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Design, Modeling and Control of the Heavy Truck Engine Waste Heat Recovery Unit

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Abstract: This paper presents a feasibility study of an organic Rankine Cycle (ORC) technology waste heat recovery system for heavy duty trucks. The new features are: I the plant configuration proposed and ii) the feasibility study that covers the entire preliminary plant workflow, including the optimization of the thermodynamic cycle and the preliminary size of the components, and the dynamic modeling and development of a PI control system. The design ORC turbo generator is a fluid with a maximum rating of approximately 5 kW, corresponding to the truck cruise speed and the diesel power generation of 85 km h−1 and 100 kW, respectively. This turbo generator uses hexamethyl isyl disiloxane (MM) as a work-fluid. In terms of dynamic performance, an advanced control system is required due to a higher response time of the ORC unit compared to that of the Diesel engine. In particular, the simulation of a PI-based control system shows that thermal deterioration of the working fluid cannot be avoided if the engine is continuously running at high power levels. The case study reveals the importance of the dynamic performance and control design study already in the early design phase for automotive ORC applications.

Keywords: Centralized control, Dynamic modelling, ORC, Waste Heat Recovery.

INTRODUCTION

Following nearly 30 years, there has been a renewed interest in heating systems of diesel engine (DE) from original equipment manufacturers (OEMs). This was demonstrated by the many recent research activities. The reasons for this are I the growing regulations on the emissions of trucks and ii) the limited efficiency improvements that are now possible thanks to the changing modern diesel systems. Nearly half the fuel consumption is to remain at heat in future DE, which will have an expected fuel efficiency of some 50 per cent, notably in the engine cooling jacket (15% of fuel thermal input), NOx exhaust gas recirculation (10% in) and the turbocharger exhaust gas (25%). The Organic Rankine Cycle (ORC) turbine generators, thanks to their relatively high conversion efficiency, simplification and the ability to tailor the working fluid to comply with limited volume on board trucks and the safe and environmental requirements, are arguably the best choice for recuperation of thermal waste engine energy[1]–[3].

A single cycle system consists of the ORC configuration, generally recommended for the application which absorbs the thermal power from engine exhaust gasses downstream of the turbocharger and the exhaust after-treatment unit (ATU). But this approach is, despite its simplicity, less appealing due to the substantial improvements in DE technology over the past decade. The temperature at the ATU outlet in cruise conditions is now around 265 µl / C and is expected to decrease further in the future. This leads to an over-low power output of the ORC turbocharger, which is probably only achieved if the WHR unit is able to reduce fuel consumption by a minimum of 5%. This objective can be achieved by converting other heat waste streams into electricity, The ORC turbo generator, in parallel to recover thermal energy from exhaust gasses and the EGR, has a recuperative cycle arrangement and two evaporators Given the factors relating to the feasibility of turbo expander, the authors chose octamethylcyclotetrasiloxane (D4), a relatively complex cyclic siloxane to be the functioning fluid. Accordingly, such a WHR can produce 9.3 kW at cruise conditions that is 6.2 percent of the engine power, according to its calculations. It implies an equal reduction in fuel consumption, for a first approximation. The fact that a recovery and a molecular complex fluid can be taken into account makes the theses of the exhausts in the upstream of the ATU at significantly higher temperatures, i.e. above 300 / C instead of 265 / C, an aspect that was not explored. Indeed, the right design of the recuperator could limit the temperature drop of the exhausts throughout the ORC evaporator, preserving the performance of the ATU's selective catalytic decrease device (SCR). The advantage could be that the turbo generator's ORC power output is increased, as the improved conversion cycle efficiency achieved through recovery and higher thermal source temperature can more than offset the lower thermal energy recovered from the exhaust gas. The effect will be increasingly apparent in future, as DE output increases steadily and the exhaust gas temperature decreases[4], [5].

Nevertheless, the implementation of such a principle may entail complex monitoring problems. The key aspects at first glance are: I the parallel operation of two evaporators working at a relatively different temperature level; ii) the maintenance in the off-design conditions of exhaust gases above 200/c and iii) avoiding fluid thermal decomposition throughout the entire range of operations of the DE. Since the two last restrictions are tightly interconnected and very likely contradictory, it is possible that their achievement requires not only the design of the advanced control units, but also a change in process mechanics: for example, the retriever or other heat exchanger size to be adjusted, so as to achieve a more beneficial dynamic system. There are currently. In fact, the dynamics of an ORC unit with 2 parallel evaporators have not yet been studied to the knowledge of the authors.

The purpose of this study is therefore twofold: I to evaluate a WHR unit's potential for a modern DE, which key feature of which is the recovery of thermal energy from gas exhaust upstream of the ATU; ii) to study system dynamics and control. This latter step is considered an important step in properly assessing the concept's efficiency and its viability. The document is organized in the following way: Sec. 2 discusses definition of the thermodynamic cycle of the WHR unit, the preliminary design of the cycle components; Sec. 3 describes the development of a dynamic model of all truck powertrain through the integration of the WHR unit into DE; in the last section, the final comments will be given.

WASTE HEAT RECOVERY UNIT DESIGN

1.1. ORC configuration and design specifications:

As already mentioned, for a single pressurized cycle with two in parallel evaporators (see figure 1), both the exhaust gasses and the DE EGR system recover thermal energy from the selected configuration for the WHR. The evaporator heated by the exhaust gasses is located upstream of the ATU and internal heat recovery is not used in the operating heat recovery circuit of the EGR network. Although the previous amendment has been already discussed, the latter stems from the fact that recycled gasses must be cooled down to a lower temperature to increase the average air and waste gas mixture density in the engine inlet collector. This increases the fresh load and thus the output of DE in the cylinders. Further improvements to the power output of WHR units will likely only be possible if the plant is complicated considerably: for example, the use of a two-tier cycle of the pressure, or other potential thermal sources, such as the air from the turbo-charge or cooling system of the engine. The result is a significant improvement. The recoverable thermal energy from thermal energy sources, however, is available at relatively low temperature ranges of 100 140 centimeters. In addition, the application's small power capacity prevents the use of two different

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expanders or the installation of a dual-access turbine. Therefore, the increase in power output that these solutions enable is rather small, particularly with regard to the design challenges implied by their implementation. They were not investigated for this reason[6], [7].

Due to its greater compactness, reliability and ability to handle greater stress levels than volumetric machines, a high-speed turbine is probably the best option for the expander. The last function is especially important in order to achieve the WHR unit's high conversion efficiency. The assumed turbine configuration is an axial unit of two phases. This flow architecture is preferred, as the expander down speed is designed to achieve high reliability, which would seem to perform better than the radial-inflow architecture.

The work fluids considered for the ORC unit belong to the siloxane family. These fluids show a high molecular complexity, allowing efficient turbines to be built even for power outputs of just a few kW according to the specific application. In addition, it is produced in large volumes and has a limited flammability, low toxicity and excellent thermal stability, which is demonstrated by its use both as heat transfer mediums and as working fluids in ORC high-temperature systems. Amidst the siloxans, the best candidate is hexamethyldisiloxane (MM), as the lowest point of boiling and critical temperature are present. It therefore prevents high condenser vacuum levels and makes it possible for the temperature profiles of the evaporator to fit correctly. History. 1 summarizes the design and the system parameters supposed to perform the WHR unit thermodynamic analysis and preliminary design. The working fluid condensing pressure, depending on the temperature of the cooling water supplied by the camera radiator, is a key parameter for the performance of a system.

In cruise conditions, the engine's mechanical power output is approximately one third its full capacity, i.e. the engine design point of the ORC turbo expander. Therefore, the radiator's cooling power is not fully excavated. This additional capacity can be used to cool down the cooling water to 70 ° C rather than 8590 ° C, as needed for the motor cooling system to work. Before it is sent to the cooling jacket the cooling water has to be pre-heated e.g. with the ORC condenser or by mixing with warmer waters, which pass the radiation tor. Such a solution allows the mechanical output of the ORC unit to be significantly improved without using a dedicated cooling belt for the same system. At cruise conditions, the condensing temperature is assumed to be 85 /C at a first approximation[8].

For example when the truck speed and engine power output is 85, h−1 km or 100 kW, the DE data for modeling the thermal sources of the unit have been provided by a leading truck manufacturer for Euro 6 DE at cruise conditions. The values of the parameters used to estimate ORC component output are instead defined in line with common practice for commercial ORC system.

1.2. Preliminary Design Methodology:

The WHR unit is preliminarily designed through an innovative and integrated design method which allows for the parameters of the thermodynamic cycle and the main geometric features of the expander to be simultaneously optimized. I the possibilities of discarding cycle configurations which can lead to an unachievable solution to a turbine design; (ii) identifying the optimum interaction between the cycle and expanding efficiency; And iii) a more consistent evaluation of ORC unit efficiency, as the designer does not prioritize turbine output and holds the whole thermodynamic cycle design space continuously. The design and performance of the turbo-extensive system therefore depend heavily on the choice of the input thermodynamic conditions and the expansion ratio for small-scale ORC applications. Fig 1 ORC component layout[9].

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$\dot{m}_{EXH}$	kg s⁻¹	0.131
$\dot{m}_{EGR}$	kg s⁻¹	0.066
T_{EGR}	°C	400
$T_{EXH,1}$	°C	314
$T_{EXH,2}$	°C	265
T_{cond}	°C	85
$T_{min,exh}$	°C	200
$\eta_{is,pump}$	-	0.65
$\Delta T_{sh,min}$	°C	5
$\Delta P/P$	–	0.01

Table 1. Model assumptions

Source	$\dot{W}_{mec}$ kW	$\dot{Q}_{EXH}$ kW	$\dot{Q}_{EGR}$ kW	p_{eva} bar	ΔT_{sh} °C	$\eta_{is,turb}$ -
EGR+EXH[1]	4.8	15.6	20.7	12.6	21.5	0.715
EGR+EXH[2]	4.0	21.2	20.7	6.4	6.5	0.747
EXH[2]	2.0	22.3	-	6.4	8.6	0.723

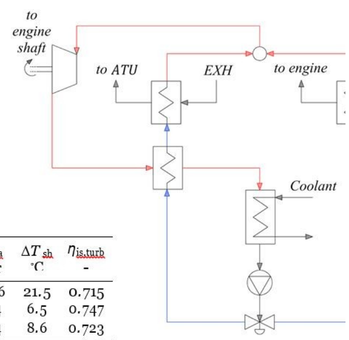

Figure 1: ORC Component Layout

The WHR preliminary design consists, from a mathematical point of view, of solving a restricted optimization problem whose objective function is to produce net power of the device. The design variables chosen are evaporation pressure, surcharge of the operating fluid at the evaporator outlet, key geometrical parameters in the cascading turbine (e.g. inlet blade height, blade outlet angles), pressure ratio and reaction level in every turbine phase. The following are indicated: Imposition of constraints on key geometrical parameters, such as minimum height and maximum flaring angle of the turbine blades, ensures the feasibility of the design solution. In order to avoid convergence into local solutions, an optimizer based on a genetic algorithm is used because of the large number of design variables.

The code was implemented in a programming environment for general purposes. In conjunction with an internal median model for multi-stage ORC turbines, it is paired with an external code library for the measurement of thermo-physical properties of the operating liquid. For an overview of this tool and its validation are referred to and for further details on an integrated design methodology adopted. The reader is referred to. As far as the heat exchangers are concerned, their optimum design is done by a well-known commercial software for process optimization. Plate heat exchangers (PHEs) have been chosen because of their compactness and their possible low cost in case of high volume production. Note that installation problems, such as limited space on board the truck, may greatly affect the performance and design of the system's heat transfer equipment.

For the potential inquiry, an evaluation is therefore left of their impact on the performance of the ORC turbo generator. Finally, to limit the increase in turbo-charge backpressure, it is necessary to say that the maximum pressure drop in the exhaust gases across the evaporator is not exceeding 5 kPa[10].

RESULTS

In Tab are reported the results of the design procedure. 2. The WHR can supply the Diesel engine with an additional power supplies of 4.8 kW in the two evaporators with a power recovery of 36.3 kW. The evaporation pressure and a degree of superheating of around 12.6 bar and 21.5 µl respectively achieve this performance. Such values provide the best balance between efficiency of the thermodynamic cycle, recoverable thermal energy and turbine output. In effect the optimal solution would have had the minimum degree of superheating necessary to prevent the presence of liquid goutlets at the expander inlet had fixed turbine effectiveness taken into consideration in the thermodynamic cycle models. In addition, tab. In addition. 2 reports an optimal design solution for the collection of thermal power from the sole exhaust gases by the ORC configuration and that of a WHR unit that also recovers thermal energy from EGR but also the ATU downstream. It is clear that the former is performing too low for the economical heat recovery of long-distance truck engines.

In contrast, despite the same level of complexity and the increased level of thermal energy recovered from exhaust gas stream, the latter generates a power output approximately 20 percent less than the proposed configuration. In Tab, too, notice. 2 isentropical efficiency deterioration of the turbine as the stress ratio increases.

Ultimately, the HEXs have a weight of around 150 kg and are within acceptable limits. That ensures that the truck weight is 0.4 percent higher.

Dynamic Modeling:

A dynamic model of the entire truck powertrain has been developed using Modelica, the open source, equation-based language used for the modeling of differential-algebraic equations (DAEs) systems, in order to study its dynamic performance and its control system. The ORC models were taken from the ORC library, which in a prior study was validated by two of the current authors against experimental data from a small-scale ORC turbo generator for waste heat recovery. Instead, the DE system models were specifically developed for this work. The Modelica reference charts for DE and ORC, respectively, are shown in Figures 2 and 3. The PHEs for the WHR system are modeled by a pure 1D counterturned approach to approximating their topology and by simplifying the transfer of heat and the drop of pressure. These empirical equations were targeted according to the forecasts of the static state models during the preliminary design phase so that the performance of the components in off-design conditions was replicated with reasonable accuracy.

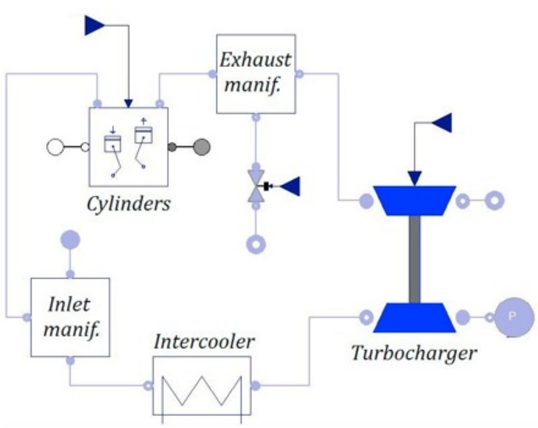

Figure 2: Object Diagram of the Engine

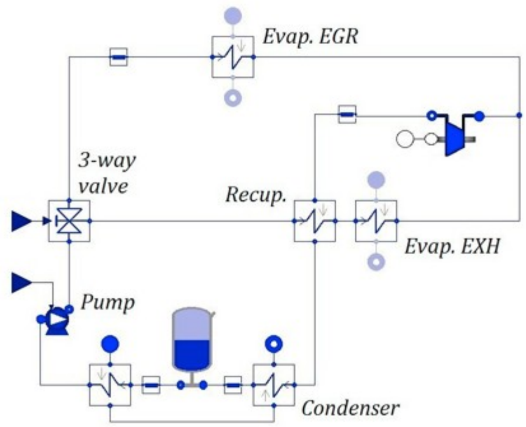

Figure 3: Object Diagram of Bottoming Cycle

The ORC pump and turbine are represented by quasi-static models that use recalibrated maps to predict the efficiency and flow of both machines. Tables are, in particular, composed of values to report reduced flow or isentropical efficiency in the working range of the considered variable as a function of the pressure ratio and revolution speed. The turbine maps were calibrated according to the methodology proposed in, while the pump maps were extracted by scaling and fitting the existing volumetric pump data, designed specifically for operating conditions similar to the applications under discussion. A gearbox connects the turbine shaft with that of the engine.

A lumped parameter model is based on the DE dynamic model. The complex features, the cooling loop, the turbocharger and the EGR mechanism are taken into account. On the other hand, the dynamics of the burning process are neglected as it is of magnitude order quicker than the WHR unit and the total burning is considered. Before being combined with the dynamic model of the ORC Unit and other components, this engine model needs to be extended. Notably the changes required are: I introducing a sub model for the replication of the motor control unit operation and ii) determining the turbocharger outlet exhaust gas temperature. To this end, the static control map was calibrated on the basis of the

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measurements made available from the truck producer for each controlled variable-i.e. the fuel consumption, the opening of the EGR valve, and the actuator position regulating the turbo-charge nozzle van.

These experimental data include test bases at various motor loads and stability conditions. In accordance with the motor's speed and torque, the maps developed define the optimally controlled variables. With respect to the forecast of the exhaust gas, the expansion cycle in the turbine is calculated by a polytropical expansion, the index of which n has been tuned according to experimental information. This enables the thermal losses in the turbocharger to be easily taken into account.

1. Waste Heat Recovery Control:

The two principal requirements for the WHR control system to ensure the proper functioning of the engine are I the prevention of too low an exhaust gas temperature in the SCR system and ii) the prevention of the organic fluid thermal decomposition. So, exhaust gas temperature Tscras, the evaporator outlet and the degree of superheating at the turbine inlet seem to be the most relevant regulated variables. The controller should simultaneously:

1. Maximize the mechanical output of the ORC under all operating conditions;

2. Guarantee a minimum overheating degree, to prevent the existence of liquid turbine inlet droplets;

3. To avoid raising the ICE output, cool down EGR gas flow as far as possible;

4. Thus prevent jeopardizing the mechanical integrity of PHE the evaporator pressure remains subscripted.

Since the control variables of the system are only two, namely the pump rotating speed and the three-way valve at the evaporator inlet, these goals can be pursued only simultaneously through the selection, according to specific conditions of operation, of appropriate sets of controlled variables. For this purpose, two optimal set-up maps covering the entire range of the WHR unit shall be used by the controller. Their calibration was achieved by optimizing ORC output in off-design conditions according to the specifications of the control system.

1.1. Control Architecture:

The design of the control system depends on reciprocal relations between manipulated and controlled variables and the criteria for control efficiency. The measurement of the device Relative Gain Array (RGAs) matrix, which feature of the transfer process, gains the correct link between manipulated and controlled variables (sees Eq.2). These interactions can be evaluated by measuring the RGAs. If the RGA is similar to the ID matrix, a decentralized control architecture with good performance may be implemented, otherwise it is important to have a centralized architecture. For linearized models around the nominal operating point and in a limited number of off design conditions this study has been conducted. The mean value of Ś11 is about 2.5; thus, there's a strong connection between the system and a central control architecture.

Statically decoupled systems, shown in the figure, are the simplest centralized control architecture. 4. The K (2x2 constant matrix) disconnected is so designed that it is an identity matrix of the cascaded disconnect connection between the decoupler and the process K G(s) which can be controlled by independently input PI(D) controllers with a single output. With the operating point, the transfer functions of the linearized process change considerably, so a fixed parameter controller does not work for the entire operating field. The two evaporators ' mass flow rates, which are normalized to their optimum values during defining optimization is useful for using the two virtual control variables as a way to reduce the system's non-linearity.

Eventually, the study of Right-Half-Plane (RHP) multivariable transmission nulls of G(s) transmission function shows that the cycle is non-minimum. This means that the controllers are constrained to the maximum bandwidth of~− 0.01 rad s−1. It is clear that this constraint is inherent to complex systems, irrespective of the complexity of the control system; it would be necessary to redefine the process itself to loosen this restriction. For future research, this is an interesting subject, but not part of this work.

CONCLUSION

A new ORC configuration was examined, along with a simple control strategy, for heated waste heat recovery from heavy-duty truck engine. The study findings suggest the following points:

1. In cruise conditions, the power output of the proposed ORC configuration is 4.8 kW, corresponding to nearly 5 percent of the mechanical power supported by the DE. The latter should decrease by the same percentage if a linear correlation is formed between engine power and fuel consumption.

2. Significant increments of energy efficiency are achieved by extracting thermal exhaust-gas energy upstream of the ATU engine and 0.8 kW out of 4,8 kW of ORC's turbo generator are calculated. This higher output is due to the increased cycle conversion efficiency achieved by recovery and higher thermal source temperature, which prevails over thermal energy decline recovered from the exhaust gasses.

3. But it is at the expense of having a sophisticated control system that this improvement comes. Parallel control of the two evaporators tends to be a rather complex task, because the control bandwidth is restricted by the existence of RHP transfer zeroes, which is much less than the harmonic content of disturbances in true driving cycles at a crossover frequency about 0.001 s−1 rad. Because of this, its refusal and its tracking of set points are not satisfactory, as the disturbance is faster. A simple centralized control system consisting of a static disconnected and two PI loops is not sufficient to ensure acceptable dynamic performance while running the WHR.

One of the main conclusions of this study is that simple control strategies are not sufficiently effective to handle this type of system. It appears that dynamic feed-for-control actions and a fully central optimum control, which are the topic of further research, are necessary for the solution of control performance issues discussed in this last section. One way to get better dynamic efficiency is to study the process configuration (for example the heat exchanger weight and volumes). The device response can be accelerated and non-minimum step activity attenuated, while static output and costs are preserved (or theoretically improved). This is accomplished.

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