Energy integration on a gasoline engine for a vehicular application

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Abstract:

The internal combustion engines are the most applied energy converters in the passenger cars. In order to increase their efficiency, energy integration techniques are used to recover the waste heat. An adapted methodology is required to design the ORC as a waste heat recovery technology and to test the influence of the external temperature on its efficiency. The idea is to clusterize the external temperature profile on typical external temperature multi-periods. The energy system design is then tested on these typical multi-periods.

In this article the methodology is applied on a vehicle with a small gasoline engine, in order to define the energy integrated configuration of the vehicle and to estimate the cost of the additional equipment. The performances indicators of the energy integration technology of the internal combustion engine is done and discussed, according to the multi-periods. The energy recovery potential of a single stage organic Rankine Cycle for a small gasoline engine is assessed for different temperature profiles. The organic Rankine Cycle equipment is simultaneously pre- sized and its cost is estimated. The highest powertrain efficiency improvement due to the waste heat recovery for a small gasoline engine is estimated to 7%.

Keywords:

Energy integration, engine efficiency, Organic Rankine cycle

1. Introduction

The internal combustion engines are the most applied energy converters in the passenger cars. Their main energy service is to insure the vehicle propulsion. Orders of magnitude for efficiencies and energy balances of internal combustion engines are given in [1]. The heat dissipated in the cooling liquids and the exhaust gases of the engine can be used to heat the cabin, or to be converted in electricity or cooling.

The Rankine cycle with organic fluids is seen as most promising candidate for waste heat recovery.

Li studied in [2] several working fluid candidates for various ORC applications based on the heat source temperature domains have been investigated for the thermal efficiency, exergy destruction rate and mass flow rate under different ORC configurations.

Yu et al. performed in [3] a simulation and thermodynamic analysis of a bottoming organic Rankine cycle of diesel engine. The ORC system is built to recover waste heat both from engine exhaust gas and jacket water using R245fa as working fluid. The combined system of diesel engine with bottoming ORC is finally investigated. Results indicate that, approximately 75% and 9.5% of waste heat from exhaust gas and from jacket water respectively can be recovered under the engine conditions ranging from high load to low load. The ORC system performances well under the rated engine condition with expansion power up to 14.5 kW, recovery efficiency up to 9.2% and exergy efficiency up to 21.7%. Combined with bottoming ORC system, thermal efficiency of diesel engine can be improved up to 6.1%.

Quoilin et al. applied in [4] a thermo-economic optimization of an ORC for small scale industrial applications. They stated that the optimum of each performance indicator- net power, thermal and exergetic efficiency of the ORC bellows to different operating points. They also correlate the cost on the ORC components size.

The needs of efficiency improvement of the vehicle energy systems require finding innovative solutions during the design process, integrating all vehicle services – mobility and comfort, and energy requirement on a vehicle system level.
An adapted methodology to design the ORC as a waste heat recovery technology and to test the influence of the external temperature on its efficiency is presented in [5]. The idea is to clusterize the external temperature profile on typical external temperature multi-periods.

This allows bringing sensitivity to the energy integration on real usage temperature profiles. Another contribution of the article is to propose an adapted cost model for the energy integration for small scale installations to 10 kW of net power output and estimation of the fuel consumption reduction due to the waste heat recovery.

2. Methodology

The integration of the vehicle services is illustrated with a methodology coming from the process engineering and is well known as energy integration. The methodology is presented in the next section.

2.1. Energy integration

The energy integration in combination with multi-objective techniques and multi-period perspectives comes from the grid non-autonomous (grid related) energy systems and uses single optimization model with MILP. This method of energy optimization is recently applied to the energy flow optimizations in the vehicular applications in [6], [7], [8]. In this article the energy integration methodology is applied on the energy flows available on the vehicle powertrain and the cabin to integrate the energy demands for mobility and comfort and optimize the total energy consumption.

The method for energy integration on multi-periods is presented in [8].

The energy integration is proposed as an option of the superstructure of the calculation tool presented in Figure 1.

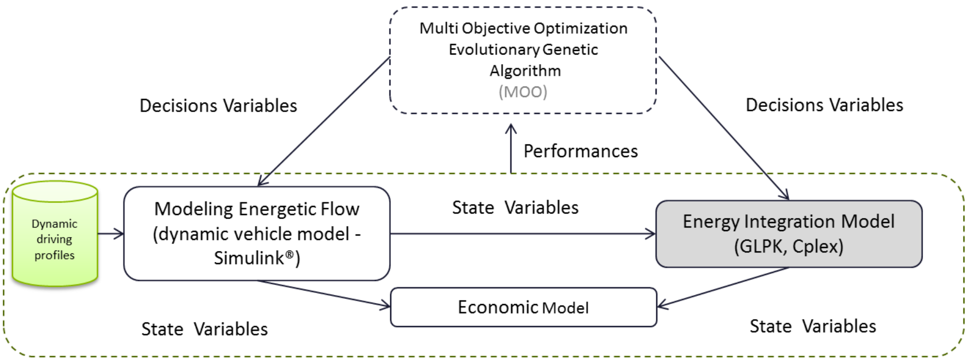

Fig. 1. Architecture of the multi-objective optimization tool

The computational platform combines the energy flow models from the vehicle dynamic model. It is built under Matlab/Simulink® .The energy is exchanged between the unitary processes. In a first time, the energy flow model calculates the energy transformations in the process unit for a specific configuration and operating conditions. The resulting mass and energy flows are used to generate the energy integration model, with optimization of the heat and power production, by minimizing the total exergy and operating cost. The multi-objective optimization is based on an evolutionary genetic algorithm.

The energy integration is a slave optimization part of the Superstructure and in this article the Energy integration model is studied in particularly. The energy integration calculates the best operating strategy of selected equipment by solving a mixed integer linear model (MILP) (Figure 2). The aim is to optimize the energy balance and the heat and power cascade. The input data used in the energy integration includes the values of the master decision variables, the resources, the power dynamic profile and the comfort profile. To do the energy integration optimization multi-periods is needed, reducing the number of variables and the calculation time.

The energy integration is based on the process optimization proposes and is a structured methodology to identify, characterize and quantify the waste heat of a system. The process integration (PI) methodology dates back to the 1970s and the first developments are in the heat integration [9]. The PI is deployed in the oil, chemical and energy industry [9]. Linnhoff in [10] illustrates the pinch method for heat exchangers design and heat integration. Some developments in the PI applied to the chemical engineering research are summarized in [11].

Figure 2 illustrates the applied energy integration method and its steps. The engine operating points are defined from the vehicle simulation model and the drive profile. The energy balance is done for each operating point. After the definition of the unitary streams and the energy integration technology energy integration is done, the performances are evaluated and the ORC is sized.

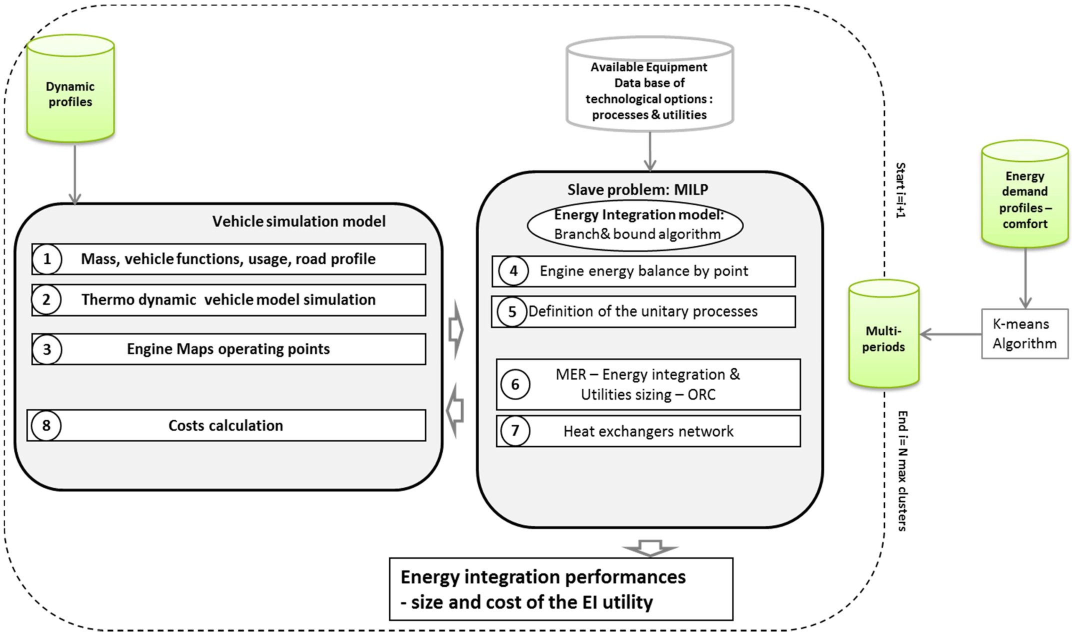

Fig. 2. Energy integration and temperature multi-periods model

In this article the comfort demand profiles are clusterized on stationary typical multi-periods. The problem of selecting typical operating periods based on the energy demand variation has been approached by using an evolutionary algorithm optimization approach to select a typical production scenario for an industrial cluster in [12]. In this article k-means proportional algorithm is used to define the typical temperature periods impacting the comfort demand of the vehicle. The centroid cluster k-means algorithm is described in [13], where the typical power demand profiles for a district heating network are researched.The authors illustrated the algorithm is illustrated in [14]. The obtained results are different temperatures of the typical periods: T_1 = 0.5 °C, T_2= 8°C, T_3= 15°C, T_4=24°C.The extreme level of 0°C and 30°C are selected to represent the winter and the summer case. The novelty of the article is to propose a methodology for integrated mobility and comfort services on vehicular application with equipped with a small gasoline engine – only 1.2 l of a displacement volume.

3. Results on energy integration and temperature multi-periods

The energy recovery technology selected for the energy integration model is the organic Rankine cycle. The results of the energy integration are presented for different external temperatures.

3.1. Energy recovery technology

The organic Rankine cycle (Figure 3) is well known and is used as utility in the energy integration to recover the waste heat from the engine and to produce mechanical power for propulsion or for integrating the comfort demands of the vehicle.

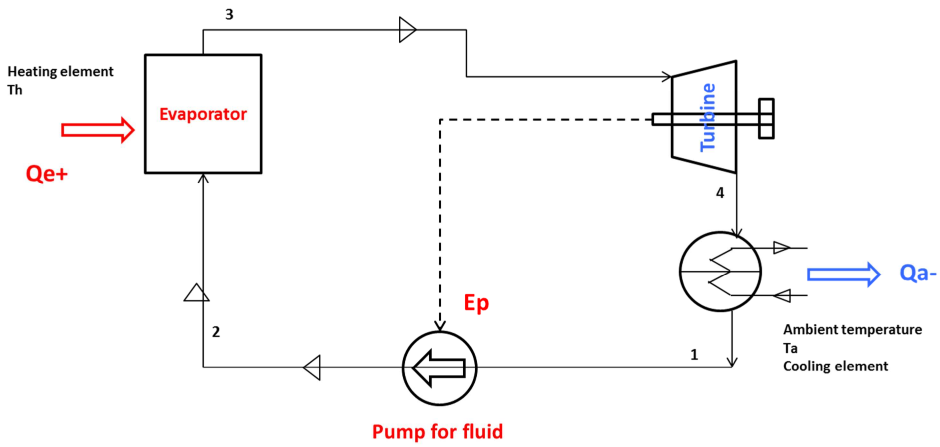

Fig. 3. Organic Rankine Cycle installation

The assumptions made for the implementation of the organic Rankine cycle are:

- The working fluid is ammonia because it is considered as the most efficient fluid
- The efficiency of the turbine is 70%
- The efficiency of the compressor is 80%
- The evaporation temperature is determined by the engine cooling water outlet temperature and engine exhaust gases outlet temperature
- The condensation temperature is determined by the ambient temperature
- The values of $\Delta T_{min}/2$ are 15°C for the gas, 5°C for the liquid, 2°C for the evaporating and 3°C for the condensing.

The ORC cycle is sized with the application of the energy integration methodology exposed in part Energy Integration.

3.2. Engine characteristics

This study considers a small size gasoline engine with 1.2 l of a displacement volume and 60 kW of rated power. The main characteristics are given in Table 1.

Table 1. Internal combustion engine characteristics

	Gasoline
Displacement	1.2 l
Respiration	Natural Aspirated
Rated Power [kW]	60
Number of cylinders	3
Max Torque [Nm]	120
Max Rotation speed [rpm]	6000

3.3. Streams definition

The energy system considers the engine and the vehicle cabin. The energy integration targets the waste heat recovery from the engine and heating or cooling of the vehicle cabin. The integration concerns the mobility and the comfort service of the vehicle. The heat cascade is based on the energy streams.

The different energy streams and their general equations are listed below to construct the heat cascade:

- Cabin heating: from the ambient temperature to the desired temperature (if the cabin has to be heated). The optimal temperature for the comfort is 22°C. Equation 1 gives the heat stream for the heating.

$$\dot{Q}^{+}_{cabin\_heating} = \dot{M}_{air\_cabin} * c_{p_{air}} * (T_{desired} - T_{ext}) \qquad (1)$$

- Cabin cooling: from the ambient temperature to the desired temperature (if the cabin has to be cooled down). Equation 2 gives the heat stream for the cooling.

$$\dot{Q}^{+}_{cabin\_heating} = \dot{M}_{air\_cabin} * c_{p_{air}} * (T_{ext} - T_{desired}) \qquad (2)$$

Equations 3 to 5 give the heat streams for the engine.

- Engine water cooling

$$\dot{Q}^{+}_{water} = \dot{M}_{water} * c_{p_{water}} * (T_{water\_outlet} - T_{watar\_inlet}) \qquad (3)$$

- Engine oil cooling

$$\dot{Q}^{+}{}_{oil} = \dot{M}_{oil} * c_{p_{oil}} * (T_{oil,outlet} - T_{oil,inlet}) \qquad (4)$$

- Exhaust cooling:

$$\dot{Q}^{+}_{exhaust} = \dot{M}_{exhaust} * c_{p_{exhaust}} * (T_{exhaust,oulet} - T_{ext}) \qquad (5)$$

Equations 6 to 10 give the heat streams for the energy integration technology, which is the ORC.

- ORC preheating:

$$Q^{+}_{ORC\_preheating} = \dot{M}_{fluid} * (h_3^{'} - h_2) \qquad (6)$$

- ORC evaporation:

$$Q^{+}_{ORC\_evaporation} = \dot{M}_{fluid} * (h_3^{''} - h_3^{'}) \qquad (7)$$

- ORC superheating:

$$Q^{+}_{ORC\_\sup erheating} = \dot{M}_{fluid} (h_3 - h_3^{''}) \qquad (8)$$

- ORC cooling:

$$Q^{+}_{ORC\_cooling} = \dot{M}_{fluid} (h_4^{''} - h_4) \qquad (9)$$

- ORC condensation:

$$\dot{Q}^{+}_{ORC\_condesnation} = \dot{M}_{fluid} (h_1 - h_4^{''}) \qquad (10)$$

The streams and the utilities are defined in the Energy Integration module of the computational tool (Figure 1). The energy integration is a slave optimization problem, which uses a mixed integer nonlinear programming to be solved. The energy balance of the engine and the cabin is established and the heat cascade is defined. The internal combustion engine is a system with excess of heat and the organic Rankine Cycle is selected as utility to recover the heat and convert it into mechanical power.

The composites curves with the energy integration of organic Rankine cycles for different engine loads are given in Figures 4 and 5:

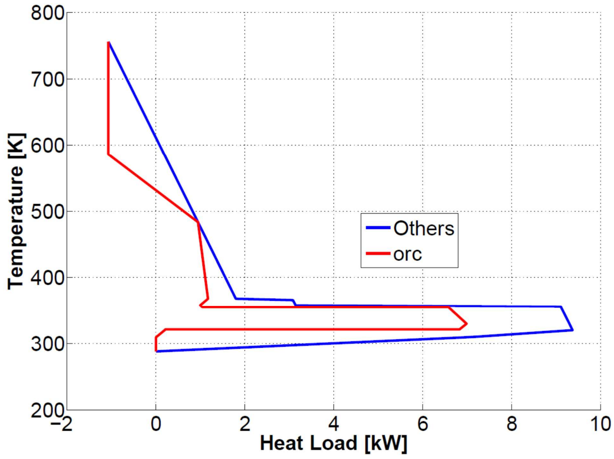

Fig. 4. Energy integration on 2000 rpm 2 bar operating point

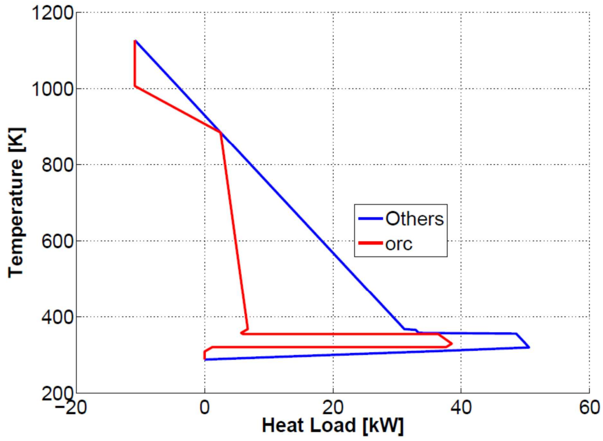

Fig. 5. Energy integration on 5000 rpm 8 bar operating point

For the engine low speed and low load point – 2000 rpm 2 bar of BMEP (Figure 4) , the engine water circuit presents the most important potential of heat recovery – 5 kW and the exhaust gases are the second important heat source with a potential of 2.5 kW heat load. At high load and high rotation speed (Figure 5), the main heat source is the exhaust gases, which reach an important temperature of 1100 K and their heat recovery potential is 30 kW. The water circuit is the second source for heat recovery, with 12 kW of heat load. One can conclude that the heat recovery potential and the order of importance of the two main heat sources depend on the operating point of the engine.

3.4. Thermodynamic and economic evaluation of the organic Rankine cycle

The energy integration approach is extended on all operating points from the engine operating field. The performances indicators of the energy integration utility – the ORC are presented on the Figures 6, 7 and 8. The performances indicators are illustrated for the winter case with external temperature of 0 °C. This allows illustrating the dependence of the performances indicators of the energy integration of the ambient temperature. Matlab/Simulink® is used for the models.

3.4.1 Performance indicators

The net power in kW is the absolute value of the mechanical power, delivered by the organic Rankine cycle.

$$\dot{P}_{net} = \left|\dot{P}_{\exp ansion}\right| - \left|\dot{P}_{pumping}\right| \qquad (11)$$

Energetic efficiency: The energetic efficiency is here defined as the ratio of the net mechanical power output over the thermal energy available in the main hot sources – the engine exhaust gases and the water circuit), with a reference of the ambient temperature.

$$\varepsilon = \frac{\dot{P}_{net}}{\dot{Q}_{exhaut\_gas} + \dot{Q}_{cooling\_water}} \qquad (12)$$

The available heat is the amount of energy present in the streams and depends on the engine operating point and the ambient temperature.

Exergetic efficiency: The exergetic efficiency is the ratio between the net power-work delivered by the ORC and the exergy- heat received by the ORC system.

$$\eta = \frac{\dot{P}_{net}}{\dot{E}_{exhaut\_gas} + \dot{E}_{cooling\_water}} \qquad (13)$$

$$\text{with } \dot{E}_{exhaust\_gas} = \int_Q (1 - \frac{T_a}{T_{exhaust\_gas}}) \delta\dot{Q}_{exhaust\_gas} \quad (14)$$

$$\text{and} \qquad \dot{E}_{cooling\_water} = \int_Q (1 - \frac{T_a}{T_{cooling\_water}}) \delta\dot{Q}_{cooling\_water} \quad (15)$$

If one assumes a constant specific heat of the fluid constituting the heat source between its initial and final temperature (T_{in} and T_{out} respectively), the initial form can be replaced by the expression below, using the logarithmic temperature difference.

$$E_x = (1 - \frac{T_a}{T_{LTD}}) * Q \text{, with } T_{LTD} = \frac{T_{in} - T_{out}}{\ln \frac{T_{in}}{T_{out}}} \quad (16)$$

$$\text{So } \dot{E}_{exhaust\_gas} = \int_Q (1 - \frac{T_a}{T_{exhaust\_gas}}) \delta\dot{Q} = \dot{Q}_{exhaust\_gas} * (1 - \frac{T_a}{T_{exhaust\_gas}}) \quad \text{(17), and}$$

$$\dot{E}_{cooling\_water} = \int_Q (1 - \frac{T_a}{T_{cooling\_water}}) \delta\dot{Q}_{cooling\_water} \qquad (18)$$

After solving (18), the expression for the exergy of the cooling water is given in (19).

$$\dot{E}_{cooling\_water} = \dot{Q}_{cooling\_water} * (1 - \frac{T_a}{\frac{T_{cooling\_water\_in} - T_{cooling\_water\_out}}{\ln \frac{T_{cooling\_water\_in}}{T_{cooling\_water\_out}}}}) \qquad (19)$$

In the present case of an ORC in waste heat recovery applications, the thermodynamic optimization aims at maximizing the net power output. The other thermodynamic indicators – the energetic efficiency and the exergetic efficiency are additionally used to characterize the thermodynamic behavior of the system. The cycle thermal efficiency – the energetic efficiency is an indicative parameter of the quality of the heat converted into power (12).

The different evaporation pressure varies the heat transferred to the power cycle. An increase in the evaporation pressure reduces the amount of heat transferred to the power cycle and the amount of heat rejected at the condenser.

Figures 6, 7, 8 illustrate the ORC performances indicators on the global engine field for external temperature of 0 °C.

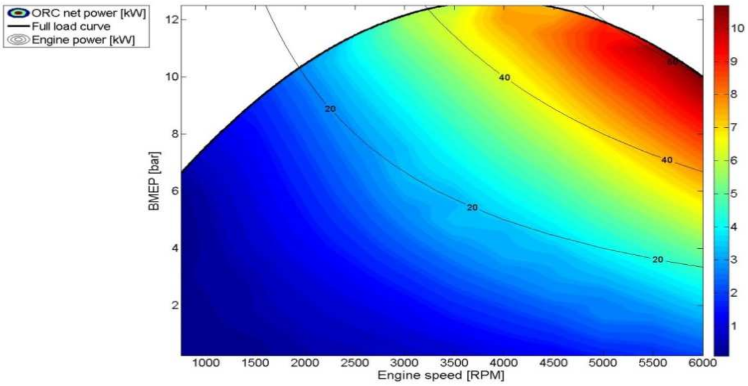

Fig. 6 : ORC net power on the gasoline engine operating field, T ext = 0°C

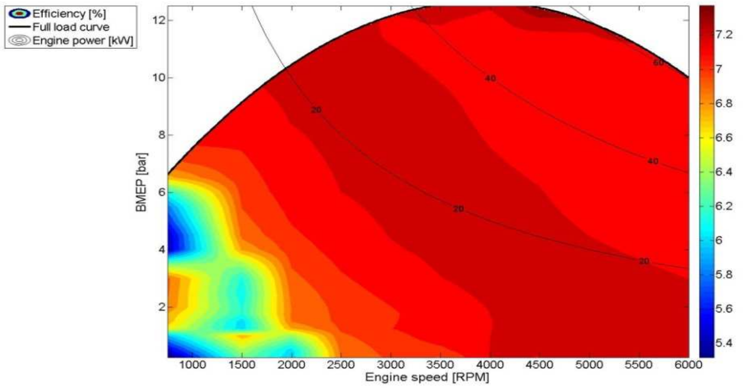

Fig. 7 : ORC energetic efficiency on the gasoline engine operating field, T ext = 0°C

An optimum is obtained for the three parameters but for different pressure values. Figures 6, 7 and 8 represent the performances indicators on the gasoline engine. It is visible that the maximum power zone is different of the maximum energetic efficiency zone (Fig. 7) and the exergetic efficiency zone (Fig. 8). The exergy map of the engine shows the zones with maximum heat recovery potential – the high speeds and the high loads zone, where the temperature of the exhaust gases in important. Thus the gasoline exergy efficiency is important – superior to 30%. The high temperature water in the engine cooling circuit is an additional source, visible on the exergy maps, in the zone of low rotation speed and low loads. In case that the heat recovery is for free, because the heat in the exhaust gases and the water circuit is lost. In these two zones the ORC net power is maximized.

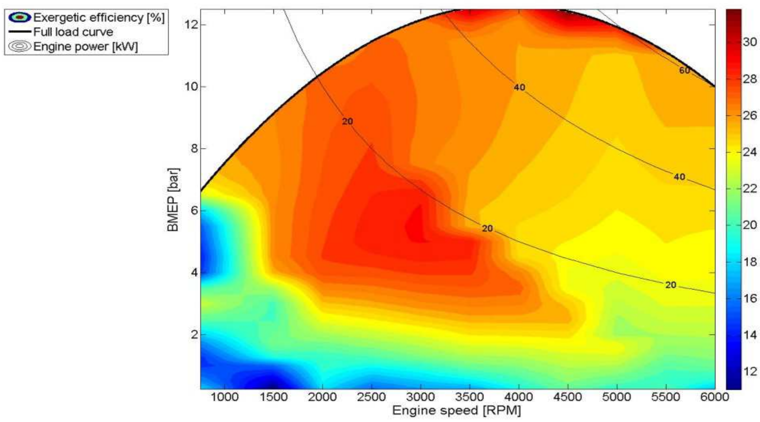

Fig. 8 : ORC exergetic efficiency on the gasoline engine operating field, T ext = 0 °C

3.4.2 Economic model of the energy integration technology

The economic model for the components of the energy integration utility is based on estimations for the cost of the utility components: pump, turbine, heat exchangers (condenser and evaporator).

The equations for every component are coming from the cost analysis of parts already known and with a large scale application in the automotive industry. The reference elements for the pump and the heat exchangers are the components from the air cooling installation in the vehicle. For the turbine cost the engine turbo machines are benchmarked. The cost model is valuable for utilities with small size and installations producing between 5 and 50 kW of net power.

The total utility cost is defined by the following equation:

$$Cost_{utility} = Cost_{condenser} + Cost_{evaporator} + Cost_{compressor} + Cost_{turbine} Cost_{pipes} + Cost_{fluide} \quad \text{in [€] (20)}$$

Pump: $C_{pump} = 165 + 83 * P_{pump}$ in Euros with P_{pump} in [kW] (21)

For a gasoline engine at $T_{ext} = 0°C$ the cost is 178.53 €

Turbine: $C_{turbine} = 15 * P_{turbine} - 30$, in Euros with $P_{turbine}$ in kW (22)

For a gasoline engine at $T_{ext} = 0°C$ the cost is 134.14 €

The cost equation of the turbine comes from the linear correlation between the power past through the turbo shaft of two turbo, mounted on serial gasoline engines. The power of the ORC is in the same range that the power passed through the turbo shaft.

Condenser:

The cost equation for the condenser and the evaporator are obtained from the linear correlation between the total exchange area and the cost for serial cross flow condensers and evaporators, massively used in the vehicles. The technical data are available in the detailed specification of the condenser of serial vehicles.

The cost equation for the heat exchanger is:

$$C_{HEN} = (0.0401 * A_{ex} * 0.5 * 100 + 4.096) + (0.1449 * A_{ex} * 0.5 * 100 + 18.84) \quad (23)$$

The factor 100 multiplies A_{ex} to convert the total exchange area from m² to dm². The factor 0.5 is because of the assumption that the evaporator and the condenser have the same area.

For a gasoline engine at $T_{ext} = 0°C$ the total cost is 49.10 €

The cost of the pipes and the fluid is considered constant and is presented in (24) and (25).

$$Cost_{pipe} = 26 \,\text{[in Euros]} \quad (24)$$

$$Cost_{fluid} = 10 \,\text{[in Euros]} \quad (25)$$

The total cost for the gasoline ORC with maximal power of 10 kW is 386.87 €. This cost represents the technical cost and is based on estimations in the domain of small scale installations.

3.4.3 ORC net power, energetic and exergetic efficiency maps for the summer case

The evolution of the indicators for the summer case is represented in the figures 9, 10 and 11.

Summer case, T=30°C

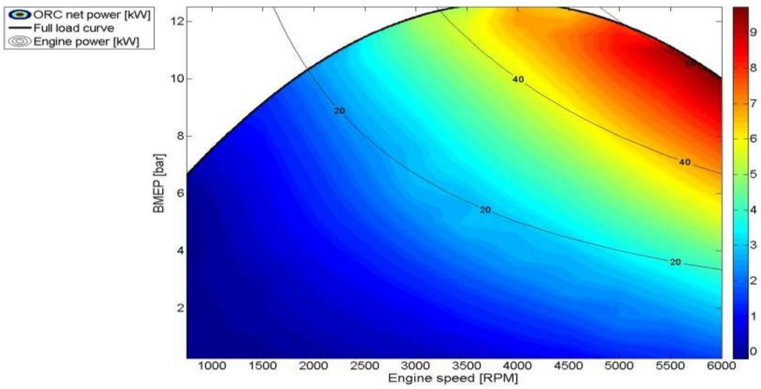

Fig.9. ORC net power on the gasoline engine operating field, T ext = 30°C

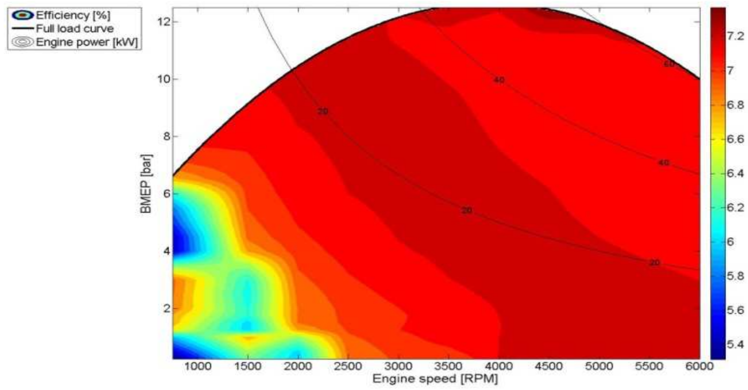

Fig. 10. ORC energetic efficiency on the gasoline engine operating field, T ext = 30°C

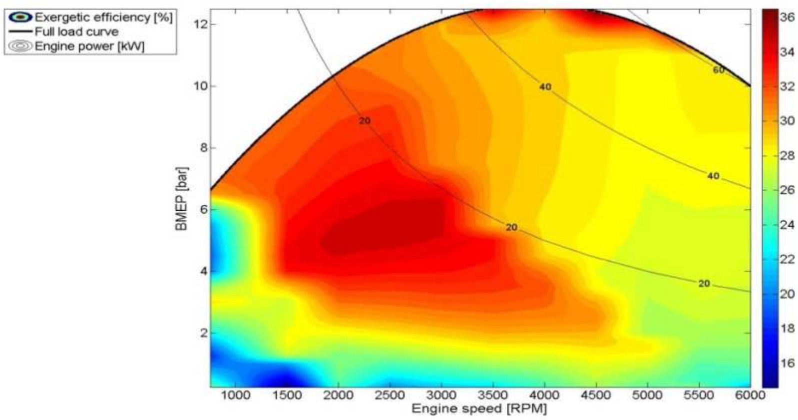

Fig. 11. ORC exergetic efficiency on the gasoline engine operating field, T ext = 30°C

The ORC net power output decreases with the ambient temperature increase. The heat pump is present in the energy integration when $T_{ext} > T_{desired}$ and the power takes into account the power needed for the heat pump compressor and the cooling of the cabin. For the summer case with $T_{ext} = 30°C$ one can notice that the zone under 20 kW of the engine is representing 1 kW of energy recovery. From the energetic efficiency map one can notice that the energetic efficiency map stays constant on the engine field. The maximum efficiency is 8 % and according to the cycle design is located on the power band between 15 kW and 30 kW. The cycle efficiency is not considering the ambient temperature (12).

From the exergetic efficiency map on can notice that the exergetic efficiency increases with the increasing external temperature. The maximum is in evolution from 30% when $T_{ext} = 0°C$ to 36 when $T_{ext} = 30°C$. The heat is multiplied by the Carnot Factor, which decreases with the T_{ext} increase. The exergy of the heat source than decreases and the exergetic efficiency increases ((13)-(19)).

3.5. Heat recovery simulation results on a vehicle

In the previous section net power and efficiency maps were created for an organic Rankine Cycle integrated on a 1.2 liter gasoline engine. These maps are now put to use as they are fitted into the quasi-static vehicle simulation model. Simulations are run for the normalized new European driving cycle (NEDC) on a pure thermal powertrain type. The vehicle used for this simulation is considered with a mass of 1100 kg and is equipped with a 1.2 l gasoline engine and a manual gear box.
Table 3 summarizes the results.

Table 3: Results of the ORC integration on a thermal powertrain: $P_{MAX,ORC}$ maximum net power reached by the ORC; E: energy output during the cycle; η_{pt}: powertrain efficiency in traction; $\eta_{pt,ORC}$: powertrain efficiency taking into account the ORC net power; Delta η efficiency improvement.

Cycle	T_{ext} [°C]	$P_{MAX,ORC}$ [kW]	E [kWh]	η_{pt} [%]	$\eta_{pt,ORC}$ [%]	Delta η
NEDC	0	7.22	0.28	18.50	26	7.50
NEDC	30	4.20	0.14	18.50	21.30	2.80

The maximal benefit of the heat recovery technology integration of 7.5 % on the powertrain efficiency is obtained for the lowest external temperature of 0°C. The powertrain efficiency improvement is reduced with the increasing external temperature and for T_{ext} = 30°C. Thus the improvement is 2.8%.

4. Conclusion

This article presents a methodology for the design the waste heat recovery technology on a vehicle energy integrated system. The performances indicators of the waste heat recovery system are introduced. The external temperature variation is represented as a typical multi-periods characteristic on a simplified clustering way. The influence of the typical external temperature on the waste heat recovery performances indicators is studied. An economic model for the technical cost estimation of the ORC technology is presented. This model is done for small scale waste heat recovery systems and is based on costs of components already known in the automotive industry. The objective of its usage is simply to fixe some orders of magnitudes in the domain of the small scale waste heat recovery installations. The integration of the engine and the comfort system of heating and cooling is estimated for different temperature clusters. It allows estimating the effectiveness of the integration for different temperatures. Tests of the integrated installation could be performed on a test bench.
The conclusion on the main performance indicator – the delivered net power – is that the ORC net power output decreases with the ambient temperature increase. Thus the fuel saving contribution of the ORC is sensitive on the ambient temperature. The maximal fuel consumption benefit of the waste heat recovery for a small gasoline engine is 7% for the winter case. The benefit is reduced to around 3% for the summer case.

Nomenclature

$\dot{E}_P$ pump power, kW

$\dot{E}_T$ turbine power, kW

$\dot{Q}_E^+$ heat load evaporation, kW

$\dot{Q}_A^-$ heat load condensation, kW

$\dot{Q}_X$ heat load for x element, kW

$\dot{M}_x$ flow of the x fluid, kg/s

$\dot{P}_{net}$ net power output, kW

$\dot{P}_{\exp ansion}$ expansion power, kW

$\dot{P}_{pump}$ pumping power, kW

h_x Enthalpy of x transformation, J/kg

MER minimum energy requirement, -

T_a ambient temperature, °C

T_{LTD} Logarithmic temperature difference, -

C_x cost of x element, €

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