\dot{m}_i - h_o\dot{m}_o + \sum_{j\neq 0} \dot{Q}_j + T_0[s_o\dot{m}_o - s_i\dot{m}_i - \sum_{j\neq 0} \frac{\dot{Q}_j}{T_j} - \dot{S}] \\ &= (h - T_0 s)_i\dot{m}_i - (h - T_0 s)_o\dot{m}_o + \sum_{j\neq 0} \dot{Q}_j(1 - \frac{T_0}{T_j}) - T_0\dot{S}\end{aligned}$$

This expression can be simplified thanks to the following reasoning:

- As maximum work is the value we are looking for, reversible conditions have to be considered and thus entropy generation should be equal to 0
- Outlet state is the state of equilibrium with environment, as settled by exergy definition
- Since we are looking for the work that can be extracted from a generic mass flow, the system should be considered to be adiabatic
- The system is in steady state, which means that $\dot{m}_i = \dot{m}_o$

These hypothesis lead to the following expression:

$$\dot{W}_{max} = (h - T_0 s) - (h - T_0 s)_0$$

That is:

$$\dot{ex} = (h - h_0) - T_0(s - s_0) \tag{2.8}$$

This simple demonstration shows a very interesting aspect of energy. If only work is introduced into a system, exergy changes as following:

$$\Delta ex = W$$

If a given quantity of heat is introduced instead, exergy change is given by:

$$\Delta ex = Q_j(1 - \frac{T_0}{T_j})$$

where T_j is the temperature of the heat exchange Q_j. The asymmetry between heat and work, as well as between thermal and mechanical energy, is here quantified. In this sense, exergy put together energy and entropy concepts: while the first gives information about quantity, but not quality and the second does the opposite, exergy is able to quantify energy quantity with an eye kept on its quality. Exergy analysis also explains, once again, that even the most effective thermal engine cannot convert all the energy content of a given quantity of thermal energy into work. A part of it is always lost, despite engine quality, and it is called anergy.

2.2.2 Maximum work potential: connection between energy and Carnot cycle

In classical thermodynamics maximum work potential is estimated using a Carnot cycle. However, in this case, this is not directly possible. In fact, since the goal is to evaluate the work potential of a given quantity of matter, this will necessarily involve a thermal exchange leading to an evolution in fluid temperature, which is incompatible with a Carnot cycle.

Nevertheless, these two concepts must have something in common, since they are supposed to measure analogous quantities. That is what in fact happens, and this connection will be explained in the following paragraphs.

Consider a thermal exchange between two different flows. It is possible to suppose to divide the exchange in several smaller ones, each of them at a constant temperature (Fig. 2.1). Each of these exchanges can be considered as a Carnot cycle between exchange temperature and ambient temperature. Measuring the surface of the cycle

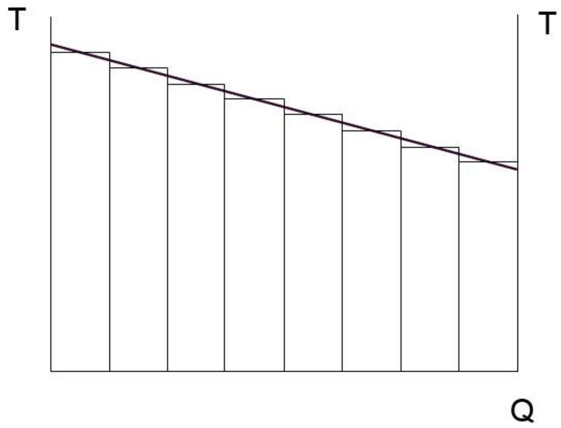

Figure 2.1: Sub-division of a thermal exchange in smaller exchanges at constant temperature

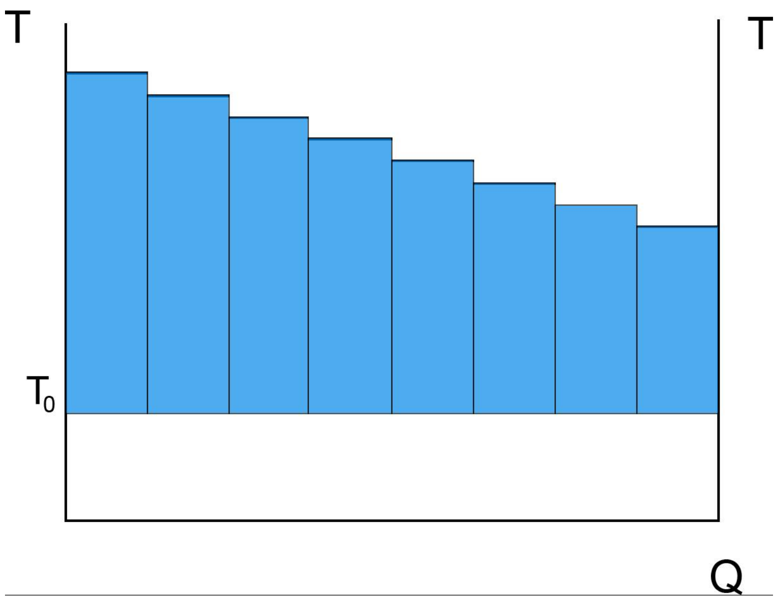

Figure 2.2: Equivalent Carnot cycles

lead to an estimation of the amount of work that could have been harvested by the Carnot cycle (Fig. 2.2). Therefore, this value represents the work production potential (exergy) lost by the hot flow because of the heat exchange. So:

$$ex_{l,j} = \eta_{Carnot,j}\delta Q = (1 - \frac{T_0}{T_j})T_j\Delta s$$

where $\eta_{Carnot,j}$, T_j, and $ex_{l,j}$ are respectively the Carnot efficiency, the temperature and the exergy loss of the j-heat exchange. The total of exergy loss caused by heat exchange value is:

$$ex_l = \sum_j ex_{l,j} = \sum_j (1 - \frac{T_0}{T_j})T_j\Delta s$$

If the heat exchange subdivision is brought to an infinite number of smaller cycles, an integral expression is obtained:

$$ex_l = \int_A^B (1 - \frac{T_0}{T})T ds$$

that is:

$$ex_l = Q - T_0(s_B - s_A)$$

Given that a heat exchange is being considered, where the only way for energy transfer is through heat exchanges, the result is the following:

$$ex_l = (h_A - h_B) - T_0(s_A - s_B) \tag{2.9}$$

This is the same expression as 2.8, obtained thanks to the estimation of the work of a Carnot cycle, thus making evidence of connection its with exergy.

2.2.3 Entropic temperature

The Carnot cycle is composed by 4 processes:

- Isentropic compression
- Isothermal heating
- Isentropic expansion
- Isothermal cooling

Of course, the Carnot cycle is ideal. Performing an isentropic compression or expansion means having access to totally adiabatic components with no friction losses. It is to be noted, however, that these restraints are connected to technological imperfection. Isothermal heat exchanges, instead, are way more difficult to achieve. To make them possible, it has to be that:

- The flow has an infinite thermal capacity
- Flow pressure decreases in order to keep temperature constant
- Heat exchange takes place during phase transition

Having an infinite thermal capacity is clearly not possible[1], and being able to decrease pressure precisely enough to keep a constant exchange temperature is not only difficult, but also unseemly. Instead, several industrial applications make use of transition-phase heat exchanges, for different reasons; however, this is only a particular case, that cannot be taken as the general one.

Therefore, in general, heat exchanges lead to temperature variations that do not fit to a Carnot cycle. Therefore Carnot efficiency cannot simply be used as a reference of maximum efficiency, since it is needs to define precise temperature values for heat exchanges, that are not well defined in common cycles.

However, it is possible to find two equivalent temperatures, called entropic temperatures. The cycle resulting from this choice must respect two restraints face to the original cycle:

- Same difference in entropy
- Same amount of heat injected into the cycle
- Same amount of heat rejected by the cycle

To make this possible, each entropic temperature must be equal to:

$$T_{en} = \frac{Q}{\Delta s}$$

[1]having a very high thermal capacity face to the heat exchanged is indeed possible, but rarely useful in industrial applications

It is therefore possible to have a maximum and a minimum entropic temperature (called respectively $T_{en,max}$ and $T_{en,min}$) permitting to evaluate an equivalent Carnot efficiency as following:

$$\eta = Q(1 - \frac{T_{en,min}}{T_{en,max}})$$

where Q is the amount of heat injected into the cycle.

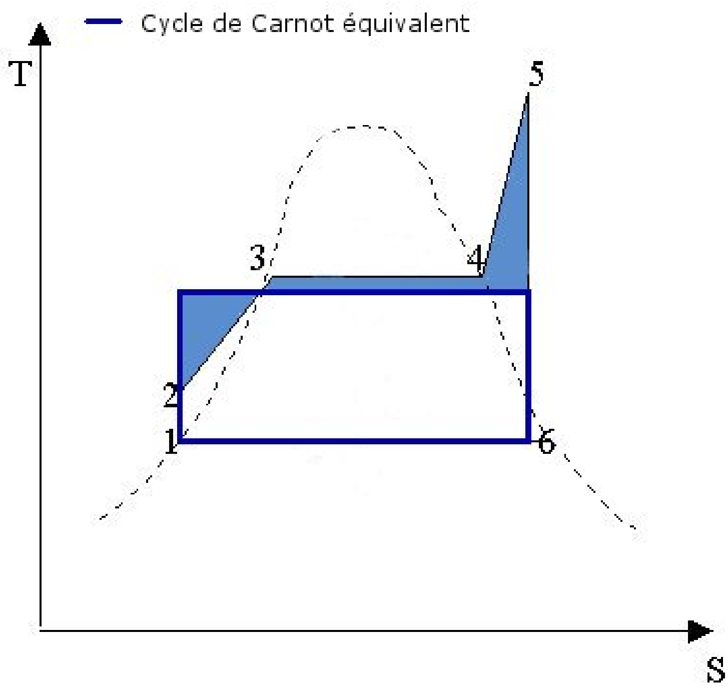

Figure 2.3: Example of an equivalent Carnot cycle for a Hirn cycle

This concept can be applied to exergy analysis. As already said, in fact, the exergy content of a given quantity of matter in a given state is the maximum work that can be harvested by bringing it to equilibrium with its environment. This can be seen as nothing more than a thermodynamic cycle between the initial state (1) and the equilibrium state (0). As every thermodynamic cycle, it can be associated with an equivalent Carnot cycle between two entropic temperatures. Given that exergy is the maximum amount of work that can be harvested, just like work produced by a Carnot cycle, this expression is obtained:

$$ex = (h - h_0)(1 - \frac{T_0}{T_{en,max}}) \tag{2.10}$$

It easy to notice that this expression leads to the same result of equation 2.8. This can be seen by performing the calculation of entropic temperature in the ideal gas hypothesis. This leads to:

$$
\begin{aligned}
T_{en} &= \frac{Q}{\Delta s} = \\
&= \frac{c_p(T - T_0)}{c_p \ln \frac{T}{T_0}} = \\
&= \frac{(T - T_0)}{\ln \frac{T}{T_0}}
\end{aligned}
$$

that, when subsituted in equation 2.10, gives exactly the same result as equation 2.8.

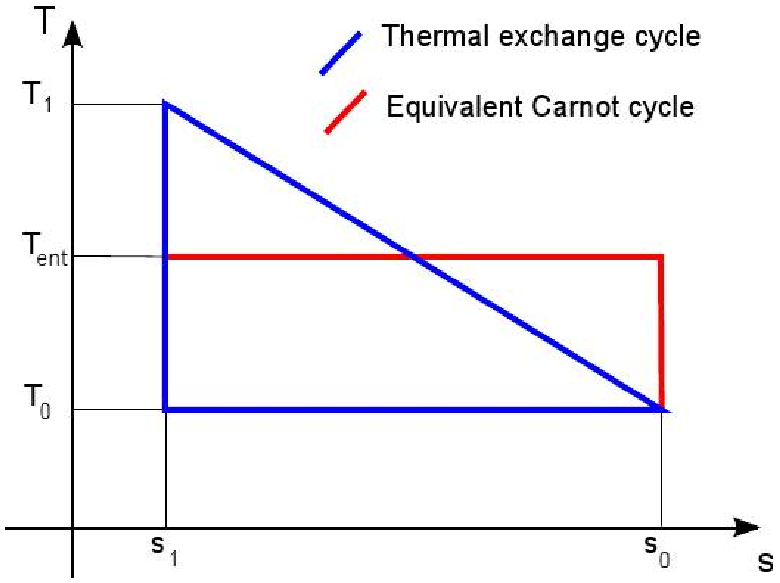

Figure 2.4: T,s diagram showing the comparaison between the exergy content and the Carnot cycle associated to the entropic temperature of the cycle

In figure 2.4 the comparison between the two concepts is shown, highlighting the cycles' output work. In figure 2.5 an example of an ideal cycle made for exergy estimation is shown. This model, anyway, is only valid where state 1 is at the same pressure as its environment, since evidently a heat exchanger is not able to recovery pressure differences. In this case exergy is calculated as the work produced by a Carnot engine working between ambient temperature and entropic temperature of the thermal exchanges bringing the considered mass flow from its original state to thermal equilibrium with environment.

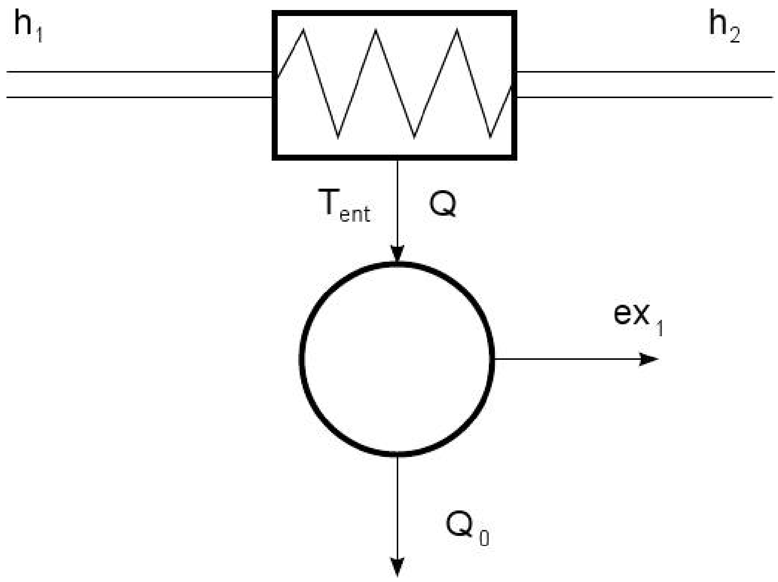

Figure 2.5: Cycle diagram showing the role of entropic temperature in exergy calculation

Therefore, entropic temperature can be a good system to estimate quality and quantity of energy, with respect to the second law, just as exergy is. Entropic temperature can be very useful because it is easier to understand and it allows to make a more direct comparaison between different heat sources.

2.3 Exergy analysis

In previous paragraphs exergy has been introduced, giving its definition, explaining its meaning, and showing why other thermodynamic properties, such as energy and entropy, are adequate for estimating energy quantity and quality at the same time. Exergy, as said before, is helpful for making comparisons between thermal energy and other forms with respect to their quality.

However, one important remark need to be made. Exergy is not the ultimate solution for estimating energy quality, and its use needs to be restrained with respect to its limits.

2.3.1 Limits to exergy use

Firstly, exergy is strictly connected to heat exchanges and thermal energy. It allows to compare them with work and mechanical energy (as well as with other energy forms such as electrical or kinetic) with respect to their difference in quality. This means that if a process makes no use of thermal energy, exergy analysis will not give any further information than a simple energy analysis. This can be the case of power plants such as hydro-electrical systems or wind turbines, where no heat exchange is involved. A good example of this issue is shown in Dincer's paper entitled "Energy and exergy utilization in transportation sector of Saudi Arabia"(17). In this case kinetic energy is the final product which is considered in the exergy analysis, where the chemical energy of the fuel is the inlet flow. Exergy and energy have identical values in both cases, leading to identical results for energy and exergy efficiency. Therefore, in this case, there is no point in using exergy efficiency. More interesting results would be obtained in the case of a thermodynamic study of engines: in this case, heat transfers come to light and an exergy analysis reveal interesting aspects of the system.[1]

Secondly, the fundamental principle of exergy analysis is the major value given to work as process consumption or product. This statement, that thermodynamically speaking is always true, can drive to incorrect conclusion from the technological and economical point of view. In fact, in the industrial sector, several processes only need heat at a certain temperature as process utility, which means that there is no point in comparing product quality with work potential. A very good example is given by house heating sector. If only the system for hot water production is considered, it is clear that a conventional boiler has an efficiency way higher than a heat pump, since it is theoretically capable to produce a hot stream at 300-400°C. But if the whole heating system is considered, in particular the final part, it is also clear that a system combining a heat pump with radiant panels can be way more effective than a conventional boiler provided with a radiator system. In this case, in fact, even if the exergy of the outlet flow is different, both systems manage to achieve the same result, and only inlet exergy consumption should be taken into account.

Thirdly, a very important aspect that should be taken into account is the choice of the environment state. A precise and proper definition of the environment state is very important. When considering heating systems, for instance, the fact that external

[1]The only difference in Dincer paper lies in the γ coefficient, representative of the ratio between HHV (Higher Heating Value) and LHV (Lower Heating Value) of considered fuel. This is simply due to different conventions: in energy analysis the energy content of a fuel is represented by the LHV, while in exergy analysis it is measured by the HHV

temperature substantially changes throughout the year must be taken into account. Bibliography gives important examples of how results become very different whether a correct analysis of the environment state is performed or not. In Dincer's paper entitled "Energy and exergy use in public and private sector of Saudi Arabia" (18) a very low exergetic efficiency is associated to the air conditioning sector. The issue here is strongly connected to the choice of the environment state: choosing a fixed temperature of 10°C, as Dincer does with reference to previous papers, leads to a misunderstanding, since Saudi Arabia's climate features much higher temperatures. This reasoning implies that the need of air conditioning at 20°C is connected to a "cold" exergy rather than a "hot" one. Choosing 10°C as reference temperature means that a flow of cold air at 20°C as a lower exergy content than one at 30°C, while the real condition is the opposite. A way more precise approach is used in Wall's paper "Exergy conversion in the Swedish society" (19). In this case the author develops a quite simple model for estimating external temperature depending on the season and the moment of the day. This tool allows a very precise evaluation of heating system exergy content, that as said in previous sections strongly depends on the environment state. Le Goff et al. (20) go into this subject in depth, explaining the existence of several strategies for defining environment temperature and comparing their advantages and disadvantages. What is pointed out in this latter paper, is that the maximum efficiency of a generic energy system do not depend on its environment. This concept allows, however, to estimate maximum efficiency, while real efficiency depends, in fact, on environmental conditions, as pointed out in previous paragraphs.

2.3.2 Exergy analysis domains

It is therefore important to identify what domains exergy analysis is more suitable for.

One of the analysis that could take most advantage of the exergy application is the study of systems energy consumption. In this case, in fact, we have often to face with complex processes that need different energy sources, such as electrical, thermal and mechanical energy, and a simple energy-based analysis of these different needs would be unfair. Even if this domain has been largely studied in the past and in present years, Szargut work "Exergy analysis of thermal, chemical, and metallurgical processes" (21) still remains the reference study. The main feature of this specific analysis is that major attention is pointed on exergy consumption, where the goal is to minimize it for a given product. In this case, thus, the exergy content of entries rather than of products is generally taken into account.

Another possible use for exergy analysis comes from a secondary definition. Exergy is in fact also representative of the “departure from environmental state”, and several authors, such as Rosen, Dincer (22) and Lior (8), make reference to the use of exergy in environmental impact analysis. This concept can be very useful when considering thermal pollution, but also (considering the chemical part of exergy), for evaluating pollutants reactivity and its impact on the environment. In this case, however, it is important to notice that only the energy aspect is taken into account, while there is no mention of toxicity or other process by-products features. This means that this analysis, even if correct and useful, remains incomplete.

2.3.3 Exergy analysis for Waste Heat Recovery

In this study attention will be concentrated on exergy analysis applied to the evaluation of recovery potential in WHR systems where additional work is required. This is maybe the domain exergy analysis is best fit for, since it allows an estimation of maximum work potential of a current, which is exactly what has to be evaluated to define the efficiency of this kind of WHR systems.

In fact, several authors followed this path. Concentrating on WHR from Diesel engines, which is the subject of this study, Aly (23), Bourhis et al.(4), and Edwards et al.(2) made use of exergy analysis, pointing out the differences with reference to the energy analysis, in this field.

In figure 2.6, taken from Edwards' paper (2), these differences become clear. What is easy to notice is that exergy recovery potential is much lower than energy one (In this study the author chose to take into account only EGR and exhaust gas recovery potential, without any interest in engine oil). This result becomes obvious under the light of previous sections: the second law of thermodynamics clearly explains that there is no way to convert all heat into work, and that the portion that can definitely be converted depends on flow temperature and is quantified by exergy. In Edwards' paper, for instance, it can be seen that the 69 kW of energy output in form of work keeps the same value in exergy analysis, while the 43 kW of energy output in form of waste heat (EGR and exhaust gas) become only 15 kW of exergy flow.

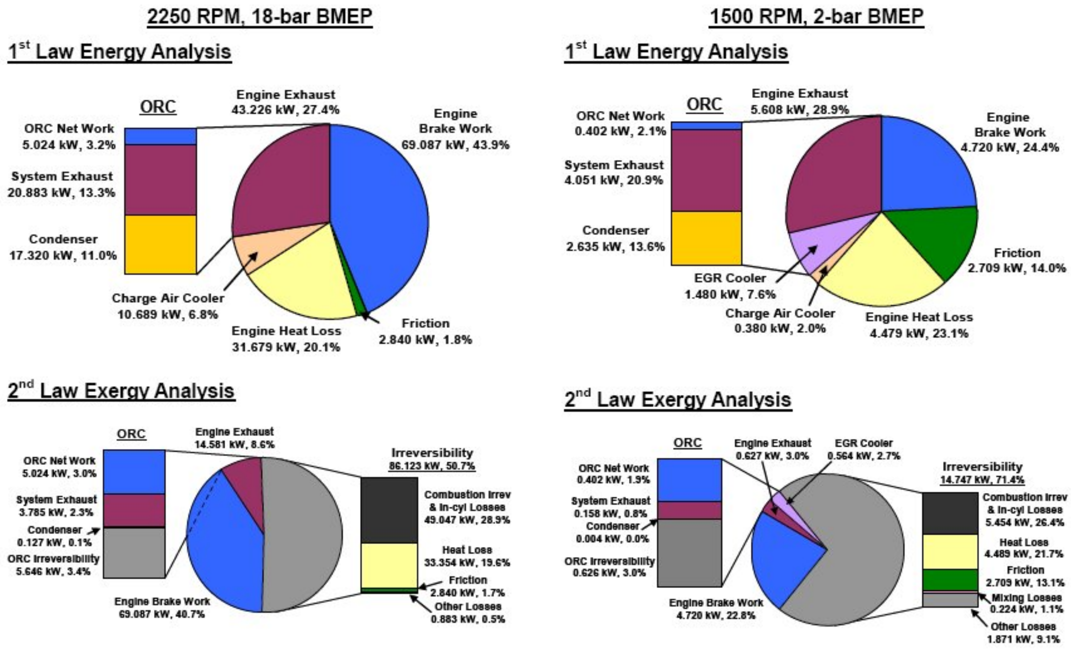

Figure 2.6: Energy and exergy analysis results in Edwards' paper (2)

In Bourhis paper (4), instead, exergy and energy analysis are more detailed and the repartition between different waste flows is more complex (more contribution are explicitly evaluated, such as intercooler, oil, cooling water etc.). Anyway, results do not substantially change: several irreversible processes lead to exergy destruction (energy degradation), and only a small part of waste heat can be recovered.

Ending the chapter, there are a phew points on which further studies are necessary:

- The recovery potential, as well as thermodynamic properties that characterize mass flows (temperature, enthalpy, entropy, etc.) strongly depends on engine load operating point. This shows the need to investigate more on how much this actually concerns the possibility of waste heat recovering and in how transient operations can influence the choice of the WHR system that should be adopted

- In Edwards' paper, only EGR and exhaust gas waste heat was recovered in system simulations. Other studies, instead, consider also the possibility to exploit energy content in cooling water or engine oil. This choice depends on several parameters, but one of major importance, that is often neglected, is the surface needed by heat exchangers. Further considerations have to be done in order to understand whether it is possible to conceive a recovery system on outlet flow that have a very low exchange coefficient, such as for exhaust gases, in a situation where space and weight are limited, such as on tractors.

- Exergy analysis is not enough to make a complete evaluation on the convenience of a WHR system. In fact, apart from exergy content of an outlet flow (that is possible to estimate without reference to the WHR system), exergetic efficiency of the exchange has to be considered because of its large influence on system performance. In this context, even low-exergy currents can be useful, for instance, in order to heat up the thermovector fluid and thus to enhance the exergy efficiency of the heat exchange placed on higher-exergetic flows

3

Testing equipment

TSAN (Technologies for safety and performance of agricultural machinery) unit in Cemagref research center of Antony disposes of two test benchs for tractors.

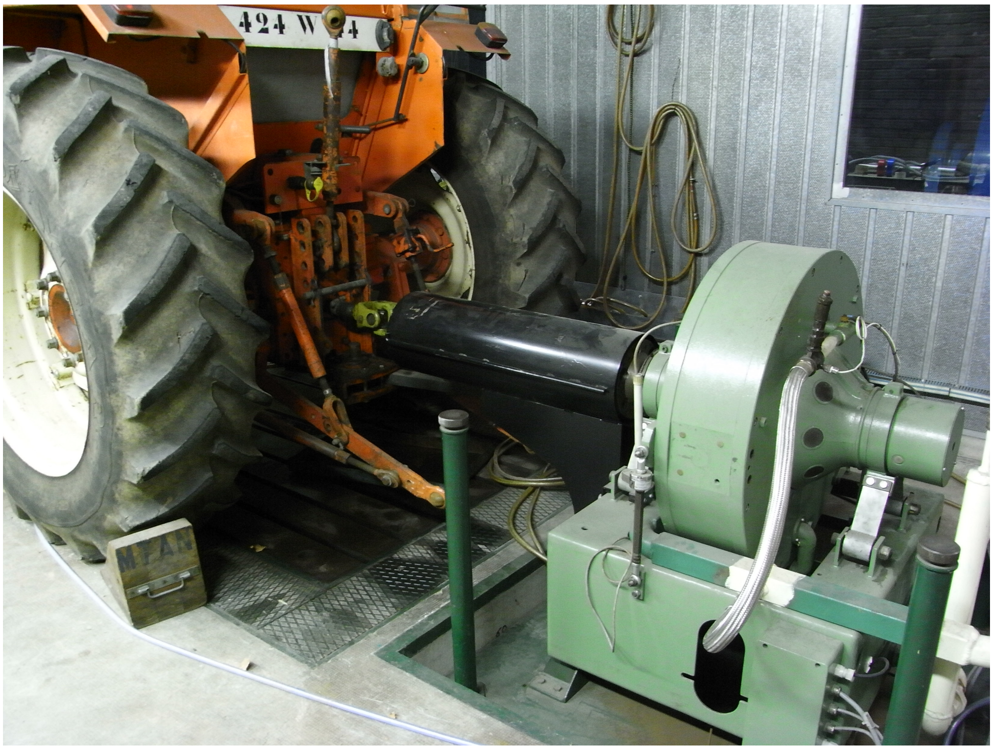

Figure 3.1: Tractor rear view. Focus on the PTO shaft connection and the electric brake

Selected test bench, normally used for standard performance tests (OCDE), is provided with a water cooled electric brake, based on Foucault currents, which is able to raise up to 400 kW of brake power. The connection between the electrical brake and the tractor's engine is granted by a direct connection to the power take-off (PTO). A focus on the PTO shaft connection with the electric brake is shown in figure 3.1. Cooling water for the brake is circulated by a pump which drives it to two external heat exchanger, meant to cool it down if its temperature rises over 30°C.

Other test bench main features are listed below

- Engine performance sensors for engine torque, speed and brake power
- Thermocouples for temperature measurements[1]
- A gas aspiration system (Figure 3.2) allowing to power up the engine inside the test bench building without any danger for operators
- Control strategy: both direct online control and input-file control are available
- Interface: results are shown directly on the screen and saved on output files

Test bench has been also recently provided with a gas-analyser, allowing an online measure of exhaust gas temperature, composition and pressure (Figure 3.3

Each sensor is connected to a main section (Figure 3.4) where all signals are channeled to the main processing unit. Each signal is treated in order to make it available to test controller, permitting a direct check on variables evolution with time.

Test bench is also equipped with an emergency stop system, connecting the tractor to test controller position. In case of any dangerous or unexpected system behavior, it is always possible to suddenly turn off the engine by simply pulling a metal wire.

Test bench software “duck” allows to command the tractor engine with a control over these parameters:

- Torque
- Rotating speed
- Throttle
- Power

[1] All thermocouples have been calibrated thanks to a calibrating system and a reference sensor. This allowed to define linear correction coefficients for the sensors that needed

Figure 3.2: Gas aspirations system

Figure 3.3: Gas analyzer

Figure 3.4: Sensor box

Instructions are given by controlling two of these parameters at once, that are chosen depending on test needs. Every instruction over the variation of a parameter must be associated to a transient-duration instruction, that means telling the software how much time the engine must take to pass from the operational condition A to B.

3.1 Tractor

The tractor is a Renault 851-4 R 7664, shown in figure 3.5. It is provided with a 4-cylinder engine, of 4156 cm^3 of volume (cylinder cross section: 105 cm^2, stroke: 120 cm). It is provided with a turbo-compound system exploiting exhaust gas energy and it is automatically lubricated. Measures taken at the test bench gave the following results:

Nominal power : 57.5 kW

Speed at nominal power : 2350 rpm

Torque at nominal power : 233.4

Maximum power : 57.5 kW

Maximum speed : 2480

Maximum torque : 277 Nm

Figure 3.5: Renault 851-4 R 7664

3.2 Sensors

A certain number of standard measures are taken for every kind of test, therefore being available for this study too. It si the case of:

Temperatures :

- Ambient air
- Air filter
- Fuel
- Engine oil
- Engine cooling liquid

Mass flows : fuel

Engine parameters :

- Power
- Engine torque
- Brake torque
- Engine speed
- Brake speed

3.3 Heat exchangers

3.3.1 Exhaust gas

The first heat exchanger is placed on the exhaust gas circuit in order to estimate its recovery potential. A schematic view of this system is shown in figure 3.6

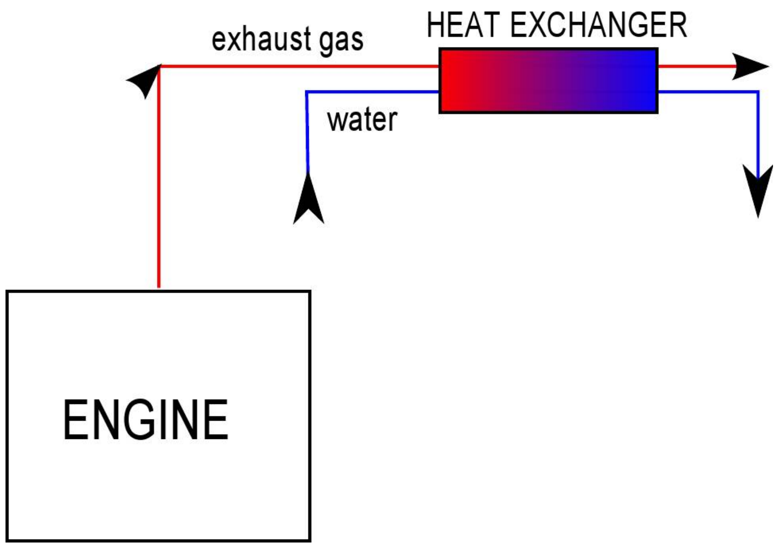

Figure 3.6: Schematic model of the heat exchanger for exhaust gas experiments

The heat exchanger is made of a copper pipe (diameter: 1 cm , length: 41 cm) concentric to the exhaust pipe, connected to a water circuit coming from the lab tap and leading to the waste drain. Inlet and outlet temperature for both the hot and the cold side are measured. Type K thermocouples (chromel - alumel) were used on the hot side, the temperature of the flow being able to reach more than 400°C, while normal type T (copper-constantan) thermocouples have been used for this circuit. Figure 3.7 shows a picture of the practical realization of this device

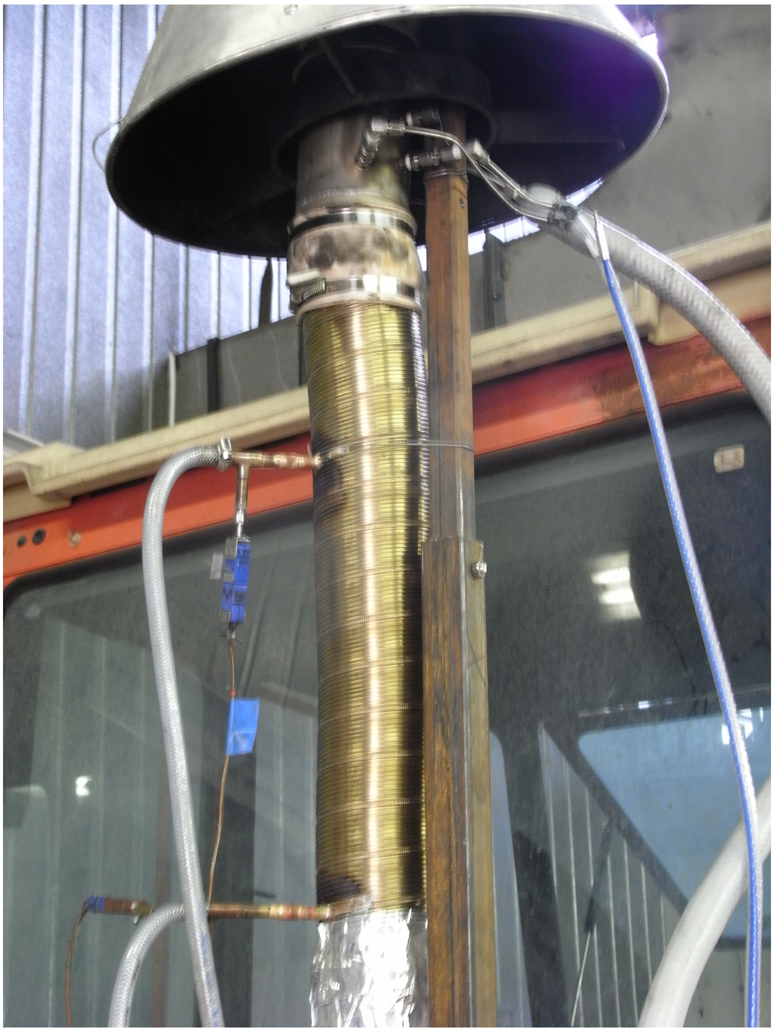

Figure 3.7: Heat exchanger and gas analyzer for exhaust gas experiments

For every measure other than temperature, such as pressure and mass flow, HORIBA OBS-2200 gas analyser was used, which allowed an online measurement and data uploading, strictly necessary feature with respect to this study. It allowed in particular to measure gas mass flow, pressure and composition.

The OBS-2000 series, apart from measuring exhaust gas temperature, also calculates mass emissions for CO, CO2 (NDIR analyzer without water extraction), THC (FID analyzer), NOx (CLD analyzer), and fuel consumption. OBS features include an exhaust flow meter, based on a Pitot tube, a GPS receiver for global positioning data and other sensors to monitor engine parameters and atmospheric conditions in real time. However, for this work, only temperature and mass flow measures were directly used, while data concerning exhaust gas composition were employed for a repeatability analysis.

For this exchanger, thermal insulation was very difficult. Exhaust gas temperature is too high to allow the use of a thermal coating, thus making it impossible to reduce the strong influence of heat exchange with the environment.

3.3.2 Engine oil

The second heat exchanger is placed on the engine oil circuit in order to estimate its recovery potential. A schematic view of this system is shown in figure 3.8

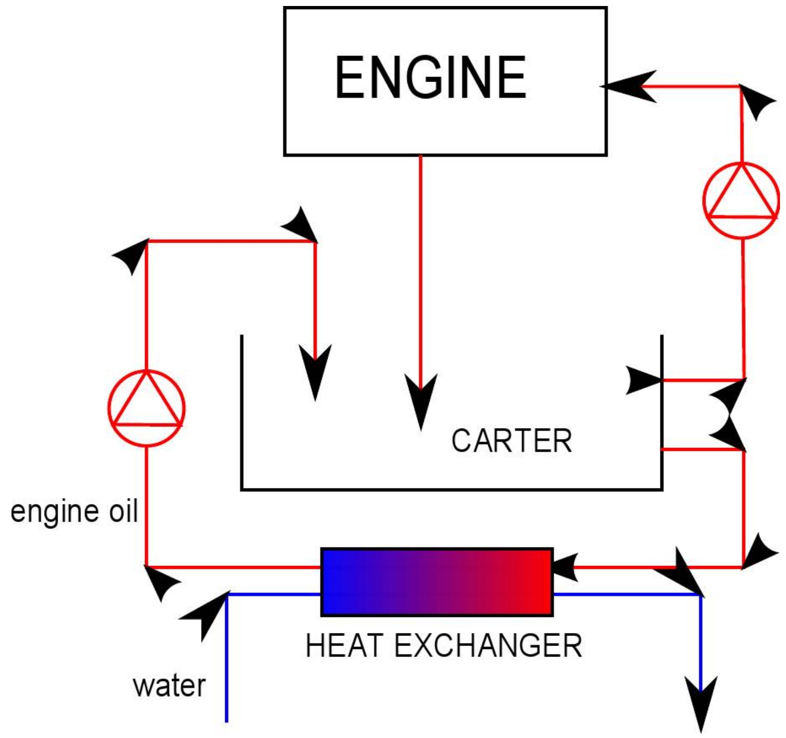

Figure 3.8: Schematic model of the heat exchanger for exhaust gas experiments

Oil is driven out of the carter toward the heat exchanger first through a metal pipe, then through a rubber one. After passing through the heat exchanger, oil is then processed by a pump allowing him to go up to the top of the carter, where the fluid is put back into the storage, closing the circuit. The pump is powered by a rectifier at 20V DC rather than at its nominal voltage (24 V, DC) in order to reduce oil mass flow that would risk, otherwise, to empty the heat exchanger. Furthermore, if oil mass flow became too high, water flow would not be able to cool it, leading to no difference between oil inlet and outlet temperatures. Water coming from the lab tab is driven through a rubber pipe to the heat exchanger, after which another rubber pipe drives it to the waste draining. For technical reasons the heat exchanger is set with a parallel-flow arrangement, despite its lower exchange efficiency. Anyway the exchange surface and, thus, the heat exchanged is too low for this to have a real negative influence on the matter.

Figure 3.9 shows the practical solution adopted for the engine oil heat exchanger Five thermocouples are placed on engine oil circuit:

- Oil, temperature in the carter
- Oil, heat exchanger inlet temperature, placed on the rubber pipe between carter exit and heat exchanger entrance [1]
- Oil, heat exchanger outlet temperature, placed on the rubber pipe between heat exchanger exit and pump
- Water, heat exchanger inlet temperature, placed on the rubber pipe before the heat exchanger
- Water, heat exchanger outlet temperature, placed on the rubber pipe after the heat exchanger

All thermocouples were calibrated with reference to a standard one, using a specific device, in order to have the maximum available instrumental precision. Thermocouples of type T (copper-constantan) have been used for this circuit.

A value for engine oil mass flow is needed in order to correctly evaluate energy and exergy quantities. A mass flow measurement system is therefore necessary. For the engine oil circuit the sensor is placed just after the pump, and the result is directly

[1] Originally, the thermocouple was directly placed on the metal pipe, without any use of rubber pipe in this part. The changement in system configuration was caused by the high thermal capacity of the metal pipe, that was remarked to influence oil temperature measurement

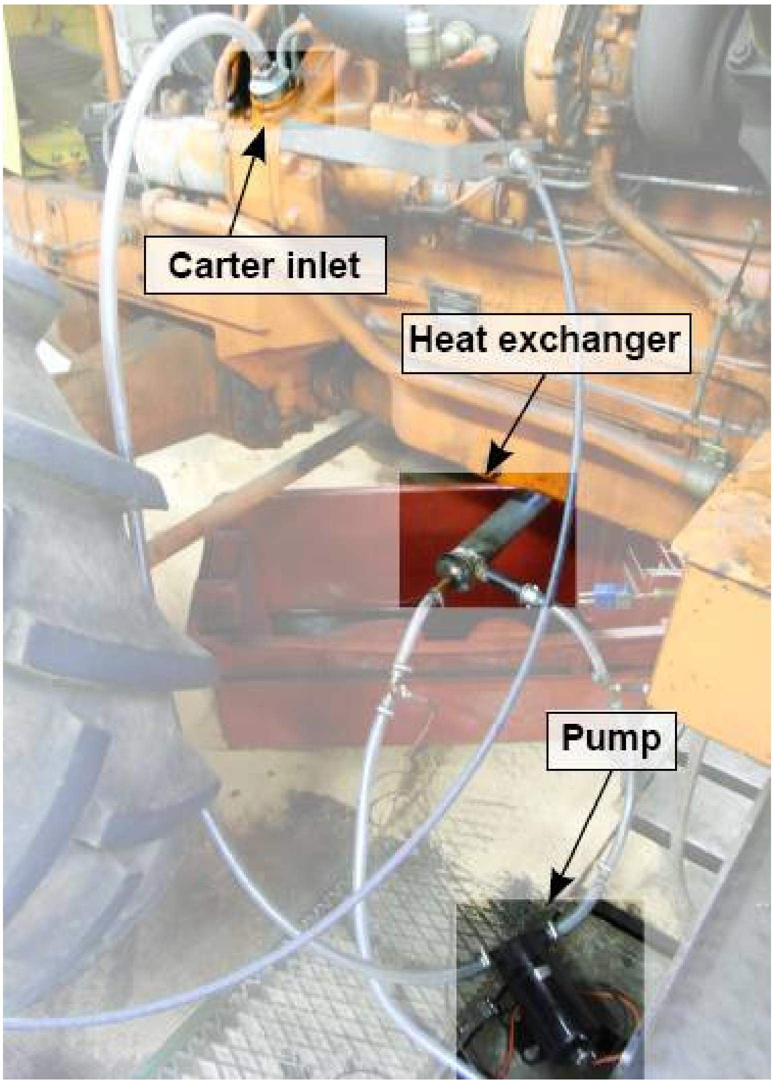

Figure 3.9: Heat exchanger and oil pump

shown on controller interface. Concerning the water circuit, the measure is taken at the end of the circuit, just before the waste draining[1].

The variations measured in water mass flow led to adapting a different solution. Water circuit was equipped with an intermediary storing system, meant to increase system inertia and, thus, to reduce variations in water mass flow. The height at which the storing system was placed was of primary importance for defining water flow. This parameter was set to a value allowing to have a mass flow low enough to see variations in temperature, but high enough to fill the pipe, thus making the exchange easier to be described analytically.

Water and engine oil mass flow setting was a quite important subject discussed at the beginning of the test series. This quantity should in fact be set to a value allowing to measure a temperature difference between inlet and outlet flow high enough to get rid of measure incertitude. Thermocouples available at Cemagref test bench have, in fact, a measurement incertitude of about ± 1°C, which means that a difference in temperature between inlet and outlet flow of less than approximatively 5 °C would be strongly affected by measurement incertitude. At the same time, mass flow should be high enough to allow the fluid to complete fill-up the pipe, since it would be otherwise impossible to give a correct estimation of the exchange surface. Moreover, decreasing the mass flow over a certain limit would cause troubles in its measurement.

First measurements led to a dramatically strong impact of the heat exchanger non-adiabatic behavior, making it very difficult to separate the two components. For this reason the heat exchanger, initially set with the configuration shown in figure 3.9, has been thermally insulated. Two different levels of thermal insulation were inserted:

- A coat of a low-conductive material (normally used for heating pipes insulation), for reducing undesired heat exchanges due to conduction. The same material have been used for pipes insulation, in order to reduce as much as possible the temperature drop between the engine oil inside the carter and at the inlet of the exchanger

[1] This makes mass flow measures response quite slow. However, in normal operations no variation in water mass flow should happen, which means that, theoretically, water mass flow measurement is inserted only for being able to initially set the flow, whose regulation is granted by manually opening a vane. However in real operations, variations in water mass flow have been measured during the tests. This can be caused by several phenomenons, first of all the stochastic variations caused by non-constant value of aqueduct pressure

- An external annular protection for reducing undesired convection. One of the most unexpected and undesired problems was, in fact, the strong forced-convective heat exchange cause by tractor fan. Air flow passing through the fan is, in fact, driven directly above the carter, where the heat exchanger is placed. This heat exchange has been reduced by a merely physical protection surrounding the exchanger and, thus, preventing the air flow coming from the fan from directly impacting the exchanger

Figure 3.10 shows the heat exchanger after the insulation process.

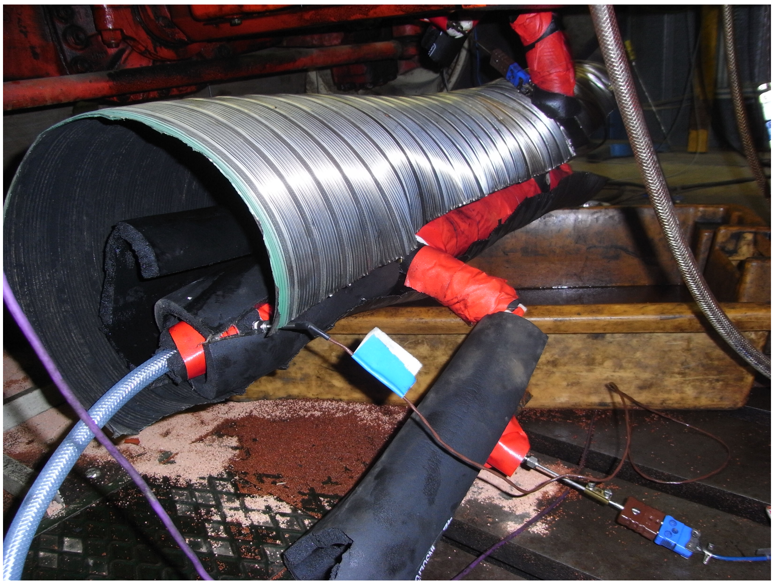

Figure 3.10: Heat exchanger and oil pump

These arrangements, together with the efforts to approach the thermocouples as much as possible to the inlet and outlet sides of the exchanger (thus reducing the influence of the heat exchange with air) made it possible to isolate the interesting phenomenon, that is the heat exchange between engine oil and water and the exchanger thermal inertia.

The other, strong issue related to the heat exchanger placed on the engine oil circuit is its thermal inertia. Firstly, in fact, engine oil itself tends to slowly react to variations in heat transfer. Furthermore tests results showed that the exchanger, whose external wall is made of a thick layer of cast iron, has a different inertia, making the modeling of the totality of the exchange even harder.

4

Discussion

Waste heat recovery potential of the engine of a tractor was the subject of this work. It will be structured in three main sections:

Steady state potential analysis : In the first part, the steady state exergy potential of both engine oil and exhaust gas will be evaluated, in order to write a chart of engine speed and power dependence of this potential.

Transient potential analysis : In the second part, the transient behavior of the exergy potential will be evaluated. The comparaison between a simple steady-state analysis and a more exhaustive transient one will be performed, in order to have a better understanding of the unsteady phenomena influence on heat recovery potential.

Heat exchange features : In the third and latter part of this project, heat exchange issues will be studied, and the influence of the different heat exchange properties of the two flows will be evaluated

4.1 Steady state tests

The first part of test series was meant to understand engine's behavior for certain specific operational points in steady state conditions. The analysis of these tests results allows to have a general knowledge of engine parameters dependence of its waste heat recovery potential.

The operational points chosen for the tests are representative of all of the possible working points of the engine, starting from idle conditions to its maximum power. However, operational points belonging to two already existent procedures were tested: the OCDE and the ISO 8178 protocol.

4.1.1 OCDE protocol

The OCDE protocol is a standard series of 6 test points originally developed in order to give a substantial estimation of engine fuel consumption over its life operations, allowing comparisons among different tractors. Even if some studies, such as the Cemagref report "Energy consumption of tractors: Remarks over OCDE and ISO 8178 protocols"(24), proved the incomplete effectiveness of this protocol, the operational conditions it takes into account are still an interesting starting point for defining engine performances. These points are defined as following:

1. 100% P_{nom}
2. 80% P_{nom}
3. 80% P_{nom} et 90% ω_{nom}
4. 40% P_{nom} et 90% ω_{nom}
5. 60% P_{nom} et 60% ω_{nom}
6. 40% P_{nom} et 60% ω_{nom}

4.1.2 ISO 8178 protocol

The ISO 8178 protocol is a second standard series of 8 points developed instead for pollutant emissions evaluation. It includes:

1. 100% C_{nom} et 100% ω_{nom}
2. 75% C_{nom} et 100% ω_{nom}
3. 50% C_{nom} et 100% ω_{nom}
4. 10% C_{nom} et 100% ω_{nom}

5. 100% C_{max} et 100% $\omega_{C_{max}}$

6. 75% C_{max} et 100% $\omega_{C_{max}}$

7. 50% C_{max} et 100% $\omega_{C_{max}}$

8. idling

Other points were inserted in order to get a chart wide enough to be representative of the entire operational range. All points taken into account for the following analysis are listed in table 4.1 and shown in figure 4.1

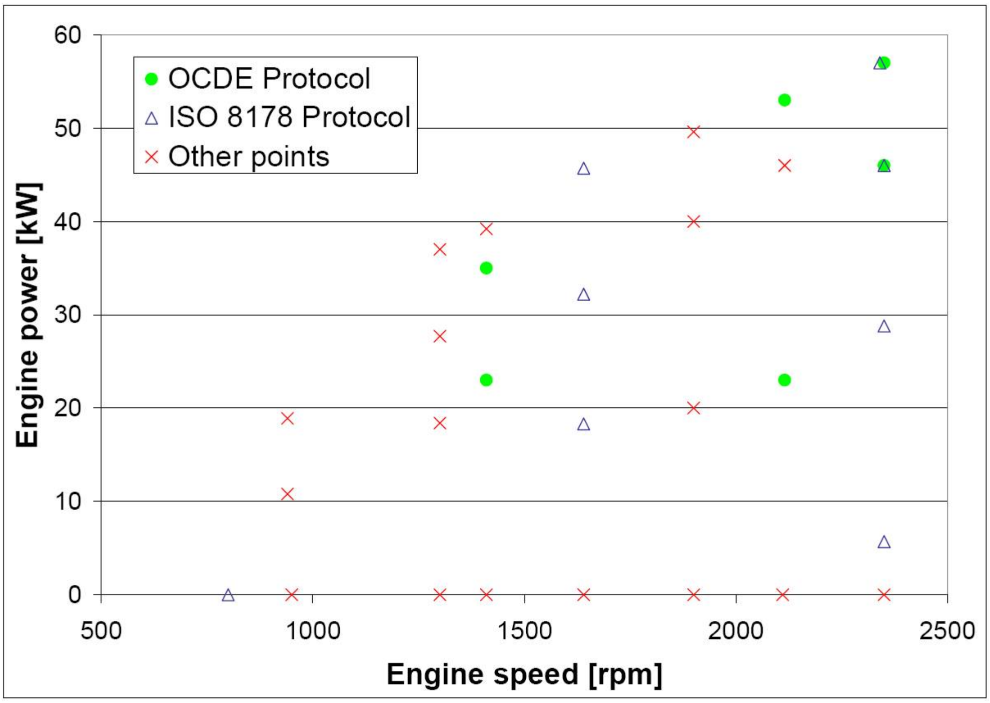

Figure 4.1: Engine power versus speed chart for steady state test points

As seen in the first chapter, exergy is the tool allowing to estimate the maximum work potential. However, the word "maximum" is not casual: in a real recovery system, only a part of this maximum potential can effectively be harvested by the system. Anyway, the objective of this work is to evaluate recovery potential, without interest to what kind of system is used for this purpose.

Protocol	Power	Speed
	rpm	KW
	0.0	2350
ISO 8178	5.7	2350
ISO 8178	28.8	2350
OCDE	46.0	2350
OCDE, ISO 8178	57.0	2350
	0.0	2115
OCDE	23.0	2115
OCDE	46.0	2115
	54.4	2115
	0.0	1900
	20.0	1900
	40.0	1900
	49.6	1900
	0.0	1640
	16.0	1640
ISO 8178	18.3	1640
ISO 8178	32.2	1640
ISO 8178	45.7	1640
	0.0	1410
OCDE	23.0	1410
OCDE	35.0	1410
	39,2	1410
	18,4	1300
	27.7	1300
	37.0	1300
	0.0	1300
	0.0	950
	10.8	950
	18.9	950
ISO 8178	0.0	780

Table 4.1: Steady-state measure points

As said in chapter 2, he reference equation for this kind of calculation is the following:

$$\dot{ex} = \dot{m}(h - h_0) - T_0(s - s_0) \tag{4.1}$$

where 0 label corresponds to equilibrium state with reference to environment. For all steady-state and transient tests, a temperature of 298.15 K and a pressure of 101325 Pa is chosen for reference conditions. This choice is driven firstly because room temperature at the test bench showed to vary in a range close to 298.15 K, and secondly because this is a standard value[1] in thermodynamics.

Equation 4.1 allows to evaluate the exergy potential of a current once its enthalpy and entropy value are known. It is however important to keep in mind that even exergy is a measure of the real amount of work it is possible to extract, but only an ideal one achievable only with reversible processes and no losses.

4.2 Experimental measurements

In order to have real data on which basing every following calculation and reflexion, a number of test series was performed. The test bench, the tractor and all the experimental devices used for this purpose are described in chapter 3. The choice of the properties and variables which were measured during these tests was based on the need for recovery potential evaluation, in terms of the exergy of each studied flow, the reference equation being the 4.1.

Two different sources of waste heat were taken into account during these tests: exhaust gas and engine oil

4.2.1 Exhaust gas experiments

4.2.1.1 Temperature

Because of its low thermal inertia, the steady state value of exhaust gas temperature could be directly measured during the experimental phase of this study. It was therefore possible to have a good approximation of the steady-state value without any need of mathematical intermediation.

[1] 298.15 K at 101325 Pa is the definition of Normal Conditions

4.2.1.2 Mass flow

With respect to Diesel engines functioning, exhaust gas mass flow is a function of:

- Engine speed, which determines air mass flow (the hypothesis implicitly made here is that the quantity of inlet air for each intake stroke is constant)
- Throttle, which determines fuel mass flow into the cylinder
- Turbo compound operations, whose activity increase inlet air pressure, thus increasing inlet mass flow

$$\Delta E_{eg} = \dot{m}_{eg} \int_{T_0}^{T_{eg}} c_{p,eg}(T) dT \tag{4.2}$$

However, we failed to establish an analytical expression for exhaust gas flow. For this reason, measured values where used for exergy evaluation.

Therefore the use of a gas analyzer was necessary in order to measure exhaust gas mass flow. This device was used only during exhaust gas experiments, while it was not in service during oil experiments.

The gas analyzer is equipped with a Pitot tube allowing the indirect measure of gas velocity and, thus, of its volumetric flow. The software installed in the control system, however, elaborates data coming from measure devices. In this sense, the data file produced by the software shows a gas flow measured in standard cubic meters per minute, where standard conditions are fixed at 293.15 K and 101325 Pa. This means that the correction coefficient needed to convert the volumetric flow into a mass flow do not depend on temperature.

4.2.2 Engine oil experiments

4.2.2.1 Temperature

A temperature value is necessary in order to evaluate engine oil exergy content. For this reason, with reference to the setup described in section 3.3, oil temperature at the inlet section of the heat exchanger has been measured.

Engine oil temperature evolution is influenced by the effects of its high thermal inertia. The time needed to get to the steady-state value is, thus, very long, as shown in figure 4.2. That is why, in order to evaluate steady-state temperatures, a regression courbe had to be calculated for each point. For each change in imposed parameters, such as power and torque, engine oil temperature evolution is supposed to follow the equation shown in equation 4.3

$$T_{eo} = T_{eo,0} + (T_{heo,\infty} - T_{eo,0}) * (1 - e^{-\frac{t}{\tau}}) \tag{4.3}$$

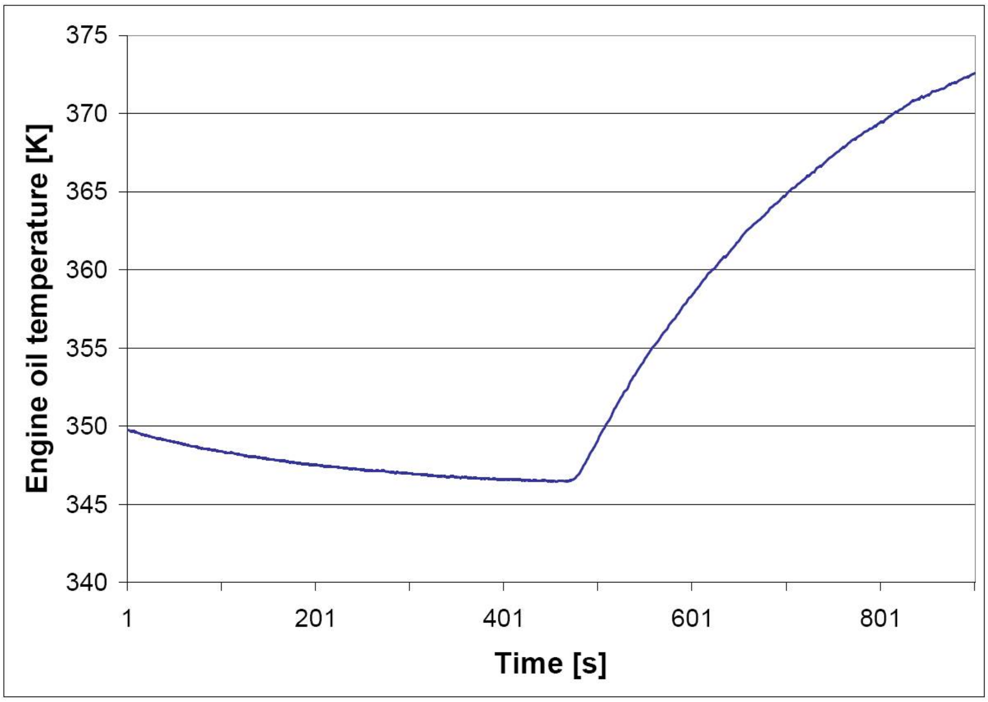

Figure 4.2: Example of engine oil temperature evolution after a variation in engine power and speed instruction is experienced

Experimental data showed that engine oil temperature closely follow this kind of evolution, especially after the first part of the transient phase. With reference to equation 4.3 for the time dependence of engine oil temperature, a time constant of the "circuit" can be calculated, defined as the time it takes for the difference between the instant and the steady-state temperature to decrease to the e^{-1} of the starting value. The value of this time constant (τ) is included in the 800-1200 seconds range for engine oil, while for exhaust gas it stands between 10 and 30 seconds.

Engine oil temperature increases with power. In fact, higher power demands directly imply higher quantities of waste heat. This phenomenon has clearly an impact on engine oil too, thus increasing its temperature.

Engine speed has a lesser influence on oil temperature than power one. However, a slight increase in engine oil temperature with higher speeds can be observed. The phenomenon can be explained by the fact that a higher engine speed is generally associated with higher friction heat losses, eventually increasing engine oil temperature

4.2.2.2 Volumetric flow

The volumetric flow induced by the pump, operating at constant voltage and, thus, at constant Δp, was considered. This did not mean, however, a constant engine mass flow. Two oil properties concerning volumetric flow, in fact, strongly depend on oil temperature:

- Viscosity
- Density

For this reason, the same problem encountered in engine oil temperature analysis arises in its volumetric flow case. One solution for this problem was supposed to evaluate the steady-state mass flow for engine oil in the circuit in the same manner as done for temperature in order to obtain the equilibrium value. Unfortunately, oil mass flow has a much less predictable behavior, and the regression performed on volumetric flow data did not lead to good results. However, as shown in figure 4.3, temperature dependence of engine oil volumetric flow seems to be approximatively linear. This lead to the conclusion that the best approximation that could be used in this case was to apply the linear regression coefficients to the equilibrium temperature, in order to get a value not too far from the real one.

4.3 Steady state exergy potential evaluation

4.3.1 Exhaust gas

Exhaust gas flowing in an open circuit, that mechanical equilibrium too should be considered for enthalpy and entropy balance.

Theoretically, a pressure difference between outlet flow and environment should be taken into account in order to evaluate enthalpy difference between the two states.

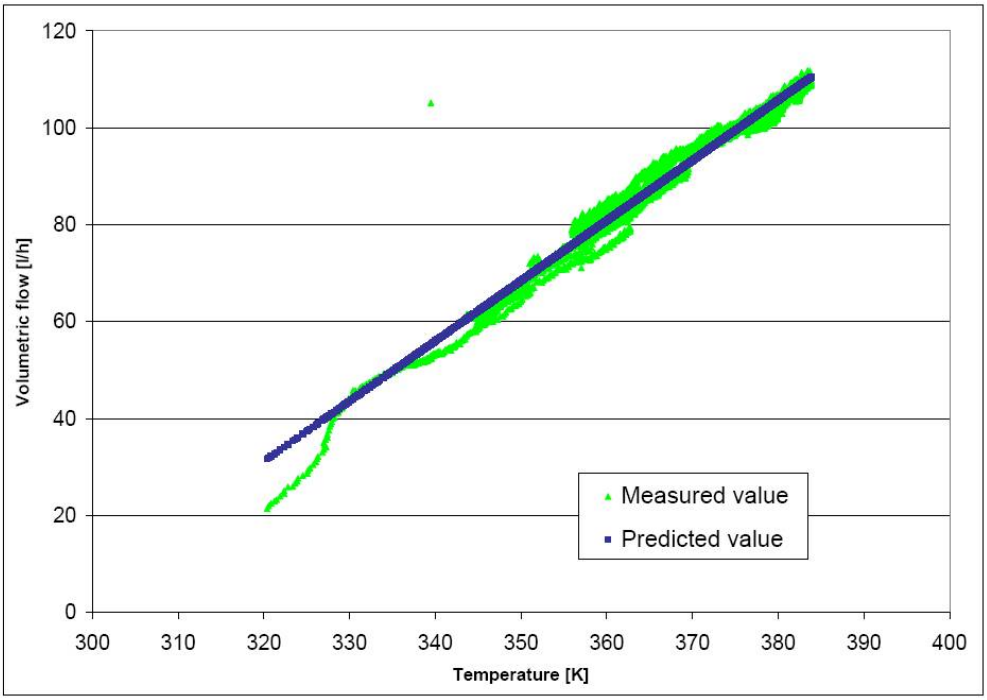

Figure 4.3: Temperature dependence of engine oil volumetric flow. Predicted versus measured value

However, according to the ideal gas approximation, enthalpy only depends on temperature, without any reference to pressure. This means that the enthalpy difference can be written as following:

$$\Delta h = \int c_p(T)dT \tag{4.4}$$

Pressure dependence of entropy, instead, should be evaluated; however, exhaust gas pressure being very close to environment pressure, the term related to the pressure difference versus the environment will be neglected. So, starting from the well known equation:

$$dh = Tds + vdp$$

Neglecting the term connected to the pressure difference and using the 4.4, we have that:

$$\Delta s = \int c_p(T)\frac{dT}{T} \tag{4.5}$$

Eventually, exhaust gas exergy content can be obtained thanks to the substitution of

4.4 and 4.5 in equation 4.1.

$$\dot{ex}_{eg} = \dot{m}_{eg} \int_{T_0}^{T_{eg}} c_{p,eg}(T)dT - T_0 \int_{T_0}^{T_{eg}} c_{p,eg}(T)\frac{dT}{T} \tag{4.6}$$

One issue in exhaust gas potential energy evaluation is related to its composition: in fact, exhaust gas is always formed by a mix of different species, and its composition changes depending on operating conditions. For a correct evaluation of exergy recovery potential, it is therefore necessary to know the exhaust gas composition, in order to find a global specific heat at constant pressure value, weighted on each component mass fraction.

Moreover, matter at gaseous form has its specific heat at constant pressure value changing with temperature. In this case, the following hypothesis have been considered:

- Specific heat at constant pressure value do not change with pressure. This hypothesis is, theoretically, quite strong. However, in this case, pressure is quite stable even in transient operation, and evidence can be given that specific heat at constant pressure variation with pressure is negligible

- Specific heat at constant pressure evolution with temperature is considered to follow polynomial function. Values for function's coefficients have been calculated through the least mean squares method, applied to a series of c_p values in the range of interesting temperatures

For a gas, thus,specific heat value is, as general statement, a function of its temperature and chemical composition. Exhaust gas are composed by a mix of several different components. Four among them have a remarkable concentration, and will be taken into account[1]:

- Carbon dioxyde (CO_2)

- Steam (H_2O)

- Oxygen (O_2)

- Nytrogen (N_2)

[1]Diesel engines always work in over-stechiometric conditions, thus explaining oxygen importance

Mass fraction value for each of the four component is calculated considering a complete combustion, starting from the measured value of exhaust gas and fuel mass flow. Therefore, we neglected soot, unburned and CO concentrations in exhaust gas composition. This approximation is, however, easy to accept since it does not affect in a remarkable way the final result. Further details of the analytical model used for these calculations are given in Appendix C.[1]

Each exhaust gas component's specific heat value depends on temperature. Tables are available in which specific heat value is given for different temperature values (25). Regressions of this data have been calculated in order to be able to evaluate specific heat value for a given temperature. Polynomial regression are often used for this purpose, and in this case a specific regression was performed, in the temperature range of interest, in order to have the best precision.

The following expression is therefore obtained:

$$c_{p,X} = a_{0,X} + a_{1,X}T_{eg} + a_{2,X}T_{eg}^2 + a_{3,X}T_{eg}^3$$

where X is a general chemical species.

In appendix A coefficients values for each chemical species are detailed.

Once the specific heat for each gas component is available, all these are used to obtain a mean value, weighted on mass fraction:

$$c_{p,eg} = \sum_i c_{p,i}(T)y_i \tag{4.7}$$

All necessary data for exergy calculation being available, equation 4.8 will be used:

$$\dot{ex}_{eg} = 1.2\dot{V}_{eg,st}\int_{T_0}^{T_{eg}} c_{p,eg}(T)(1-\frac{T_0}{T})dT \tag{4.8}$$

Figure 4.4 shows exhaust exergy flow values for each couple power/speed. Black dots show values coming from direct measurements, while the contour lines show the cubic interpolation performed by the software. A complete list of calculated exergy values for each test point is shown in table B.1 in the appendix B.

[1]It should be noticed that the gas analyzer is able to measure gas concentrations, such as those of CO_2 and H_2O. However, no measure of oxygen and nitrogen concentration is available, which means that even by using measured values a calculation should have been performed. Moreover, gas composition low influence on specific heat value and, consequently, on exergy potential, justifies this approximation

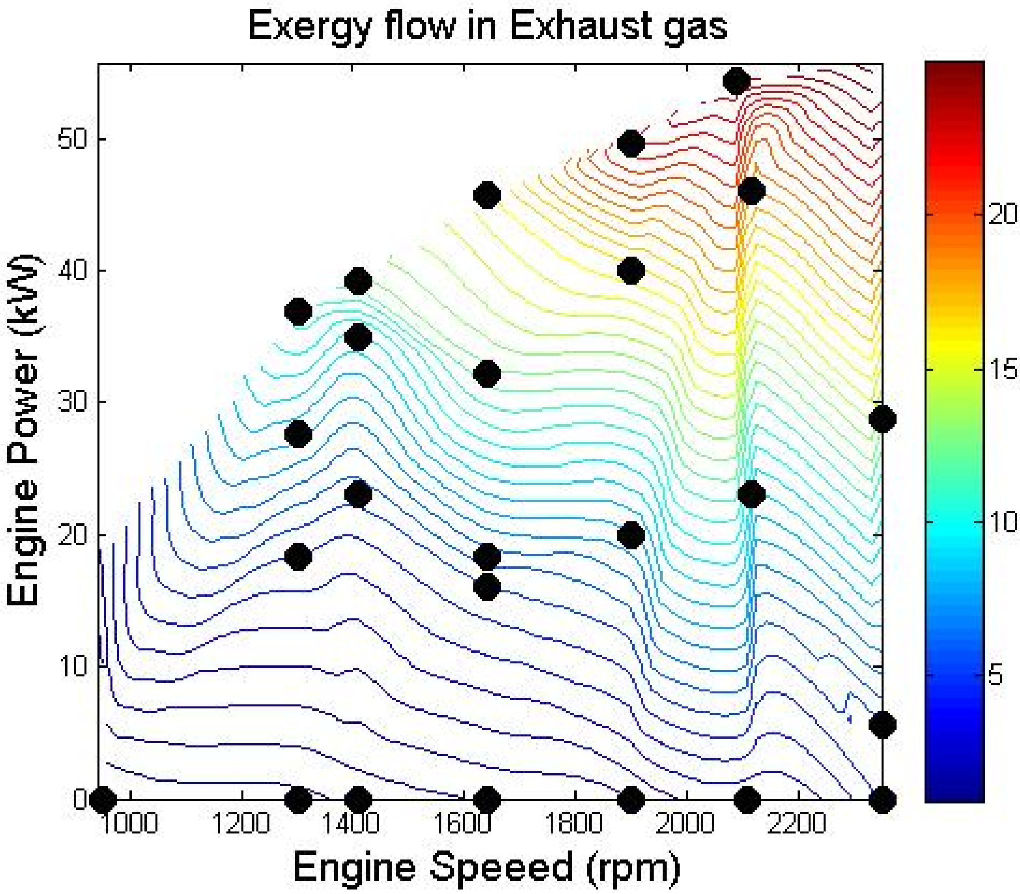

Figure 4.4: Exhaust gas steady state exergy chart

It is easy to observe that exhaust gas have a quite high exergy potential, since its maximum value exceed 25 kW (for an instant engine power of 57.6 kW). Exhaust gas exergy dependence on engine speed and power is very close to engine oil one, but is in general smoother. Recovery potential strongly grows with engine power, while engine speed's influence is much lower, even if still observable. For exhaust gas exergy the maximum measured value is quite difficult to identify, but it is approximatively located at the condition of nominal speed and power.

Figure 4.5 shows fuel exergy content dependence of the fraction of recoverable heat, in terms of exergy, for exhaust gas. It is easily observable, first of all, that in this case the pourcentage of inlet exergy that can be recovered through exhaust gas is quite high, this value almost reaching the 15%. Fuel exergy inlet dependence of the ratio remarkably increase with fuel exergy inlet, while the ratio slope $\frac{\text{Exhaust gas exergy}}{\text{Power}}$ is more pronounced.

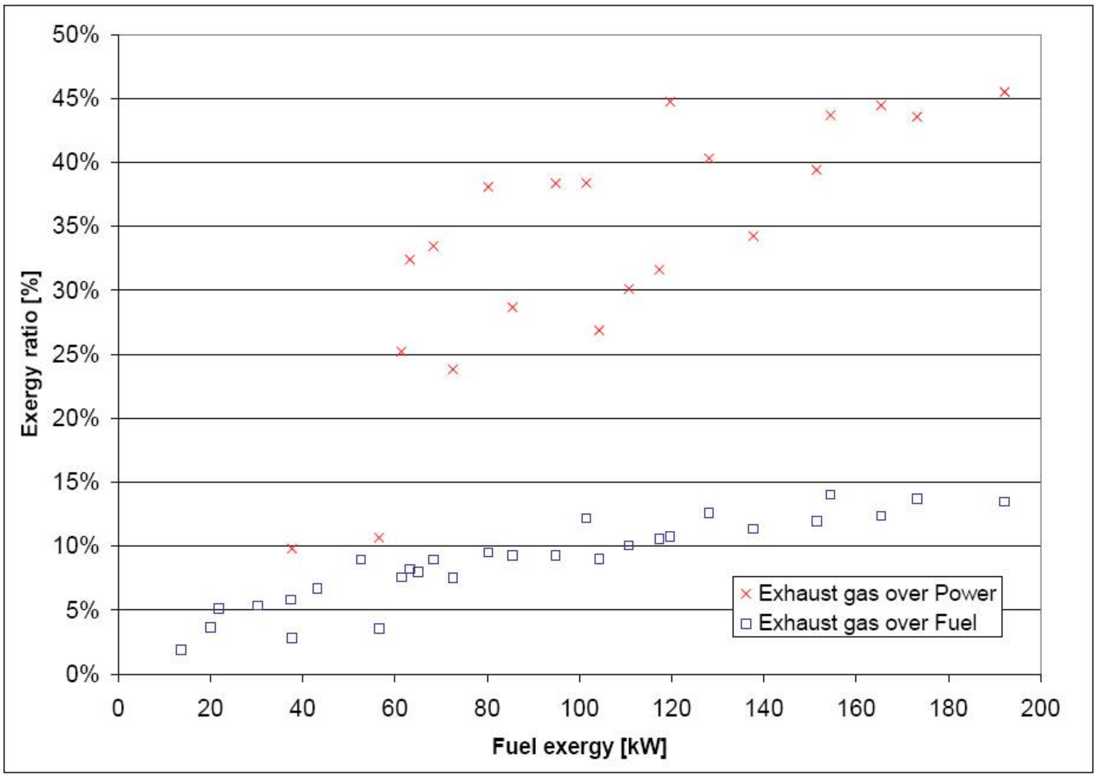

Figure 4.5: Exhaust gas over fuel exergy ratio and exhaust gas over power exergy ratio, plotted versus fuel exergy content

4.3.2 Engine oil

Being a closed circuit, there will not be any reference to mechanical equilibrium. Engine oil exergy content only depends on its temperature, and it is given by:

$$\dot{ex}_{eo} = \dot{m}_{eo} \int_{T_0}^{T_{eo}} c_{p,eo}(1 - \frac{t_0}{T})dT \tag{4.9}$$

Thus, the following properties are needed in order to estimate recovery potential:

- mass flow
- temperature
- specific heat capacity

Temperature, as shown in the previous section, has been directly measured during test series.

For what concerns engine oil mass flow, it is not directly available. As seen in section 4.2.2.2 measuring equipment made only available data for volumetric flow. A density correction need thus to be performed, as shown in equation 4.10

$$\dot{m}_{eo} = \rho(T_{eo})\dot{V}_{eo} \tag{4.10}$$

Density was available in literature ((26)) and a linear regression was calculated from available data, as shown in figure 4.6. The equation for the regression courbe is:

$$\rho_{eo} = aT + b \tag{4.11}$$

Just like density, specific heat value for engine oil is a function of its temperature. However, as shown in figure 4.7, the dependence is clearly linear in the temperature range concerning this study, and a regression could be obtained on data coming from literature (26). The equation for the regression courbe is:

$$c_{p,eo} = aT + b \tag{4.12}$$

where regression coefficients are detailed in appendix A.

The two coefficients for engine oil specific heat capacity are therefore easily available.

Figure 4.8 shows engine oil exergy flow values for each couple power/speed. Black dots show values coming from direct measurements, while the contour lines show the

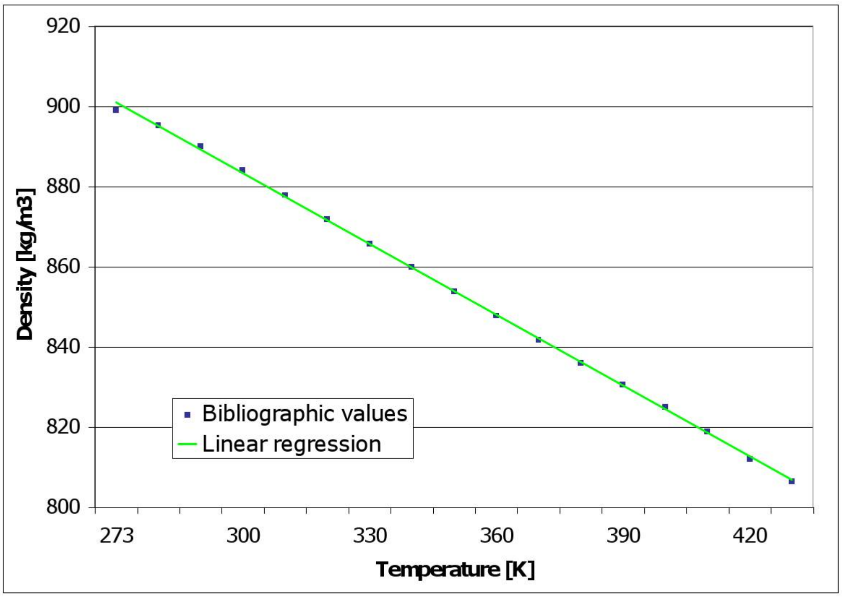

Figure 4.6: Temperature dependence of engine oil density. Predicted versus measured value

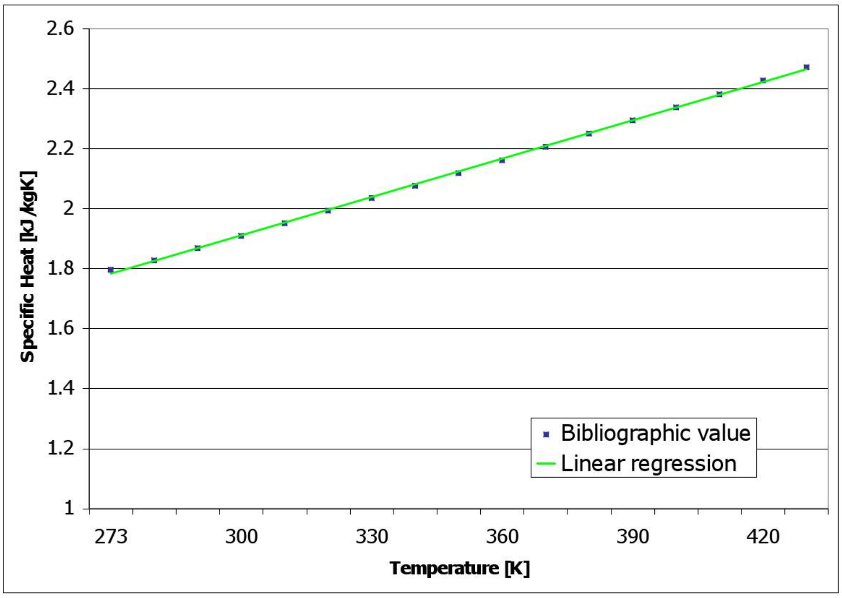

Figure 4.7: Temperature dependence of engine specific heat. Regression courbe versus bibliographic data

cubic interpolation performed by the software. A complete list of calculated exergy values for each test point is shown in table B.2 in the appendix B.

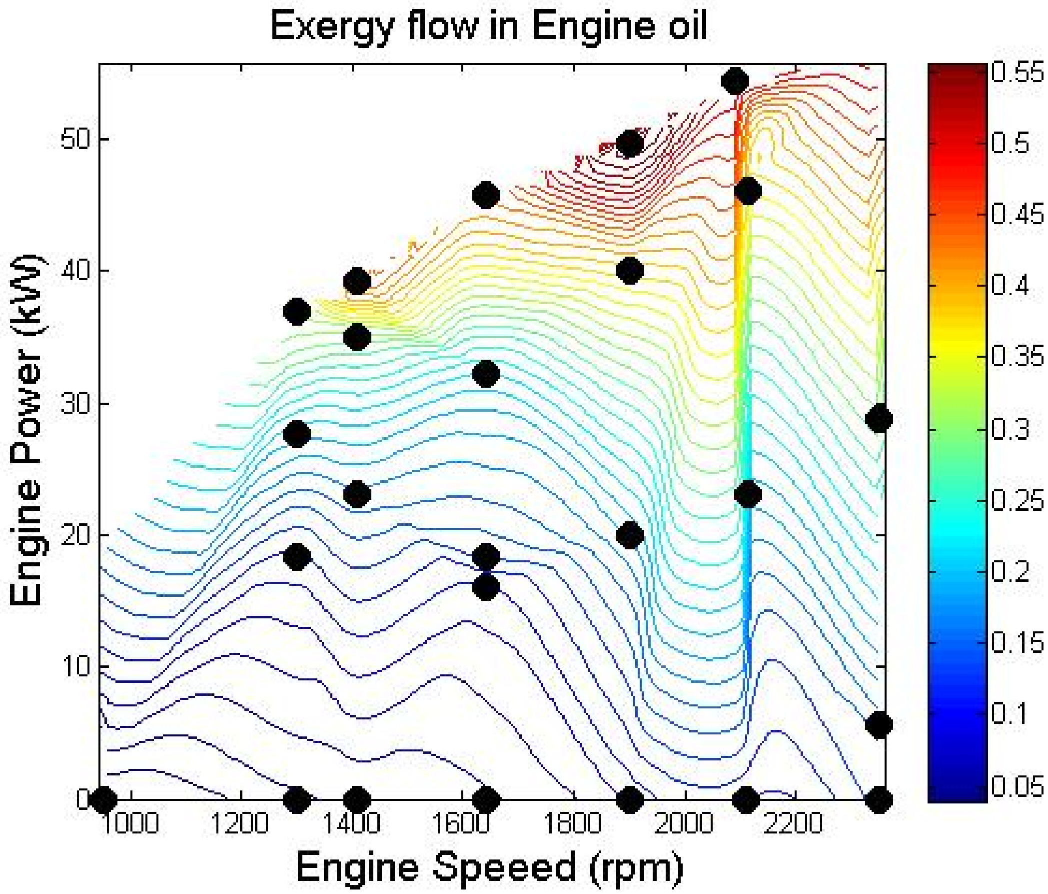

Figure 4.8: Engine oil steady state exergy chart

As shown, engine oil recovery potential in terms of exergy content is very low, never exceeding 600 W at a power demand that can get near to 60 kW. Concerning its dependence on power and speed, a general positive slope for both variables is observable. Calculated maximum value for exergy potential is located at about 1900 rpm speed and 50 kW power.

These observations do not make of engine oil and ideal candidate for waste heat recovery. Its low temperature drives to low recovery efficiencies, making it difficult to exploit it as a heat source for useful work production.

Figure 4.9 shows fuel exergy content dependence of the fraction of recoverable heat, in terms of exergy, for engine oil. It is easily observable, first of all, that the pourcentage of inlet exergy that can be recovered through engine oil is very low, this value never exceeding the 0.4%. This value only slightly increase with fuel exergy inlet, while the ratio slope $\frac{\text{Exhaust gas exergy}}{\text{Power}}$ is more pronounced.

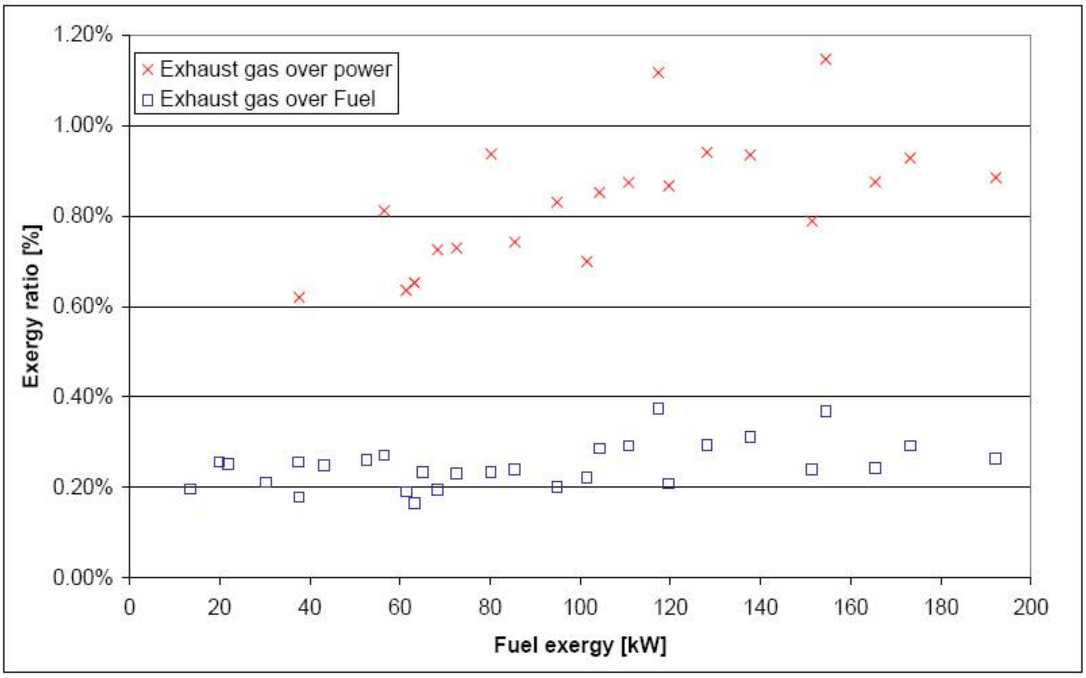

Figure 4.9: Engine oil over fuel exergy ratio and engine oil over power exergy ratio, plotted versus fuel exergy content

4.4 Transient tests

Getting to an estimation of the exergy potential of a tractor for different operational conditions is certainly an interesting result. However, the real novelty of this work is supposed to be the non steady-state analysis. As said in the introductive chapter, tractor use is somehow halfway between a steady-state working condition, (typical of small Diesel-engine driven power plants or ships) and a strongly non steady-state working condition, typical of cars. On field, tractor mainly works at a given steady state, whereas engine operations are subjected to strong fluctuations during transport phases. This reasoning led to the need of a specific study on recovery potential evolution over transient operations.

Figure 1.2 in the introduction chapter shows typical on-field operations. A first possible approach to the problem could be the choice of the periodic instructions pattern represented in figure 1.2. However, in order to have a better understanding of transient phenomena, a sinusoidal behavior was chosen. In this case in fact the results mathematical form is much easier to analyze, since the difference between the input and the output is often limited to a reduction in amplitude and the arising of a phase angle, representing the delay in measured response.

In order to understand transient phenomena, six different tests have been performed, whose instructions are listed in table 4.2.

In the first one (T1), power fluctuation amplitude was fixed between 0 and 100% of nominal power (57.6 kW), representing respectively maneuvering operations and labour conditions. Engine speed, instead, was kept at a constant value (2350 rpm) in order to avoid the mixing of too many different effects in the same experience. However, even if this engine speed is perfectly representative of labour conditions, the same cannot be said for maneuvering, where idle conditions are normally measured and, thus, an engine speed of about 800 rpm should be more consistent with real conditions.

Measures on the field show that the time intervals typical of agricultural operations strongly vary depending on different parameters, such crop kind and field size. However, a periodical behavior of 5 minutes, of which 4 of laboring and 1 of maneuvering, as been measured in on-field operations. Thus, the period of the sinus form for power instructions was fixed to 300 seconds.

n	Ave power	Min power	Max power	Min speed	Max speed	Period
	kW	kW	kW	rpm	rpm	s
1	28.8	0.00	57.6	2350		300
2	28.8	5.76	51.8	2100		300
3	28.8	14.40	43.2	2100		300
4	28.8			2000	2200	300
5	25.9	0.00	51.8	800	2100	300
6	28.8	0.00	57.6	2350		60

Table 4.2: Engine power and speed instruction for transient tests

In order to have a better understanding of the connections between engine speed and power influence on exergy and their different transient behavior, other tests were performed around a mean position, whose features are listed in table 4.2 The T2 and T3 tests, together with the T1, were meant to understand power influence, which demonstrated to be the stronger one. They all share the same mean power, while the constant value adopted for engine speed was changed from 2350 to 2100 rpm. This choice allows to keep the same mean position in the speed-varying test (T4)

A fifth test was performed with both engine speed and power varying over time with a sinusoidal behavior. The two signals were not "in phase", in order to give to the power-speed courbe a form as showed in figure 4.10. In this case the power fluctuation was fixed between 0 and 90% of nominal power, while engine speed was restricted to the interval 800 - 2100 rpm.

A last test (T6) was performed in order to study waste heat recovery evolution over faster fluctuations. In this case power periodical variations between 0 and 57.6 kW was achieved in 60 seconds only.

The total duration of each test was set to 10 complete periods for the first 5 tests and to 20 complete periods for the sixth one. This value was set in order to allow engine oil to get near to a steady-state behavior.

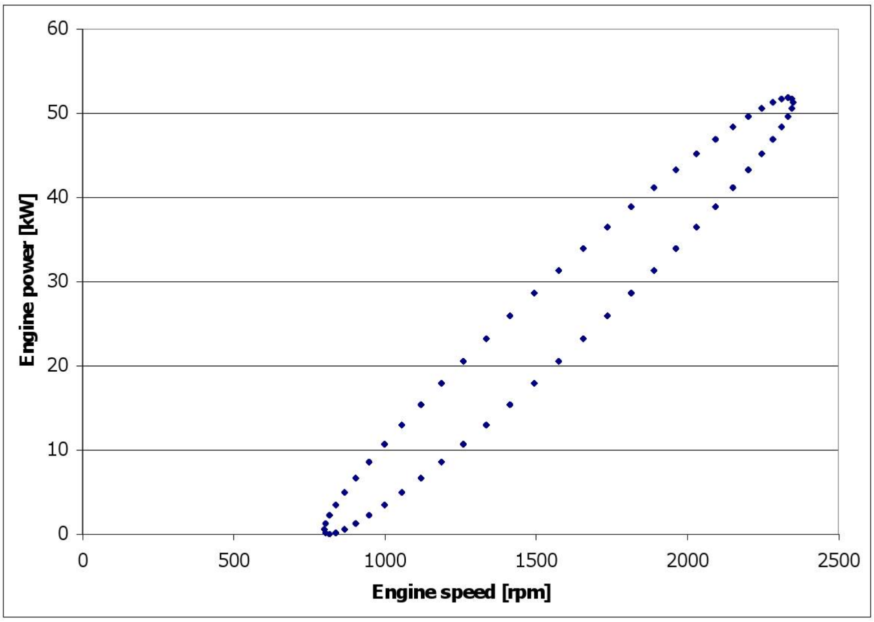

Figure 4.10: Power instructions for T5 test

4.4.1 Exhaust gas

Figure 4.11 shows the evolution over time of exhaust gas exergy potential for T1, T2, and T3 tests, where only power instructions are subject to variations. Exergy response follows a sinusoidal shape that is shifted and smoothed compared to the power signal. This shift and smoothing are however low and, as general statement, exhaust gas exergy closely follows power instructions.

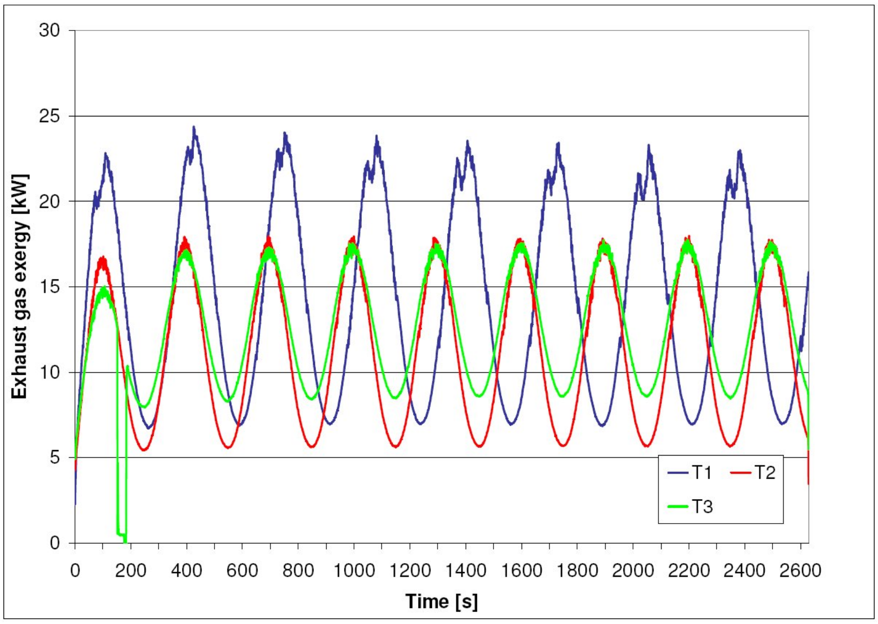

Figure 4.11: Exhaust gas exergy potential evolution, T1 to T3 tests

T1 exergy has a different period compared to T2 and T3: this is due to setting problems related to the unsteady use of a static bench, and it has no physical meaning.

The observation of the graphics lead to the conclusion that, in general, exhaust gas exergy sinus features, such as mean value and amplitude, are directly connected to those of the power instructions signal. However, comparing T2 and T2 tests, we can see that even if they share the same mean engine power and speed, they do not have the same mean value. This is clearly linked to the difference in room temperature between the two tests, shown in table 4.3

Exhaust gas exergy was also compared to the exergy input in fuel and the power output[4]. The comparison is performed on instant values for the T2 test (figure 4.12),

[4]Theoretically, inlet air exergy content should be taken into account too. However, since the

		T1	T2	T3
Power[1]	kW	28.8	28.8	28.8
Speed	rpm	2350	2100	2100
Room Temperature[2]	°C	24.1	25.6	27.9
Steady-state exergy[3]	kW	14.7	11.3	12.9

Table 4.3: Average values of engine speed and power, room temperature and steady state value of exhaust gas exergy for T1, T2, and T3 tests

where inlet flow exergy content and power output are plotted as a function of exhaust gas exergy content.

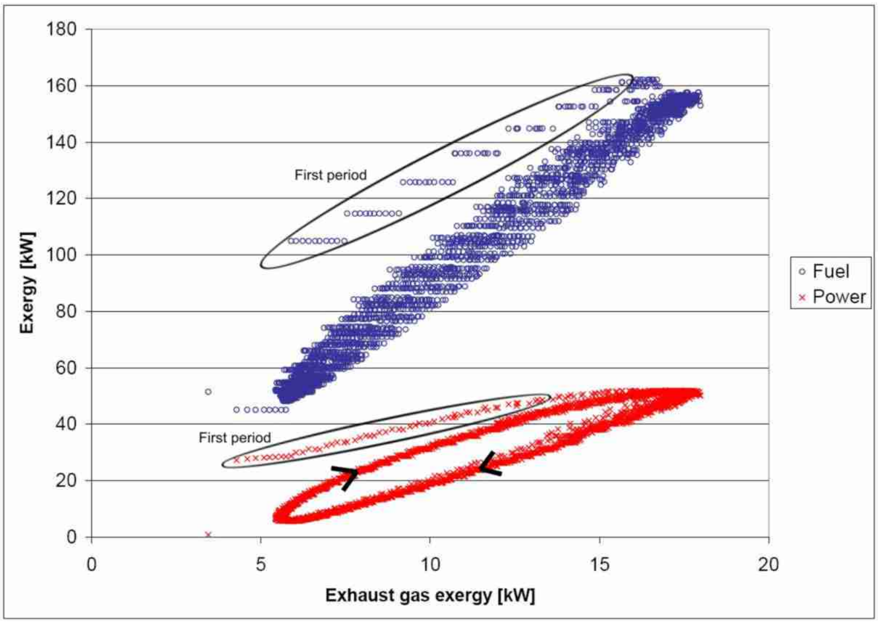

Figure 4.12: Fuel over exhaust exergy and Power over exhaust exergy instant ratios, T2 test

Exhaust gas exergy content is apparently linearly correlated to both the fuel input and the power output of the engine. Therefore, the fraction of fuel input exergy wasted in the exhaust gas is approximatively constant.

The oval pattern followed by power data (red crosses) is explained by the delay time in exhaust gas response to power demand evolution. This delay time is therefore

reference temperature was fixed to 25°C and air temperature always stands in the range between 24°C and 29°C (see table 4.3), its exergy content is clearly neglectable connected to the exhaust gas circuit inertia, whose first origin can be the physical time needed by the gas itself to pass through all the exhaust pipe up to where temperature is measured. The only part not following this pattern is related to the first transient behavior, where in fact relatively high power values are associated to low exergies.

The same reasonings can be made for inlet fuel exergy. Once again, in fact, the oval form of the correlation between exhaust gas and fuel input exergies is due to the time delay between the perturbation (fuel inlet variation) and the response (exhaust gas exergy variation). In this case, however, a new phenomenon needs to be taken care of, which is shown by the data scattering (see the difference between engine power and fuel inlet exergies behavior). Fuel consumption and its measurement have a certain delay compared to power instructions and its adjustment is in fact slower and less accurate. This point will be further developed in the 4.4.3 section.

Figure 4.13 shows the equilibrium values for exhaust exergy in transient tests together with the steady state points already shown in figure 4.5, in the form of the ratios $\frac{\text{Engine oil exergy}}{\text{Fuel inlet exergy}}$ and $\frac{\text{Engine oil exergy}}{\text{Power}}$. In this figure it is possible to see that the approximation of the fraction of fuel exergy input wasted in exhaust gas being constant is too strong, since it grows with inlet fuel exergy from about 5% to almost 15% for the highest fuel consumptions. This is not surprising, since for lower values of fuel inputs exhaust gas temperature is weak, thus having a lower exergy content. It is necessary to remark that in an exergy analysis, this increase in the exhaust gas exergy content is not connected to a reduction in engine efficiency, which could be the case for the energy analysis. The increase in exhaust gas exergy content comes from the reduction in the combustion irreversibilities due to the higher exhaust gas temperature.

It is possible to observe that transient tests results show approximatively the same values as the steady state tests do, which is, for high exergy fuel inputs, about the 12%. This means that **about the 12% of the fuel input exergy content is lost through the exhaust gas and can be recovered**. This consideration is mostly true for the $\frac{\text{Engine oil exergy}}{\text{Fuel inlet exergy}}$ ratio, while for the $\frac{\text{Engine oil exergy}}{\text{Power}}$ ratio a slight deviation from transient to steady state results can be observed.

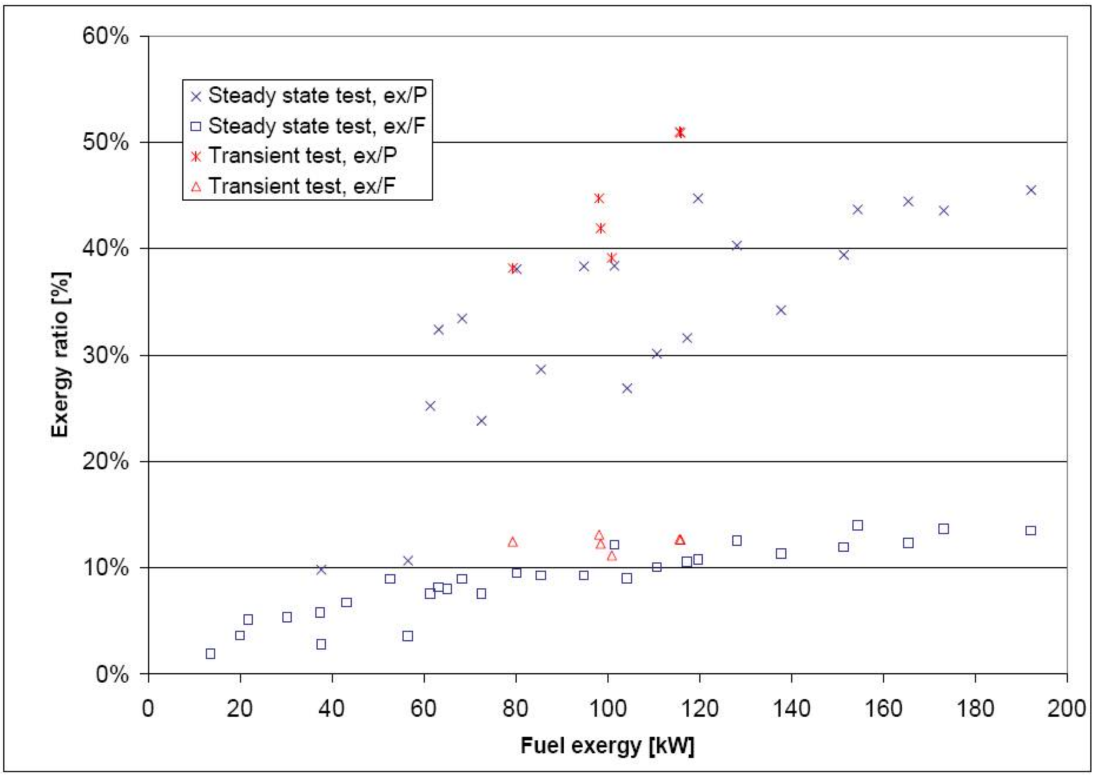

Figure 4.13: Exhaust gas over fuel exergy ratio and exhaust gas over power exergy ratio, plotted versus fuel exergy content

4.4.2 Engine oil

Figure 4.14 shows engine oil gas exergy content evolution for T1 to T3 tests, that all share the same mean power (28.8 kW) and period (300 s[1]), while the constant engine speed is set to 2350 rpm for the T1 test and to 2100 for the T2 and T3.

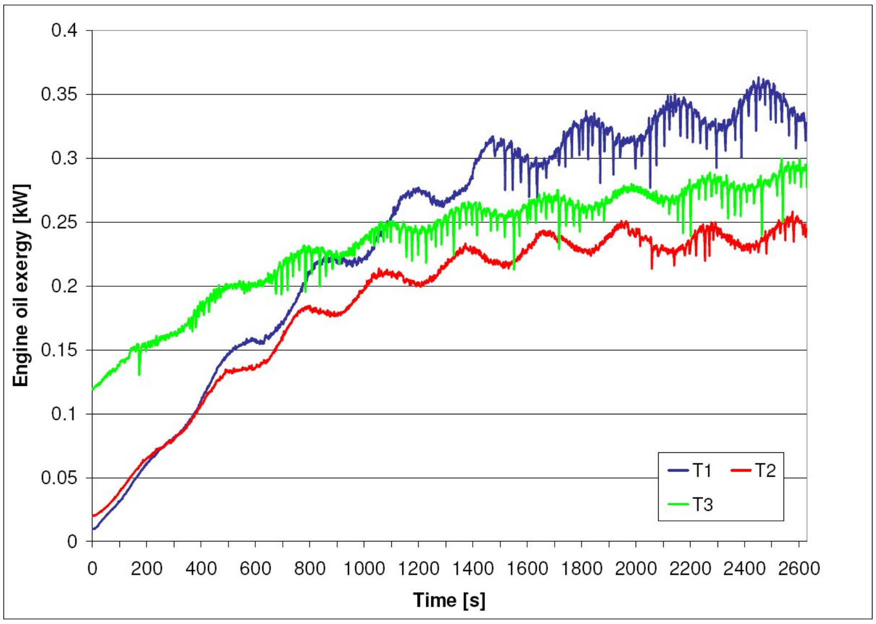

Figure 4.14: Exhaust gas exergy potential evolution

High engine oil thermal inertia appears clearly in transient tests as it does in steady-state ones. Variations in temperature (and, thus, exergy) are smoother, and the time needed to get to equilibrium is very long. This is also shown in figure 4.15, where tests T4 and T6 are taken into account, the first one having a smaller amplitude of the original oscillation, the second a shorter period. These tests show no remarkable sinusoidal behavior in engine oil exergy potential.

In order to stress exergy evolution toward the equilibrium value, figure 4.16 show exhaust gas versus engine oil mean exergies, calculated over a period. These courbes make it easier to see the difference in inertia, since exhaust gas exergy potential reaches its equilibrium value only after a short time has elapsed. Values are expressed by the

[1] From the graphics, however, there seems to be a quite strong difference between the T1 test and the others. As will be further explained, in fact, this is caused by a software problem issued during the first test that was solved for following tests. The real period of the T1 test is in fact 324 seconds

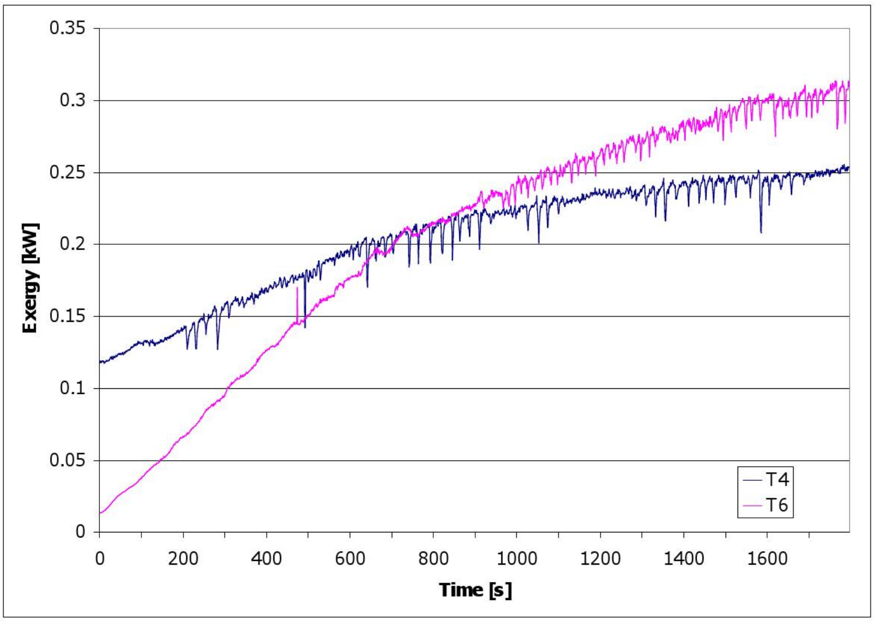

Figure 4.15: Engine oil exergy evolution, T4 and T6 tests

ratio to the steady-state value, in order to a have an easier comparison between values that would otherwise be very different one to each other.

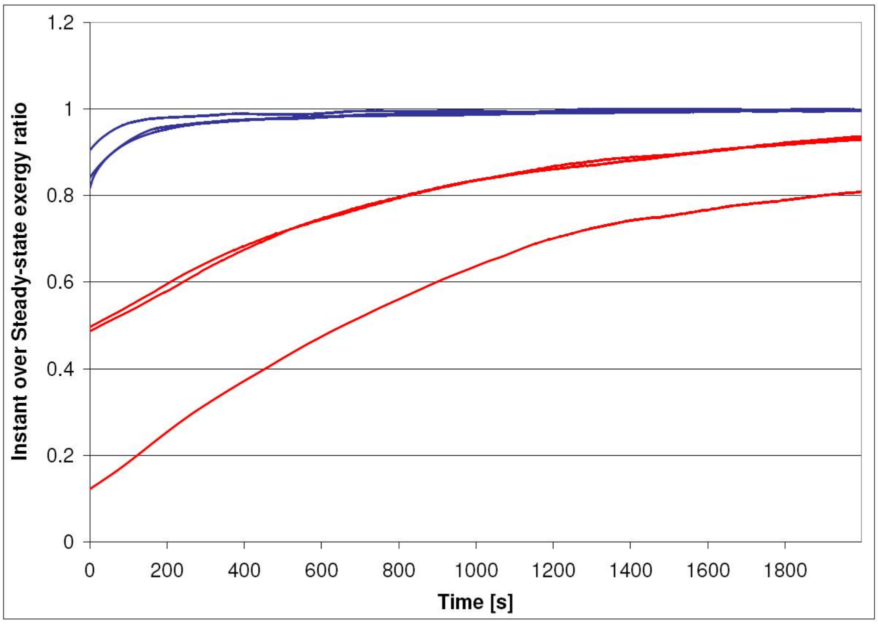

Figure 4.16: Exhaust gas and engine oil mean exergy potential evolution, T1 to T4 tests

Figure 4.17 shows the equilibrium values for engine oil exergy in transient tests together with the steady state points already shown in figure 4.9. Figure 4.17 allows to observe that transient tests results stand in the same range as steady state values, meaning that the average recoverable exergy do not change with respect to engine operations transient nature. This consideration is mostly true for the $\frac{\text{Engine oil exergy}}{\text{Fuel inlet exergy}}$ ratio, while for the $\frac{\text{Engine oil exergy}}{\text{Power}}$ ratio some deviation from the standard behavior can be observed, easily explainable with the role played by engine speed.

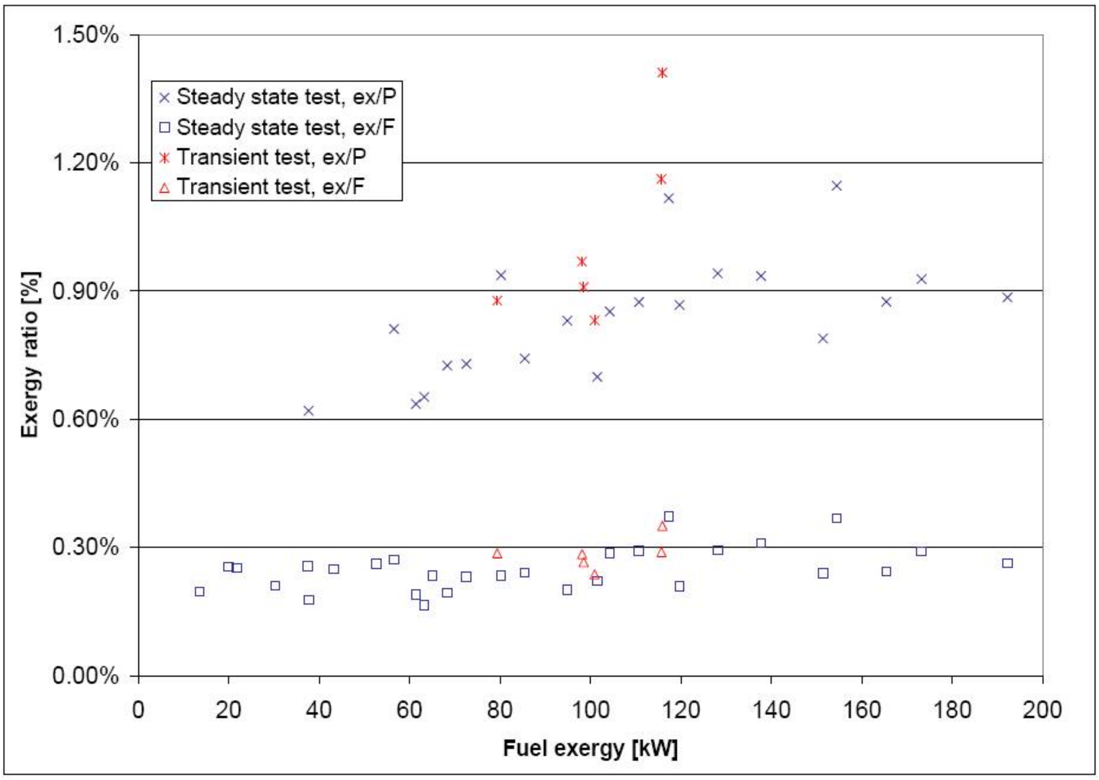

Figure 4.17: Engine over fuel exergy ratio and engine oil over power exergy ratio, plotted versus fuel exergy content

Oil exergy signal presents small disturbances, linked to the high sensitivity of the flow rate sensor. These perturbations do not remarkably influence the understanding of the graphics and can thus be neglected.

4.4.3 Test bench accuracy

The test bench used for both steady-state and transient tests is a static one. This means that for the latter, its use is, theoretically, improper.

Results of transient tests were used in order to evaluate test bench performance faced to dynamic instructions and to estimate the amplitude of transient instability.

Figures 4.18 show a comparison between power instructions and direct measurements for T1 test.

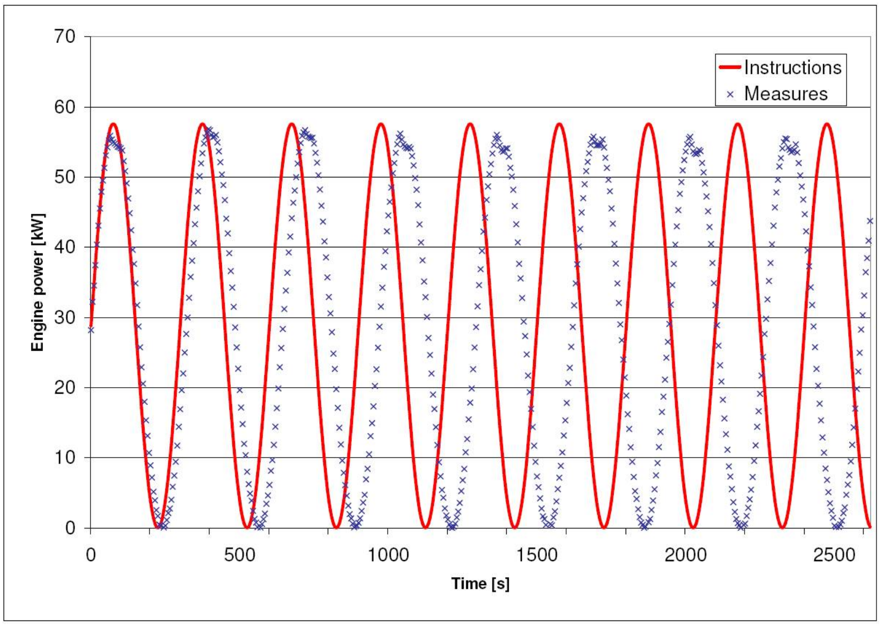

Figure 4.18: Engine power instructions versus measures, T1 test

T1 shows two pronounced differences between measures and instructions:

- Measured period of T1 test was 324 seconds, while it was supposed to be 300 seconds, meaning that the delay between the two courbes increases over time. This was caused by an improper use of test bench software, whose further development made it possible to solve this problem. Further tests, like T2, were successfully conducted.

- At high engine power values the tractor could not achieve values demanded by instructions. This was due to the fact that the engine is not always able to reach its maximum power values, depending on external conditions. This problem was solved simply by reducing the amplitude of power variations.

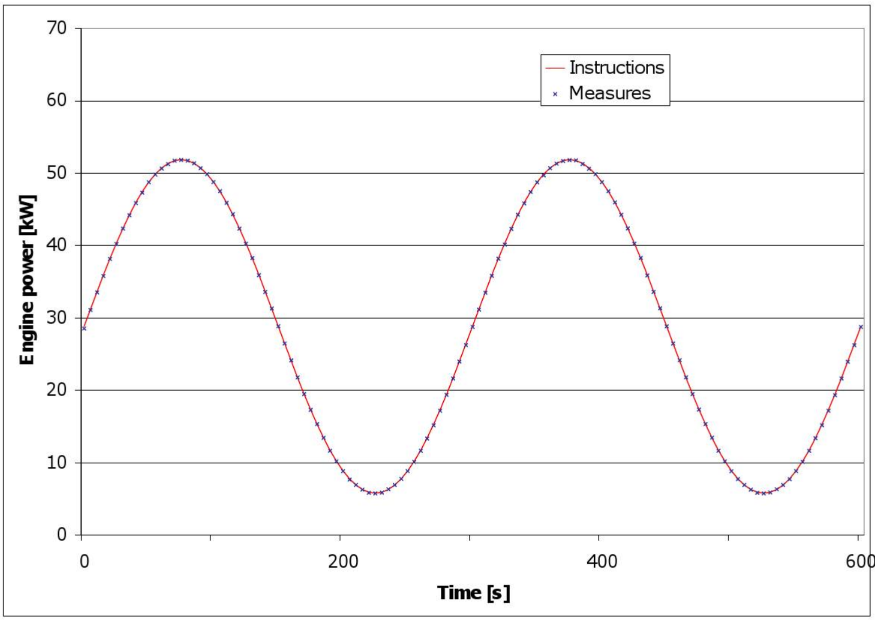

Figure 4.19: Engine power instructions versus measures, T2 test

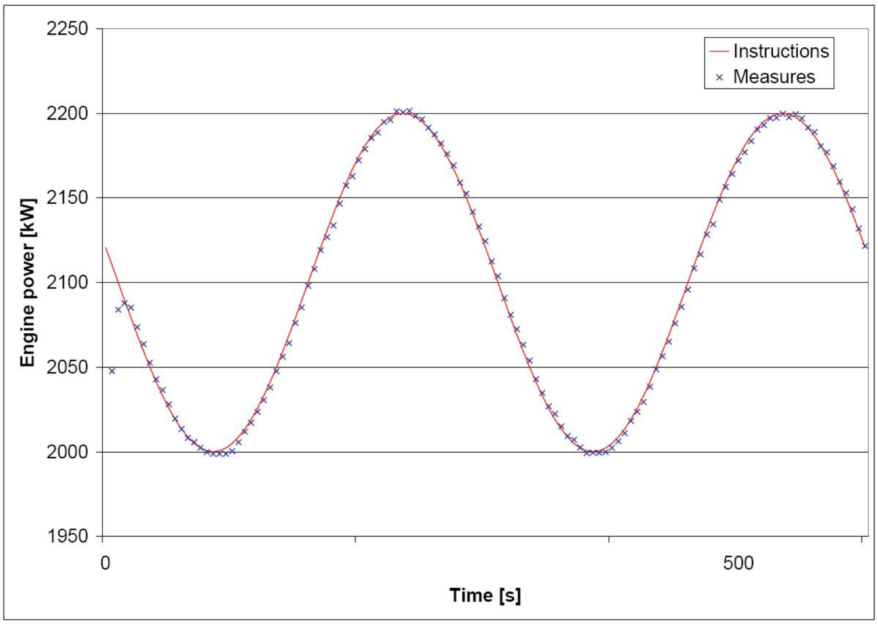

Figure 4.20: Engine speed instructions versus measures, T4 test

Figure 4.19 and 4.20, referred to the first two periods of the T2 test, shows the gaps between instructions and measured values for engine power and speed. In this kind of tests, it is possible to notice that test bench accuracy is precise enough for work's aim, even if slightly small discrepancies can be observed when engine speed is the controlled variable.

Figure 4.21, referred to the first two periods of T2 test, shows instead the response in fuel mass flow (here presented with its exergy content, where only the constant LHV value stands between). The delay between the two signals is a consequence of two main effects:

- Fuel mass flow is adjusted by the engine, while the measurement is placed on the pipe connecting test bench storage to the fuel pump. This means that before being able to measure a variation in mass flow, the system needs to wait for the fuel to go through all the pipe, thus leading to a delay time.
- Throttle adjustment as a consequence of variations in power and speed instructions is not very accurate, since it is a variable dependent on other parameters. This is the reason why small oscillations are generally observable in the first part of the transient between two different instructions.

These last observations give a better explanation of the phenomenon shown in figure 4.12. Mass flow measurement is not accurate enough to highlight the correlation with exhaust gas exergy potential with the same precision as done in the case of engine power.

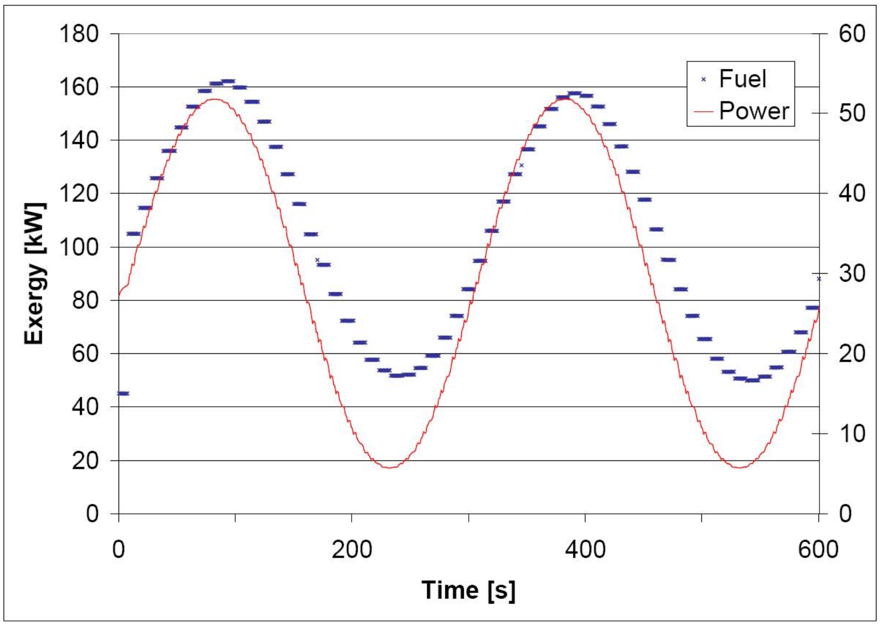

Figure 4.21: Engine power versus fuel exergy inlet, T2 test

4.5 Influence of heat exchange properties on recovery potential

As said in the conclusive part of the first chapter, one more constraint concerning WHR systems is heat exchange surface. In this work the possibility of recovering waste heat on two different circuits, each of them with different fluids, mass flows and temperatures, deeply affects features concerning heat exchangers that have to be used in order to extract these quantities of energy.[1]

This problem has to be taken into account in particular when the application to vehicles is to be studied. In stationary applications, in fact, there are usually no constraints on weight and space, which means that the only issue concerning heat exchangers is their influence on the economics of the system. In vehicles applications, instead, weight and space are major issues while considering system feasibility. Space is, in fact, limited, and increasing exchange surfaces too much can lead to unrealistic solutions; weight, instead, is not really limited, but it is clear that increasing weight corresponds to decreasing efficiency.

For this reason a third kind of test has to be planned, where heat transfer is taken into account. This can be done placing a heat exchanger of well-defined properties (exchange surface, material, etc) on the two different circuits. The results of these tests can give an indication of the heat exchange quality depending on waste heat source.

4.5.1 Heat exchange exergy analysis

An exergy analysis of the heat exchanges is performed, allowing to understand where exergy is exchanged and destroyed. The objective of this analysis is to compare the two sources with reference to the same exchange surface (always keeping in mind, however, that conditions in a real application would be very different).

Exergy analysis involves first of all an estimation of exergy values for every flow entering or leaving the heat exchanger. This involves using the standard equation for exergy calculation (equation 4.1).

[1] This issue actually depends on what kind of WHR system is emplogasyed. In the case of an Inverted Brayton Cycle (IBC) recovering only exhaust gas potential, there would be no heat exchanger, thus making heat surface issue totally unimportant. However, in this study, reference will be made to a more standard ORC or Stirling cycle, where energy transfer into the cycle always need a heat exchanger

Exergy and energy values for all inlet and outlet flow are thus well known. It is therefore possible to estimate the amount of exergy lost by the hot flow as well as the amount gained by the cold one. In order to understand the correlation between energy and exergy flows through the heat exchanger, in the analysis the following definitions of WHR potential exchange efficiency, related to the thermal exchange, will be used:

$$\eta_{WHR,h} = \frac{ex_{h,i} - ex_{h,o}}{en_{h,i} - en_{h,o}} \tag{4.13}$$

$$\eta_{WHR,c} = \frac{ex_{c,o} - ex_{c,i}}{en_{c,o} - en_{c,i}} \tag{4.14}$$

where h and c stand respectively for the hot and the cold flow, and i and o for inlet and outlet flow. In the following part of this work these quantities will simply be called "exergy efficiency" for a matter of simpleness [1]. This kind of efficiency allows to understand how much of the energy that is exchanged is really able to produce useful work. These efficiencies will clearly result very low because of the small heat exchange surface, especially on the cold side.

It is important to remark the new choice for reference temperature in this new case. For steady state tests a temperature of 298 K was chosen, representative of the "normal" conditions often used in engineering. In this case, however, new elements come to light making it necessary to question this choice. Water inlet temperature at the heat exchanger always stands between 9 and 11 °C, that is below reference temperature. This implies that **heating up the water in the heat exchanger would decrease its exergy content**, making no sense for the analysis.

For this reason, the new reference temperature chosen for this kind of test was the lowest measured temperature in the heat exchanger, that the measure taken at the inlet section of the heat exchanger on the water side. This choice is justified by the definition of exergy, and a better description of the choice of the reference temperature is given in section 2.2.

4.5.2 Hot side

The external side of the heat exchanger is crossed by the hot flow, this being exhaust gas or engine oil depending on the kind of experiments.

[1] It is to be remarked that the exergy efficiency of a heat exchange is generally defined in a different manner. However, once the convention is defined and clear, any definition is valid once it is useful for author's objective

Figure 4.22 shows the value of the exergy efficiency for exhaust gas experiments on the hot side of the heat exchanger. Blue crosses represent results from steady-state tests, while red circles represent transient tests results instead.

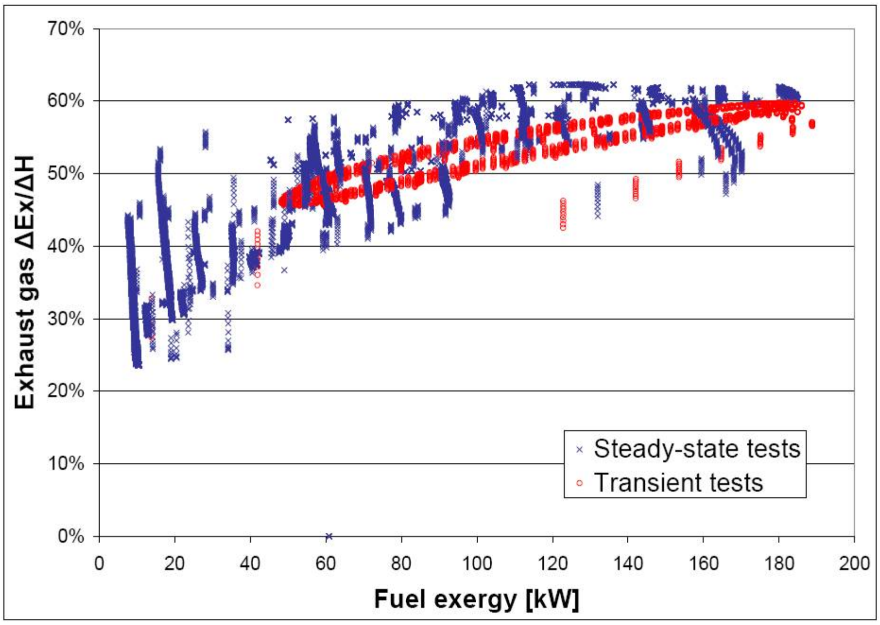

Figure 4.22: Exhaust gas experiments, exergy efficiency of the heat exchange on the hot side versus fuel input exergy content

The exergy efficiency of the heat exchange, defined by equation 4.14, is a direct function of fluid temperature. This explains why on the hot side of the heat exchanger for exhaust gas experiments this value is relatively high. Moreover, this considerations is able to give an explanation of the fact that exergy efficiency values grows with inlet fuel exergy: the same tendency can be observed for exhaust gas temperature.

It can be observed that steady state and transient tests share very similar values for exergy efficiency.

Figure 4.23 shows the value of the exergy efficiency for engine oil experiments on the hot side of the heat exchanger. Blue crosses represent results from steady-state tests, while red circles represent transient tests results instead.

The exergy efficiency of the heat exchange is, this time, very low, once again because of fluid temperature. Inlet fuel exergy dependency of this efficiency is explained by the same reasons given in the case of exhaust gas experiments. Once again, there is

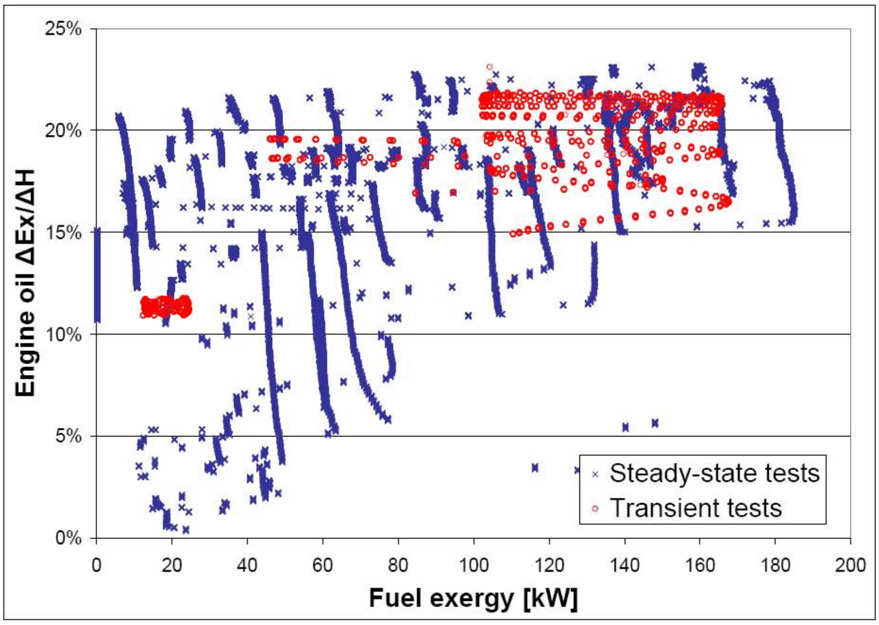

Figure 4.23: Engine oil experiments, exergy efficiency of the heat exchange on the hot side versus fuel input exergy content

apparently no influence on the efficiency given by the steady-state or transient nature of the test.

In this case, however, a strong data dispersion can be observed. This behavior is a consequence of engine oil strong thermal inertia.

4.5.3 Cold side

The cold side is in both engine oil and exhaust gas experiments made by a water flow coming from the bench tap. Inlet temperature for water stands, as said, between 9 and 11 °C, while its value at the outlet section strongly depends on the circuit and on engine operations. Water temperature increase for exhaust gas experiments can reach 18°C, while for engine oil experiments it rarely exceeds 4°C.[1]

[1]This last consideration arises some doubts about the accuracy of this measure. Thermocouples accuracy is in fact included in the range of $\pm 1K$, meaning that measure incertitude can strongly affect these results. However, as will be shown in the following part of this work, these data will never be used stand-alone, but in statistically significant quantities and they will be compared ones to each others. This kind of analysis assures reliable results even in case of a certain degree of inaccuracy. Concerns related to measurement accuracy are further detailed in Appendix D

Figure 4.24 shows the exergy efficiency value for exhaust gas experiments on the cold side of the heat exchanger. Blue crosses represent results from steady-state tests, while red circles represent transient tests results instead.

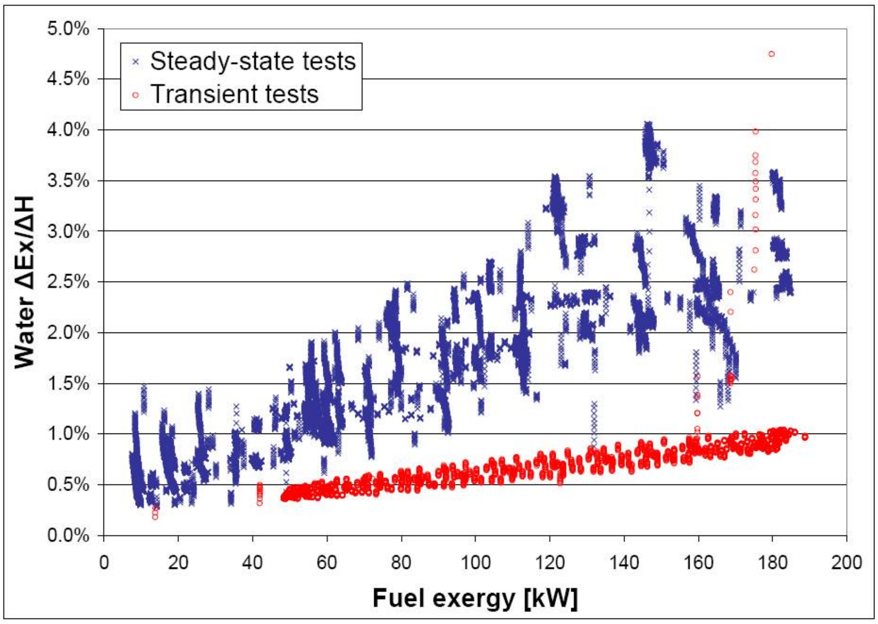

Figure 4.24: Exhaust gas experiments, exergy efficiency of the heat exchange on the cold side versus fuel input exergy content

In this case the exchange efficiency is extremely low because of water temperature (always $< 5\%$). However, it is very interesting to observe the difference between transient and steady state tests. It clearly appears from the graphics that the exergy efficiency of the heat exchange is remarkably different between steady state and transient tests, the latter being the lower. Analysis is still in work in order to give a theoretical explanation to this phenomenon, but it still interesting to observe that **a transient behavior of the engine lead to decreasing heat exchange exergy efficiency**. This means that the inertia of the heat exchanger is responsible of a remarkable decrease in the amount of work it is possible to recovery from a given heat source, in the case of this varying with time.

Figure 4.25 shows the value of the exergy efficiency for engine oil experiments on the cold side of the heat exchanger. Blue crosses represent results from steady-state tests, while red circles represent transient tests results instead.

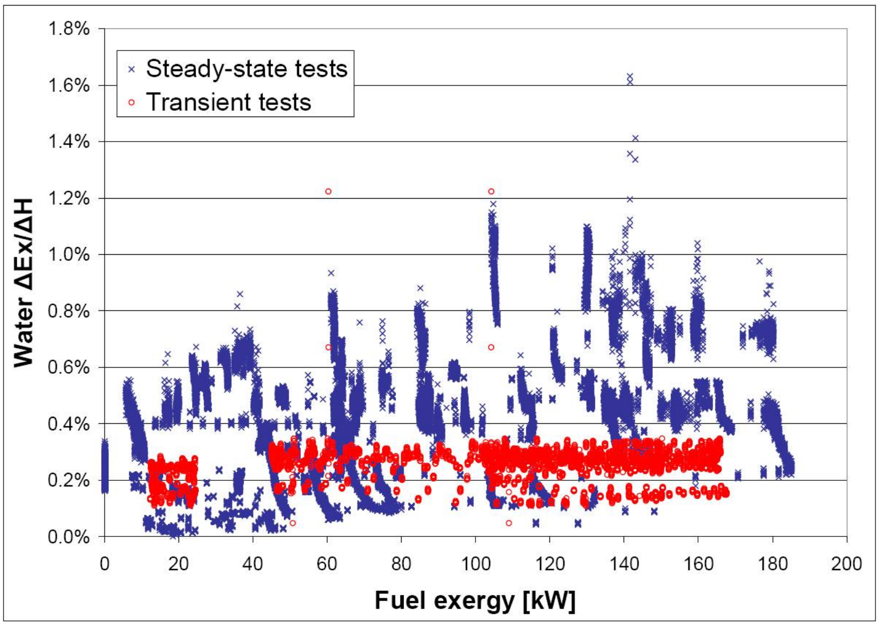

Figure 4.25: Engine oil experiments, exergy efficiency of the heat exchange on the cold side versus fuel input exergy content

The same phenomenon observed for exhaust gas can be seen in the case of engine oil. However, for this test, the effect is weaker. This is due to the fact that those that are here called "steady-state tests“ do not feature, in the case of engine oil, a real steady state behavior, because of engine oil strong thermal inertia. However there is no doubt that transient tests show a lower exergy efficiency of the heat exchange than steady-state ones.

5

Conclusions

The goal of this work was to study the WHR potential for the Diesel engine of a tractor, with particular attention to its evolution in transient operations. For this purpose, experimental tests were performed, and data thus obtained were studied, the results of this analysis being detailed in the previous chapter of this work.

Steady state tests series and the following analysis led to the conclusion that exhaust gas is a very good candidate for waste heat recovery. Approximatively 25 kW of additional useful mechanical power could be produced (in ideal and reversible conditions) from the exploitation of this source of heat, that compared to the 57 kW of power directly produced by the engine, would make increasing it by nearly the 45%. The percentage of the fuel input exergy content wasted through exhaust gas showed to be slightly growing with fuel consumption, reaching values of 15% at nominal power and speed.

The same kind of tests performed for the engine oil showed instead that this latter source of heat is not interesting for waste heat recovery. Its exergy content never exceeds 600 W, a value not even comparable to what measured in exhaust gas experiments.

Transient phenomena showed a very different behavior for exhaust gas and engine oil, the latter having a strong thermal inertia. The time constant associated to these circuits is not unique, especially for engine oil where measured values stands between 300 and 900 seconds.

The analysis of the phenomena related to the heat exchange led to the unexpected observation that the recoverable fraction of heat exchanged decreases during transient operations. This can be observed on both exhaust gas and engine oil heat exchanger.

6

Perspectives

6.1 Engine oil temperature evolution: engine thermal model

Engine oil recovery potential being low, especially compared to what measured for exhaust gas, it has almost no interest in term of waste heat recovery. However, its complex behavior connected to its high thermal inertia is something worth to be further studied.

The results of this work about WHR potential of tractors showed to have a side interest also for the project that Cemagref is carrying on in the context of the partnership with Total about oil quality influence on tractors performances.

In this work, engine oil temperature evolution over time was simply modeled with an exponential regression based on measured data. This solution allowed the estimation of an equilibrium temperature that was considered to be a good approximation of the real value. However, the situation is quite different. Several variables play their role in engine oil evolution, and even if the approximation of an exponential behavior is largely sufficient for the ambitions of this work, if it is necessary to have a better knowledge of this subject, it has to be studied with a completely different approach. The role of the thermal exchange with the environment also showed to be quite critical, even if only at a second degree approximation degree.

This study was initialized in parallel with the subject of waste heat recovery, but is still only in the start-up phase and much more work would be needed in order to have satisfactory results. Figure 6.1 show the starting hypothesis of thermal model for the engine.

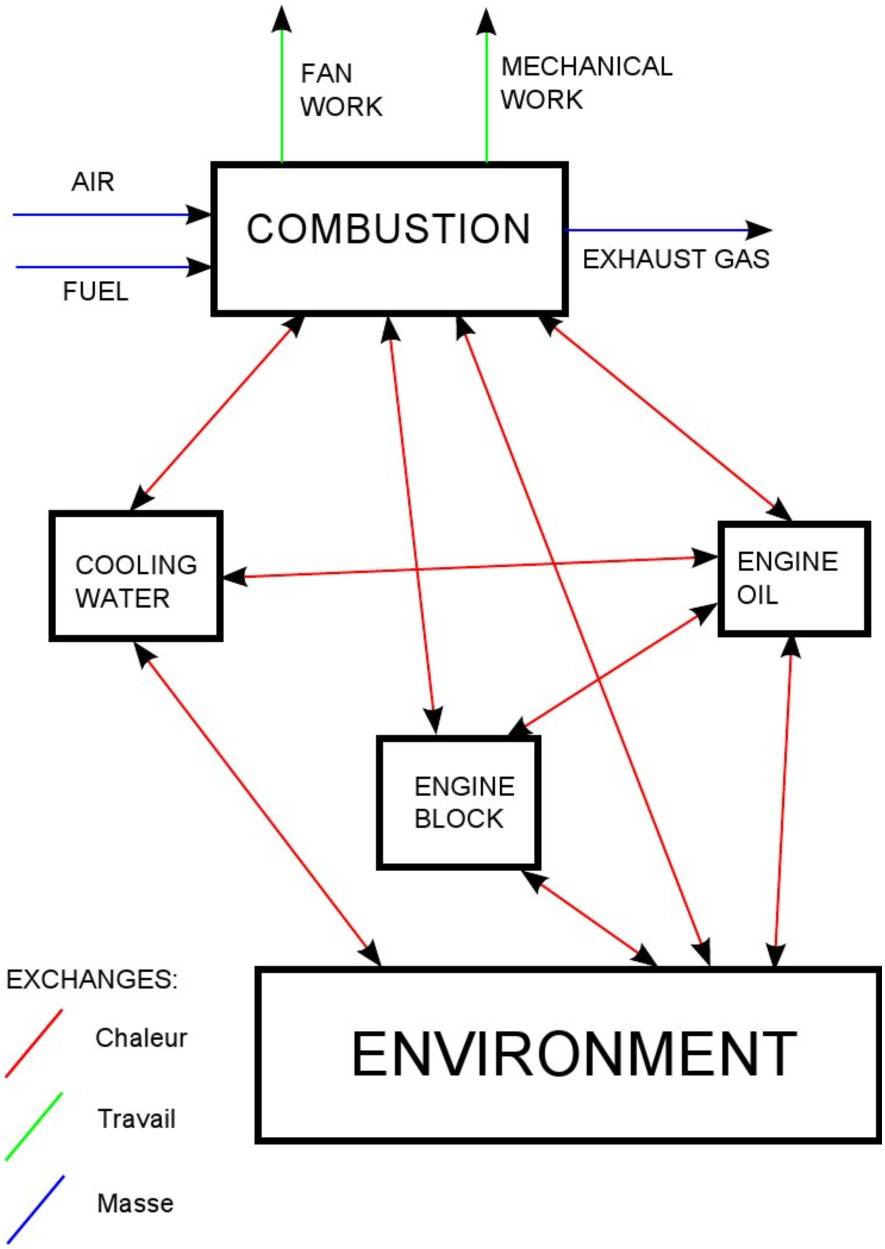

Figure 6.1: Engine thermal model for engine oil temperature calculation

Fuel and air input are considered to be the only energy source of the system. The energy thus produced by fuel combustion is redirected to several outputs:

- Useful work
- Fan work

- Exhaust gas
- Heat exchanges

If the first three outputs do not need any explanation, the fourth and latter is the critical part of the model. As shown in figure 6.1, three different components are considered in order to explain system inertia:

- Engine oil circuit
- Cooling water circuit
- Engine block

All these entities exchange heat with the combustion, with the environment and one to each other. In order to reduce the complexity of the model, the approximation here made was to consider each of these entities to have only one temperature representative of its energy content and of thermal exchanges with other components. This approximation allows to write the following equations for engine thermal equilibrium:

$$\begin{aligned} m_{eo}c_{eo}\frac{\partial T_{eo}}{\partial t} &= \alpha_{eo,eb}(T_{eo}-T_{eb})+\alpha_{eo,cw}(T_{eo}-T_{cw})+\alpha_{eo,a}(T_{eo}-T_a) \\ m_{cw}c_{cw}\frac{\partial T_{cw}}{\partial t} &= \alpha_{cw,eb}(T_{cw}-T_{eb})+\alpha_{eo,cw}(T_{cw}-T_{eo})+\alpha_{cw,a}(T_{cw}-T_a) \\ m_{eb}c_{eb}\frac{\partial T_{eb}}{\partial t} &= \alpha_{eo,eb}(T_{eb}-T_{eo})+\alpha_{eb,cw}(T_{eb}-T_{cw})+\alpha_{eb,a}(T_{eb}-T_a) \end{aligned}$$

These three equations should eventually be combined with a fourth one considering the whole tractor as a control volume:

$$\dot{m}_f LHV + (\dot{m}_f+\dot{m}_a)(T_a-T_0) - \dot{W}_u - \dot{W}_f - (\dot{m}_f+\dot{m}_a)(T_{eg}-T_0) =$$

$$= \alpha_{eo,a}(T_{eo}-Ta)+\alpha_{cw,a}(T_{cw}-Ta)+\alpha_{eb,a}(T_{eb}-Ta)$$

However, the analysis of the thermal model still needs a large amount of work. More efforts are needed, both in terms of experimental tests and data analysis, in order to verify the validity of the model and the consistency of the approximations and the hypothesis.

6.2 Thermal exchange

One of the aims of this study was to understand heat exchange-related phenomena, and in this sense maybe the most relevant result of this work is the influence that transient operations showed to have on heat transfer exergy efficiency. However, a deeper analysis of this kind of phenomena is needed, in order to verify the observations, and to give a complete theoretical explanation that is, until now, partially missing.

However, the interest in thermal exchange cannot stop to the detection of a reduction in exergy potential connected to transient behavior. Several more subjects still need to be treated, in order to have a better knowledge of these phenomena and their influence on WHR potential

First of all the influence of heat exchanger features and their optimisation. Every parameter, such as exchanger type (anular, plate, etc), dimensions, position, play a role. The pressure drop on the exhaust pipe added by exchanger presence should be taken into account also, since it can influence engine performances and, thus, change our conclusions about exhaust gas WHR potential.

Appendix A

Coefficients and parameters

Technical features		
Model		Renault 851-4 R 7664
Cylinders		4
Volume	cm^3	4156
Stroke	cm	120
Cylinder cross section	cm^2	105
Engine performances		
Nominal power	kW	57.6
Nominal speed	rpm	2350
Nominal Torque	Nm	233.4
Maximum power	kW	57.6
Maximum speed	rpm	2480
Maximim torque	Nm	277

Table A.1: Tractor technical features and engine performances

Regression coefficients		a_0	a_1	a_2	a_3
Exhaust gas					
Specific heat (CO_2)	$\frac{J}{kgK}$	4.15E-1	1.91E-3	-1.79E-6	7.39E-10
Specific heat (H_2O)	$\frac{J}{kgK}$	2.94	-6.81E-3	1.42E-5	-9.09E-9
Specific heat (O_2)	$\frac{J}{kgK}$	9.73E-1	-6.35E-4	1.89E-6	-1.24E-9
Specific heat (N_2)	$\frac{J}{kgK}$	1.09	-3.11E-4	5.71E-7	-1.43E-10
Engine oil					
Specific heat	$\frac{J}{kgK}$	622	4.29		
Density	$\frac{kg}{m^3}$	1061	-0.592		
Volumetric flow	$\frac{l}{h}$	-366	1.24		

Table A.2: Exhaust gas and engine oil polynomial regressions coefficients

Appendix B

Results

Instructions		Measured values		Calculated values	
Power	Speed	Temperature	Mass flow	Specific exergy	Exergy flow
kW	rpm	K	$\frac{kg}{s}$	$\frac{kJ}{kgK}$	kW
57.0	2340	751.2	0.133	195.16	25.945
46.0	2370	698.2	0.129	158.99	20.451
28.8	2350	618.2	0.118	109.48	12.887
5.7	2350	498.6	0.106	49.04	5.196
0.0	2350	490.2	0.104	45.41	4.722
54.4	2090	756.2	0.119	198.65	23.708
46.0	2115	697.2	0.114	158.50	18.135
23.0	2115	573.2	0.104	84.78	8.822
0.0	2110	452.2	0.094	30.80	2.903
49.6	1900	767.2	0.105	206.45	21.667
40.0	1900	704.2	0.099	162.91	16.123
20.0	1900	572.2	0.090	84.20	7.618
0.0	1900	437.2	0.085	25.66	2.172
45.7	1640	763.2	0.076	205.53	15.648
32.2	1640	723.2	0.070	176.13	12.363
18.3	1640	565.2	0.076	80.64	6.122
16.0	1640	537.2	0.078	66.69	5.187
0.0	1640	425.2	0.074	21.80	1.621
39.2	1410	775.2	0.058	215.42	12.394
35.0	1410	699.2	0.058	161.84	9.406
23.0	1410	591.2	0.058	95.14	5.481
0.0	1410	411.6	0.063	17.75	1.121
37.0	1300	745.2	0.058	193.53	11.147
27.7	1300	681.2	0.053	149.44	7.940
18.4	1300	587.2	0.050	92.90	4.642
0.0	1300	397.2	0.053	13.86	0.729
18.9	940	602.2	0.019	105.66	2.018
10.8	940	514.2	0.018	57.81	1.060
0.0	950	380.2	0.026	9.82	0.259

Table B.1: Exhaust gas experiments, measured and calculated values for steady-state tests

Instructions		Measured values		Calculated values			
Power	Speed	Temperature	Volumetric flow	Density	Mass flow	Specific exergy	Exergy flow
kW	rpm	K	$\frac{l}{h}$	$\frac{kg}{m^3}$	$\frac{kg}{s}$	$\frac{kJ}{kgK}$	kW
57.0	2340	380.4	106.13	836.0	0.0246	20.473	0.505
46.0	2370	374.1	98.38	839.7	0.0229	17.543	0.403
28.8	2350	362.5	83.91	846.6	0.0197	12.659	0.250
5.7	2350	352.3	71.26	852.7	0.0169	9.023	0.152
0.0	2350	350.4	68.88	853.8	0.0163	8.406	0.137
54.4	2090	380.4	106.15	836.0	0.0247	20.479	0.505
46.0	2115	371.4	95.03	841.3	0.0222	16.343	0.363
23.0	2115	356.8	76.78	850.0	0.0181	10.537	0.191
0.0	2110	346.1	63.59	856.3	0.0151	7.109	0.108
49.6	1900	383.9	110.50	834.0	0.0256	22.220	0.569
40.0	1900	372.4	96.19	840.8	0.0225	16.755	0.376
20.0	1900	356.4	76.30	850.3	0.0180	10.399	0.187
0.0	1900	344.2	61.22	857.4	0.0146	6.563	0.096
45.7	1640	375.7	100.37	838.8	0.0234	18.275	0.427
32.2	1640	360.2	81.08	848.0	0.0191	11.795	0.225
18.3	1640	349.8	68.13	854.1	0.0162	8.215	0.133
16.0	1640	345.6	62.97	856.6	0.0150	6.964	0.104
0.0	1640	338.1	53.64	861.1	0.0128	4.958	0.064
39.2	1410	376.4	101.20	838.4	0.0236	18.584	0.438
35.0	1410	366.6	89.01	844.2	0.0209	14.291	0.298
23.0	1410	354.2	73.57	851.6	0.0174	9.641	0.168
0.0	1410	336.2	51.18	862.2	0.0123	4.485	0.055
37.0	1300	368.6	91.44	843.0	0.0214	15.104	0.323
27.7	1300	358.3	78.67	849.1	0.0186	11.082	0.206
18.4	1300	347.6	65.36	855.5	0.0155	7.530	0.117
0.0	1300	335.1	49.93	862.8	0.0120	4.254	0.051
18.9	940	352.4	71.43	852.6	0.0169	9.067	0.153
10.8	940	338.8	54.53	860.6	0.0130	5.136	0.067
0.0	950	327.6	40.62	867.3	0.0098	2.717	0.027

Table B.2: Engine oil experiments, measured and calculated values for steady-state tests

Appendix C

Exhaust gas composition approximation

Exhaust gas composition had to be calculated in order to evaluate exhaust gas specific heat at constant pressure.

For this, some approximations were made:

- Exhaust gas is only composed by carbon dioxide (CO_2), steam (H_2O), oxygen (O_2) and nitrogen (N_2). All contribution given by unburned carbon, soot, carbon monoxide and any other gas is neglected

- The complete combustion approximation is considered, in the sense that every carbon and hydrogen molecule entering the combustion chamber is converted into carbon dioxide and steam

- Contribution of inlet air humidity content is neglected. This is maybe the strongest hypothesis, but still not severely affects calculation accuracy

- Air composition is supposed of 79% nitrogen and 21% oxygen on molecular basis

First of all, the A/F ratio (inlet air mass flow over inlet fuel mass flow) value can be calculated starting from measured value of exhaust gas mass ($\dot{m}_{eg}$) and inlet fuel ($\dot{m}_f$) mass flows

$$\frac{A}{F} = \frac{\dot{m}_{eg}}{\dot{m}_f} - 1$$

Carbon molar flow ($\dot{n}_C$). It directly depends on fuel mass flow:

$$\dot{n}_C = \frac{\dot{m}_f}{(M_C + M_H \frac{H}{C})}$$

Where $\frac{H}{C}$ stands for the atomic ratio between hydrogen and carbon in the fuel.

Hydrogen molar flow ($\dot{n}_H$) is the directly consequent of the $\frac{H}{C}$ ratio.

$$\dot{n}_H = \dot{n}_C \frac{H}{C}$$

Supposing, as said, complete combustion, carbon dioxide and steam molecular flow in exhaust gas are directly dependent on carbon and hydrogen molecular flows in fuel:

$$\begin{aligned} \dot{n}_{CO_2} &= \dot{n}_C \\ \dot{n}_{H_2O} &= \frac{1}{2}\dot{n}_H \end{aligned}$$

Nitrogen molecular flow can thus be calculated as following:

$$\dot{n}_{N_2} = \frac{79}{21}\frac{A}{F}\dot{n}_C \frac{M_C + \frac{H}{C}M_H}{M_{O2} + \frac{79}{21}M_{N2}}$$

while oxygen molecular flow is the result of a balance between inlet and outlet flows:

$$\dot{n}_{O_2} = \dot{n}_{N_2}\frac{21}{79} - \dot{n}_{CO_2} - \frac{1}{2}\dot{n}_{H_2O}$$

Indicating molecular fractions with letter x, (for mass fractions letter y will be used), they can be easily calculated thanks to values we are already in posses. "TOT" index refers to the entire exhaust gas current ($\dot{n}_{tot} = \dot{n}_{CO_2} + \dot{n}_{H_2O} + \dot{n}_{N_2} + \dot{n}_{O_2}$):

$$\begin{aligned} x_{CO_2} &= \frac{\dot{n}_{CO_2}}{\dot{n}_{tot}} \\ x_{H_2O} &= \frac{\dot{n}_{H_2O}}{\dot{n}_{tot}} \\ x_{O_2} &= \frac{\dot{n}_{O_2}}{\dot{n}_{tot}} \\ x_{N_2} &= 1 - x_{CO_2} - x_{H_2O} - x_{O_2} \end{aligned}$$

where the nitrogen molar fraction is calculated as complement to one of other molecular fractions

Specific heat at constant pressure coefficients are given with reference to gas mass fractions. For this reason this value needs to be calculated as following:

$$\begin{aligned} y_{CO_2} &= x_{CO_2}\frac{M_{CO_2}}{M_{tot}} \\ y_{H_2O} &= x_{H_2O}\frac{M_{H_2O}}{M_{tot}} \\ y_{O_2} &= x_{O_2}\frac{M_{O_2}}{M_{tot}} \\ y_{N_2} &= 1 - y_{CO_2} - y_{H_2O} - y_{O_2} \end{aligned}$$

where the mean molecular mass of the mixture had been calculated as follows:

$$M_{tot} = x_{CO_2}M_{CO_2} + x_{H_2O}M_{H_2O} + x_{O_2}M_{O_2} + x_{N_2}M_{N_2}$$

Values here calculated are then used in order to calculate each gas weight on the total value for exhaust gas specific heat at constant pressure, where regression coefficients for each component are given in appendix A

Appendix D

Measurement accuracy

D.1 Temperatures

Several different temperature values needed to be measured at the same time during test series, thus needing a number of thermocouples.

Most of thermocouples employed for this task are of type T (copperâconstantan). These thermocouples are suited for measurements in the â200 to 350 Â°C range and are often used as a differential measurement since only copper wire touches the probes. Since both conductors are non-magnetic, there is no Curie point and thus no abrupt change in characteristics. This kind of thermocouples was employed for measuring engine oil, water (both in heat exchanger and in radiator), fuel and air temperature. The type T thermocouples used during tests had a declared accuracy of ± 1 °C.

All type T thermocouples used during the tests were calibrated through the comparison with a standard thermocouple associated to a device (see figure D.1 capable of a very accurate control on temperature. All thermocouples were put inside the device, where water was kept at a fixed temperature. The comparaison of each measurement with the standard one for several different temperatures (whose choice was connected to each thermocouples role) was repeated for three tests. The results made it possible to have a satisfactory calibration of every type T thermocouple.

Two thermocouples, both of them placed on the exhaust gas because of its higher temperature, are of Type K (chromel[1]âalumel[2]). These are inexpensive and very commonly used thermocouples, and a wide variety of probes are available in their â200 Â°C

[1] 90 percent nickel and 10 percent chromium

[2] Alumel consisting of 95% nickel, 2% manganese, 2% aluminium and 1% silicon

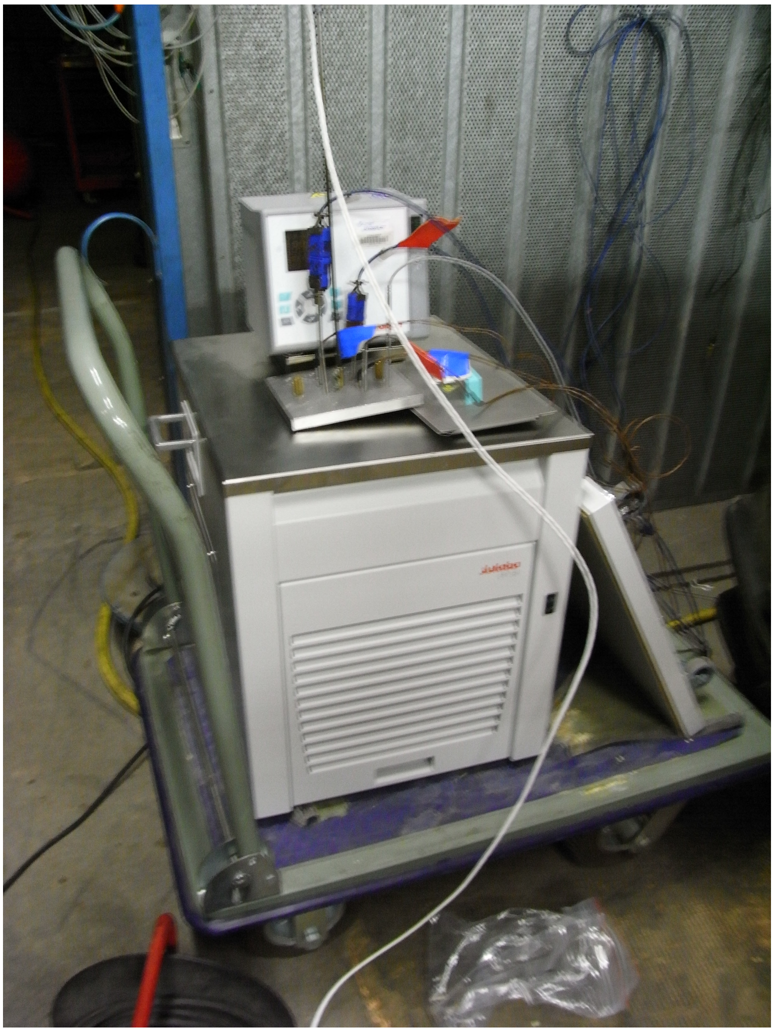

Figure D.1: Calibrating device

to +1350 Â°C range. Type K was defined at a time when metallurgy was less advanced than it is today, and consequently characteristics vary considerably between samples. One of the constituent metals, nickel, is magnetic; a characteristic of thermocouples made with magnetic material is that they undergo a step change in output when the magnetic material reaches its Curie point (around 354 Â°C for type K thermocouples). However, this phenomenon was never observed during tests even if exhaust gas temperature crossed several times this "barrier". Its influence on results is, thus, minimal.

No device was available for calibrating type K thermocouples in their working range (100 °C to 500 °C), since the calibrating system mentioned for type T thermocouples cannot exceed 100 °C. However, both type K thermocouples belonged to the test bench, and they have been successfully used for several years

All temperature measurements proved to be worth of trust. However, temperature measurements on the heat exchanger cold side for engine oil experiments presented quite of an issue. Because of the heat exchanger small surface and of engine oil low temperature, the increase in water temperature from inlet to outlet section was small, never exceeding 5 K and often being of approximatively 2-3 K. This situation lead measure incertitude to play a very important role, since its absolute value has the same magnitude as mentioned temperature difference.

D.2 Exhaust gas mass flow

Exhaust gas exergy content evaluation need a value of exhaust gas mass flow. This information is directly measured by the gas analyzer HORIBA OBS-2200 thanks to a Pitot tube, capabel of measuring exhaust gas velocity through a double pressure measurement and, from that, calculate mass flow value.

However this “direct” measurement of exhaust gas mass flow presents more than a deal. First of all, the oscillation of the measured value. This phenomenon, which is increasingly pronounced as the mass flow decreases, is easily solved by averaging measured value for each operational condition. This, however, invites to beware of Pitot tube measures.

A second problem concerns the fact that measures do not always show to be consistent with theoretical reasoning. The general tendency of exhaust gas mass flow shows to be a linear function of engine speed and power (increasing with both), while same measures do not agree with this behavior.

Last but not least, repeatability is not assured. Different tests showed that measures change from one to the other, thus increasing the incertitude on this kind of data.

One of the main problems related to the Pitot tube is that it is a very sensible instrument. Very specific conditions are needed in order to have reliable measurements, in particular a completely unrestricted flow for at least 50 cm before the Pitot tube. This condition could not be granted because of the vertical encumbrance of positioning the heat exchanger and the gas analyzer pipe one after the other. This is, in fact, the most realistic explanation to the inconsistency of measured values: the presence of the heat exchanger causality on exhaust gas pipe leads to turbulences or, more generally, to an influence on exhaust gas flow, thus causing troubles in its measurement.

Therefore another method for calculating exhaust gas mass flow is needed, in order either to confirm Pitot tube's measures or to have more reliable ones. Clearly this second manner for mass flow measurement must not rely on data coming from the Pitot tube, meaning that no pressure measurement can be used.

The solution to this deal was found in using another information given by the gas analyzer: the A/F ratio, that is the ratio between the mass of air and of fuel in the burning mixture. This result comes from the direct measurement of carbonium-based molecules in exhaust gas, in particular carbon monoxide (CO), carbon dioxide (CO_2) and unburned matter. Starting from these values and from the knowledge of the H/C atomic ratio of the fuel, it is possible to determine the A/F ratio.

This calculation is automatically performed by the analyzer software. Thus, exhaust gas mass flow can be evaluated:

$$\dot{m}_{eg} = \dot{m}_f(1 + A/F) \tag{D.1}$$

where the fuel mass flow is measured by the test bench sensors. This latter value is measured by the test bench, and its incertitude is much smaller than what measured by the gas analyzer. Furthermore, this value has been measured for every test, meaning that a large quantity of data are available in order to have a representative mean value.

This solution presents, apparently, all needed features. It is based on different measurement system, the gas concentration analyzer, which is normally more precise and, what is more important, less susceptible to incorrect mounting.

Even if there is not a perfect correspondence between the two results, they have a similar magnitude. Thus, measured exhaust gas mass flow values were used. However it should be noticed that, in a further development of this project, further work and experimentation is needed in order to confirm these measurements with higher accuracy.

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