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Design, Modelling, and Control of a Waste Heat Recovery Unit for Heavy-Duty Truck Engines

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Abstract

This paper presents a feasibility study on a waste heat recovery system for heavy-duty truck engines based on organic Rankine cycle (ORC) technology. The elements of novelty of the work are: i) the proposed plant configuration, and ii) the feasibility study that encompasses the whole preliminary design workflow of the system, namely, from the thermodynamic cycle optimization and the components preliminary sizing, to dynamic modelling and the design of a PI-based control system. The conceived ORC turbogenerator employs hexamethyldisiloxane (MM) as working fluid and achieves a maximum rated mechanical power of approximately 5 kW at the design point, corresponding to a truck cruise speed and a Diesel engine power output of 85 km h^{-1} and 100 kW respectively. Regarding the dynamic performance, the higher response time of the ORC unit compared to that of the Diesel engine makes the adoption of an advanced control system necessary. In particular, the simulation of a PI-based control system shows that it becomes impossible to prevent the thermal decomposition of the working fluid when the engine operates continuously at high power levels. This case study demonstrates the importance for automotive ORC applications of performing the investigation of dynamic performance and control design already in the early design phase.

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Keywords: ORC; Waste Heat Recovery; dynamic modelling; centralized control;

1. Introduction

After a hiatus of almost 30 years, the interest of original equipment manufacturers (OEMs) into Diesel engine (DE) heat recovery systems has been renewed, as proven by the numerous research activities recently undertaken [1]. The reasons are i) the more and more stringent regulation on truck emissions, and ii) the limited improvements in efficiency nowadays achievable by evolving current Diesel technology. In future DE, whose projected efficiency is about 50% [2], almost half of the fuel energy will be still wasted as heat, notably in the cooling jacket of the engine (~15% of fuel thermal input), the exhaust gas recirculation (EGR) system for NO_x abatement (~10%), and the turbocharger exhaust gas (~25%). Among the technical options suitable for recoverying the engine waste thermal

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energy, Organic Rankine Cycle (ORC) turbogenerators ranks, arguably, as the best option [3], thanks to their relatively high conversion efficiency, simplicity, and the possibility to tailor the working fluid in order to comply with the limited volume available aboard trucks and the safety and environmental requirements of the automotive sector.

The ORC configuration commonly proposed in the literature for the application at hand (e.g. see [4], [5]) consists of a simple-cycle system whereby the thermal energy of the engine exhaust gases is harvested downstream of the turbocharger and of the exhaust after-treatment unit (ATU). However, as a consequence of the significant improvements in DE technology attained over the last decade, this solution is becoming, despite its simplicity, less attractive. At cruise conditions, the temperature of the exhaust gases at the ATU outlet is currently about 265 °C and is expected to decrease even more in the future. It results that the power output of the ORC turbogenerator is too low for its economic viability, which is arguably achieved only if the WHR unit is able to decrease fuel consumption of at least 5%. Such a target can be reached by converting into power other waste heat streams, as in the solution proposed by Lang et al. [6]: an ORC turbogenerator with a recuperative cycle arrangement and two evaporators in parallel to recover thermal energy from the exhaust gases and the EGR system. Due to considerations concerning turboexpander feasibility, the authors selected as working fluid the octamethylcyclotetrasiloxane (D4), a cyclic siloxane characterized by a relatively complex molecule. According to their calculations, such a WHR unit is able to generate 9.3 kW at cruise conditions, namely 6.2% of the engine power. This means, to a first approximation, an equivalent reduction in fuel consumption.

An aspect which was not explored in [6] is that the adoption of recuperation and of a molecularly complex fluid makes it possible and attractive to harvest the thermal energy of the exhausts upstream of the ATU, where their temperature is significantly higher, i.e. above 300 °C instead of 265 °C. A proper design of the recuperator can indeed limit the temperature drop of the exhausts across the ORC evaporator, to preserve the performance of the selective catalytic reduction (SCR) device of the ATU, whose minimum operating temperature is about 200 °C. The benefit might be a higher power output of the ORC turbogenerator, since the improvement in conversion cycle efficiency attainable through recuperation and the higher temperature of the thermal source can more than compensate for the lower amount of thermal energy recovered from the exhaust gases. This effect is expected to become more and more apparent in the future, as the DE efficiency progressively increases and thus the exhaust gases temperature reduces.

The implementation of such a concept can, however, entail complex control issues. At a first glance, the critical aspects are: i) the operation of two evaporators in parallel, which work at two relatively different temperature levels, ii) the maintenance of the exhaust gases temperature above 200 °C in off-design conditions, as well as iii) the avoidance of fluid thermal decomposition throughout the whole operating range of the DE. Since these last two constraints are tightly interrelated and most likely conflicting, it is possible that their fulfilment requires not only the design of advanced controllers, but also a modification of the mechanical design of the process: for instance, an adjustment of the size of the recuperator or of other heat exchangers, to obtain a more favourable system dynamics. At the moment, there are no established design guidelines for the control system of the application at hand. Actually, to the authors' knowledge, the dynamics of an ORC unit with two evaporators in parallel has not been studied yet.

The aim of the study documented here is, therefore, twofold: i) the assessment of the potential of a WHR unit for a modern DE, whose key-characteristic is to perform the recovery of the thermal energy from the exhaust gases upstream of the ATU; ii) the investigation of the system dynamics and controllability. The latter is deemed an essential step to properly assess the performance and the feasibility of the proposed concept.

The paper is organized as follows: Sec. 2 deals with the definition of the thermodynamic cycle of the WHR unit and the preliminary design of the cycle components; Sec. 3 describes the development of a dynamic model for the whole truck powertain resulting from the integration of the WHR unit within the DE; and Sec. 4 reports the control system design for this combined system and the evaluation of its performance. Concluding remarks are given in the last section.

2. Waste Heat Recovery Unit Design

2.1. ORC configuration and design specifications

As already mentioned, the selected configuration for the WHR unit features a single pressure-level cycle with two evaporators in parallel (see Fig. 1), to recover thermal energy from both the exhaust gases and the EGR system of the DE. It differs from that presented in [6] in two aspects: the evaporator heated by the exhaust gases is placed upstream of the ATU, and internal heat-recuperation is not applied to the working fluid loop recovering heat from the EGR system. While the former modification has been already discussed, the latter stems from the need of cooling down the recirculated gases to a lower temperature, in order to augment the average density of the air/flue gas mixture in the engine inlet manifold. This leads to an increase of the fresh charge entering into the cylinders and thereby of the DE performance.

Further enhancements of the WHR unit power output are arguably achievable only at the cost of a substantial complication of the plant layout: for instance, by adopting a two pressure-level cycle, or by exploiting other potential thermal sources, such as the air discharged by the turbo-charger or the engine cooling system. However, the thermal energy recoverable from such thermal sources is available at relatively low temperature levels, in the range of 100 −140 °C. Moreover, the small power capacity of the application discourages the use of two different expanders or the implementation of a dual admission turbine. It follows that the power output improvement that these solutions might enable is rather limited, especially in light of the design challenges their implementation implies. For this reason, they have not been investigated.

A high-speed turbine is arguably the best option for the expander, because of the higher compactness, reliability, and ability to handle larger pressure ratios than volumetric machines [6]. Notably, the last feature is essential to reach high conversion efficiency in the WHR unit. The configuration assumed for the turbine is a two stages axial machine. This flow architecture has been preferred since it seems to perform better than the radial-inflow architecture if expander down-speeding is targeted to achieve high reliability [7].

The working fluids which have been considered for the ORC unit belong to the family of siloxanes. These fluids exhibit a high molecular complexity, which allows for the realization of efficient turbines even for power outputs of few kW [6] [8], as required by the application at hand. Furthermore, they are produced in bulk quantities, and they feature limited flammability, low toxicity, and excellent thermal stability as proved by their use both as heat transfer media and as working fluids in high-temperature ORC systems [9]. Among the siloxanes, hexamethyldisiloxane (MM) is the best candidate, since it has the lowest boiling point and critical temperature. Its use, then, avoids high vacuum levels in the condenser and enables a good matching between the temperature profiles in the evaporator.

Tab. 1 summarizes the design specifications and system parameters assumed to perform the thermodynamic analysis and preliminary design of the WHR unit. A crucial parameter for the performance of such a system is the working fluid condensing pressure, which depends on the temperature of the cooling water delivered by the truck radiator. At cruise conditions, i.e. the design point of the ORC turboexpander, the mechanical power output of the engine is approximately one-third of its maximum power. It follows that the cooling capacity of the radiator is not fully exploited [4]. This extra capacity can be used to cool down the cooling water to 70 °C instead of 85 −90 °C, as required for the operation of the engine cooling system. The cooling water has, then, to be preheated before being sent to the cooling jacket, e.g. by means of the ORC condenser itself or by mixing with warmer water bypassing the radiator. Such a solution allows for a significant enhancement of the mechanical power output of the ORC unit, without adopting a dedicated cooling loop for the same system. The condensing temperature at cruise conditions is, to a first approximation, assumed equal to 85 °C.

The DE data to model the thermal sources of the WHR unit were provided by a leading truck manufacturer and pertain to a Euro 6 DE at cruise conditions, i.e. when truck speed and engine mechanical power output are 85 km h^{-1} and 100 kW, respectively. The values of the parameters used to predict the performance of the ORC components are, instead, set according to common practice for commercial ORC systems.

2.2. Preliminary Design Methodology

The preliminary design of the WHR unit is performed by adopting an innovative integrated design method, which allows for the simultaneous optimization of the thermodynamic cycle parameters and the main geometrical characteristics of the expander. The advantages offered by this approach are i) the possibility to discard cycle configurations that can result in an unfeasible turbine design solution, ii) the identification of the optimum trade-off between cycle and expander efficiency, and iii) a more accurate estimate of the ORC unit performance, as turbine efficiency is not set a priori by the designer and kept constant throughout the whole design space of the thermodynamic cycle. Such a procedure is, then, essential in the case of small scale ORC applications, since the design and performance of the turbo-expander strongly depend on the choice of the inlet thermodynamic conditions and the expansion ratio.

From a mathematical point of view, the WHR unit preliminary design consists in solving a constrained optimization problem, whose objective function is the net power output of the system. The selected optimization variables are the evaporation pressure, the degree of superheating of the working fluid at the evaporator outlet, the main geometrical parameters of the turbine cascades (e.g. inlet blade height, blade outlet angles), the pressure ratio and the reaction degree of each turbine stage. The feasibility of the design solution is ensured by imposing constraints on main turbine geometrical parameters, e.g. minimum height and maximum flaring angle of turbine blades. Due to the large amount

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

Table 1. Model assumptions

Source	$\dot{W}_{\mathrm{mec}}$ kW	$\dot{Q}_{\mathrm{EXH}}$ kW	$\dot{Q}_{\mathrm{EGR}}$ kW	p_{eva} bar	ΔT_{sh} °C	$\eta_{\mathrm{is,turb}}$ –
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

Table 2. Optimization results

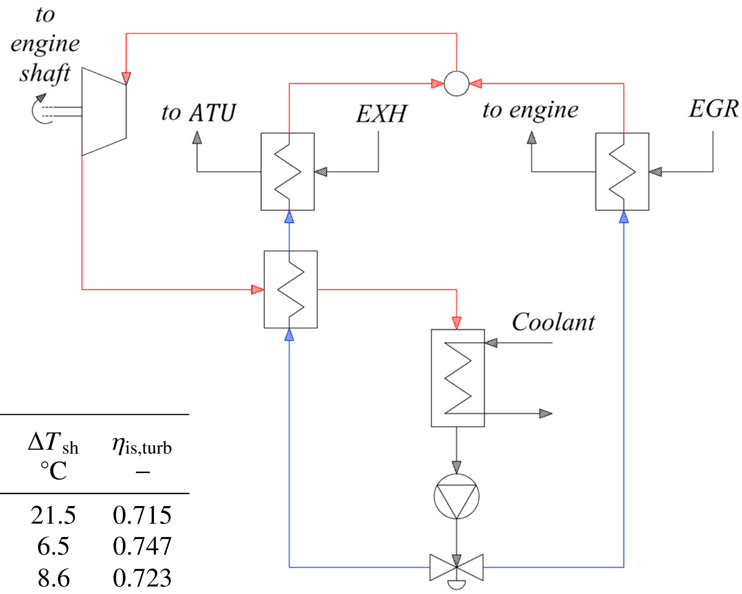

Fig. 1. ORC component layout

of design variables, and in order to avoid the convergence to local solutions, an optimizer based on a genetic algorithm is used.

The code has been implemented in a general-purpose programming environment [10]. It is coupled with an external computational library for the calculation of thermo-physical properties of the working fluid [11], and with an in-house meanline model for multistage ORC turbines. The reader is referred to the work of Pini et al.[12] for a comprehensive description of this tool and its validation, and to Bahamonde et al.[8] for more details about the adopted integrated design methodology.

As far as the heat exchangers are concerned, their optimal design is performed by a well-known commercial software for process optimization [13]. Plate heat exchangers (PHEs) have been selected because of their compactness and their potential low cost in case of high volume production. Notice that the performance and design of the heat transfer equipment for the system at hand might be strongly affected by installation issues, such as the limited space available aboard the truck. It is, however, difficult at this stage to specify constraints on the HEXs size or the maximum system volume, since they strongly depend on the specific application and they can be even met by a redefinition of the truck cabin space. Therefore, the assessment of their influence on the ORC turbogenerator performance is left for future investigation. Finally, it is worth mentioning that the maximum pressure drop in the exhaust gases across the evaporator has not to exceed 5 kPa in order to limit the increase in turbocharger backpressure, thereby the decrease of DE performance.

2.3. Results

The results of the design procedure are reported in Tab. 2. It is estimated that the WHR unit can provide 4.8 kW of additional power to the Diesel engine by recovering 36.3 $\mathrm{kW_{th}}$ in the two evaporators. Such a performance is achieved with an evaporation pressure and a degree of superheating of approximately 12.6 bar and 21.5 °C, respectively. These values allow for the best trade-off among thermodynamic cycle efficiency, recoverable thermal power, and turbine performance. Indeed, if a fixed turbine efficiency had been assumed in the thermodynamic cycle model, the optimal solution would have featured the minimum degree of superheating needed to prevent the presence of liquid droplets at the expander inlet.

[1] EXH evaporator upstream of the ATU, inlet temperature of the exhaust gases equal to $T_{\mathrm{EXH,1}}$
[2] EXH evaporator downstream of the ATU, inlet temperature of the exhaust gases equal to $T_{\mathrm{EXH,2}}$

Furthermore, Tab. 2 reports the optimal design solution for an ORC configuration harvesting the thermal energy of the sole exhaust gases and that of a WHR unit recovering also thermal energy from the EGR system, but downstream of the ATU. It is apparent that the former achieves a too low performance to make heat recovery from long-haul truck engines economically attractive. On the other hand, the latter attains a power output which is, approximately, 20% lower than that of the proposed configuration, despite the same level of complexity and the larger amount of thermal energy recovered from the exhaust gas stream. Notice also in Tab. 2 the deterioration of the turbine isentropic efficiency, as the pressure ratio increases.

Finally, the size of the HEXs is within acceptable limits and their overall weight is around 150 kg, which corresponds to an increase of 0.4% of the truck weight.

3. Dynamic Modeling

In order to study the dynamic performance of the WHR unit and its control system, a dynamic model of the whole truck powertrain has been developed using Modelica [14], an open-source, equation-based language for the modelling of systems described by differential-algebraic equations (DAEs). The models of the ORC components have been taken from the ORC library, which was validated against the experimental data of a small-capacity ORC turbogenerator for waste heat recovery in a previous study by two of the present authors [15]. On the contrary, the models for the DE system have been developed specifically for this work.

Figure 2 and 3 show the Modelica object diagram of DE and ORC unit, respectively. The PHEs of the WHR system are modeled by approximating their topology with a pure counter-current 1D arrangement and by using simplified heat transfer and pressure drop correlations. These empirical equations have been tuned against the predictions of the steady state models used during the preliminary design phase, so as to reproduce with reasonable accuracy the performance of the components in off-design conditions. The ORC pump and turbine are described by quasi-static models employing precalibrated maps for the prediction of the efficiency and the flow characteristic of the two machines. Notably, they consist of tables reporting values for reduced flow or isentropic efficiency as a function of pressure ratio and speed of revolution, within the operating range of the considered component. The turbine maps have been calibrated following the methodology proposed in [16], whereas those of the pump have been derived by scaling and fitting the data of an existing volumetric pump, specifically designed for operating conditions similar to those of the application at hand. The turbine shaft is connected to that of the engine through a gearbox.

The DE dynamic model is based on the lumped parameter model developed by Wahlstrom and Eriksson [17]. It accounts for the dynamic characteristics of the exhaust and intake manifolds, the cooling loop, the turbocharger and the EGR circuit. The dynamics of the combustion process is, on the other hand, neglected, since it is of orders of magnitude faster than that of the WHR unit, and complete combustion is considered.

This engine model [17] needs some extensions before being coupled with the dynamic model of the ORC unit and of the other components of the powertrain. Notably, the required modifications are: i) the implementation of a sub-model to reproduce the operation of the engine control unit, and ii) the evaluation of the exhaust gas temperature

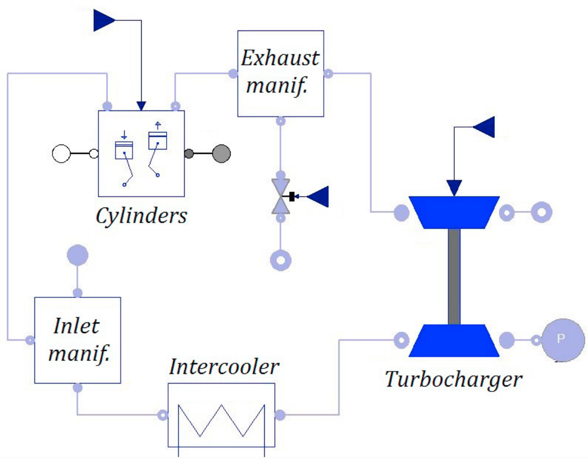

Fig. 2. Object diagram of the engine

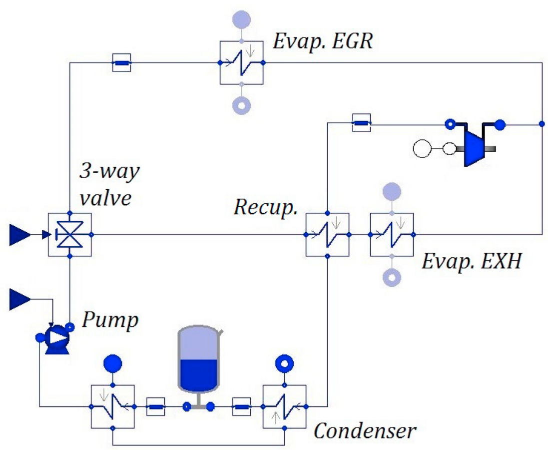

Fig. 3. Object diagram of bottoming cycle

at the turbocharger outlet. To this end, for each controlled variable, i.e., the fuel consumption, the opening of the EGR valve, and the actuator position regulating the turbocharger nozzle vanes, a static control map was calibrated on the basis of the measurements made available by the truck manufacturer. These experimental data consist of test bench measurements at different engine loads and steady-state conditions. The developed maps define the optimal set-point of the controlled variables as a function of the engine rotational speed and torque. Regarding the prediction of the exhaust gas temperature, the expansion process in the turbine is approximated by a polytropic expansion, whose index n was tuned according to experimental data. In this way, it is possible to easily account for the thermal losses in the turbocharger.

Finally, the DE dynamic model has been validated by comparison to transient data collected during a drive-test performed by the truck OEM. The interested reader is referred to Trabucchi [16] for a detailed description of the engine model and its validation.

4. Waste Heat Recovery Control

4.1. Control Objectives

The two primary requirements of the control system of the WHR unit to ensure a proper operation of the engine are i) to prevent a too low temperature of the exhaust gases in the SCR system and ii) to avoid the thermal decomposition of the organic fluid. Thus, the most suitable controlled variables seem to be the exhaust gas temperature T_{scr}at the evaporator outlet and the degree of superheating $\Delta T sh$ at the turbine inlet. At the same time, the controller should:

1. maximize the ORC mechanical power output in all the operative conditions;
2. guarantee a minimum degree of superheating to avoid the presence of liquid droplets at the turbine inlet;
3. cool down the EGR gas flow as much as possible in order not to reduce the ICE efficiency;
4. keep the evaporator pressure subcritical to avoid compromising the mechanical integrity of the PHEs.

Since the control variables of the system are only two, namely the pump rotational speed and the three-way valve at the evaporators inlet, these objectives can be simultaneously pursued only by selecting appropriate set points for the controlled variables according to the specific operating conditions of the engine. To this purpose, the controller makes use of two optimal set-points maps which cover the entire operating range of the WHR unit. Their calibration has been performed by optimizing the ORC performance in off-design conditions subject to the control system requirements.

4.2. Control Architecture

The choice of the control scheme depends on the mutual interactions between manipulated and controlled variables, and on the control performance requirements. These interactions can be evaluated by calculating the Relative Gain Array (RGA) matrix Λ of the system [18], a function of the process trasfer functions gains g_{ij} that suggests the correct coupling between manipulated and controlled variables (see Eq.1). When the RGA is close to the identity matrix, it is possible to adopt a decentralized control architecture with good performance, otherwise a centralized architecture is necessary. This analysis has been performed on linearized models [18] around the nominal operating point and a limited number of off-design condition. The mean value of λ_{11} is around 2.5, meaning that the system is strongly coupled and a centralized control architecture is necessary.

The simplest centralized control architecture is the statically decoupled scheme, shown in Fig. 4. The decoupler K (a 2x2 constant gain matrix) is designed so that the RGA of the cascaded connection of decoupler and process $K \times G(s)$ is the identity matrix, and can thus be controlled with independent single-input, single-output PI(D) controllers [19].

The transfer functions of the linearized process change significantly with the operating point, so that a fixed-parameters controller is not effective on the entire operating range. In order to reduce the non-linearities of the system, it was found convenient to use as virtual control variables the two mass flow rates of the evaporators, normalized to their optimum values found during the set-points optimization (δw_{EGR} and δw_{EGR}).

Finally, the analysis of the multivariable Right-Half-Plane (RHP) transmission zeros [18, 91] of the transfer functions $G(s)$ reveals that the process is non-minimum phase. This implies a limitation on the maximum bandwidth of the controllers, which is around $\omega_c = 0.01\ \mathrm{rad\,s^{-1}}$. It is important to stress that this limitation is intrinsic in the process dynamics, no matter how sophisticated the control system is; relaxing this constraint would require a re-design of the process itself. This is an interesting topic for future research, but it is outside of the scope of this work.

$$\Lambda = \begin{bmatrix} \lambda_{11} & 1-\lambda_{11} \\ 1-\lambda_{11} & \lambda_{11} \end{bmatrix} \quad \text{with} \quad \lambda_{11} = \frac{1}{1 - \dfrac{g_{12}\, g_{21}}{g_{11}\, g_{22}}},$$

$$G(s) = \begin{bmatrix} \dfrac{\Delta(\Delta T_{sh})(s)}{\delta w_{EGR}(s)} & \dfrac{\Delta(\Delta T_{sh})(s)}{\delta w_{EXH}(s)} \\ \dfrac{\Delta T_{scr}(s)}{\delta w_{EGR}(s)} & \dfrac{\Delta T_{scr}(s)}{\delta w_{EXH}(s)} \end{bmatrix} = \begin{bmatrix} G_{11}(s) \; G_{12}(s) \\ G_{21}(s) \; G_{22}(s) \end{bmatrix} \tag{1}$$

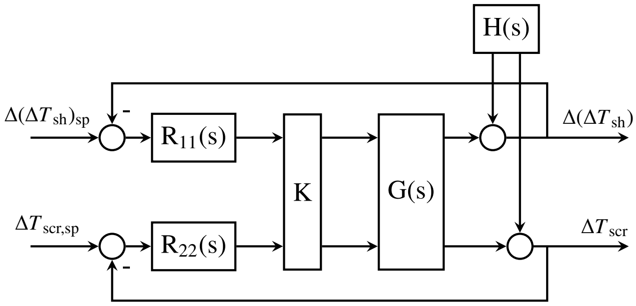

Fig. 4. Reference control scheme

4.3. Control Performance in Real Drive Conditions

Once designed, the control system has been tested in simulation with two different driving cycles. The first test (case A) consists of a series of slow velocity ramps, i.e. a disturbance H(s) whose main component in the frequency domain is around ω_c. The second test (case B) is performed with a real highway velocity profile, with a strong initial acceleration. A fast variation of the energy content and mass flow of the flue gases occurs, i.e. the disturbance variables act at a frequency higher than the closed-loop crossover frequency.

While in case A the control system successfully achieves the control objectives (Fig. 5), in case B it fails (Fig. 6): the control variables respond much faster to a disturbance variation than to a set-point variation, meaning that the disturbance rejection is very weak for signal frequencies higher than $0.01\,\mathrm{rad\,s^{-1}}$. This is not acceptable from the performance point of view, and also from the safety point of view, since the working fluid reaches too high temperatures (Fig. 7). This result is clearly explained by the presence of transmission zeros, which limit the control bandwidth to a frequency value much lower than the relevant harmonic components of disturbance. The most affected control loop is that acting on the degree of superheating: as depicted in Fig. 6, the control variable and its set-point are totally out of phase due to the non-minimum phase behaviour of the control loop transfer function during the first 20/30 seconds of simulation.

5. Conclusions

A new ORC configuration for waste heat recovery from heavy-duty truck engine has been investigated, as well as a simple control strategy. The results of the study suggest the following conclusions:

1. the power output delivered by the proposed ORC configuration at cruise conditions is 4.8 kW, which corresponds to almost 5% of the mechanical power provided by the DE. Assuming a linear relationship between engine power and fuel consumption, the latter should reduce by the same percentage.
2. the power output increase achievable by harvesting the thermal energy of the exhaust gases upstream of the engine ATU is significant and is estimated at 0.8 kW out of 4.8 kW provided by the ORC turbogenerator. This

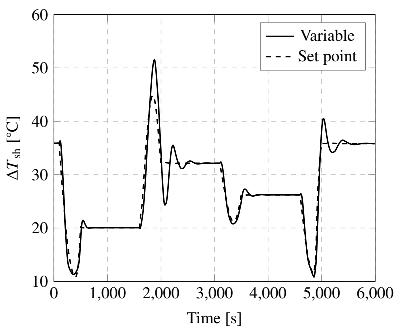

Fig. 5. ΔT_{sh}, test case A

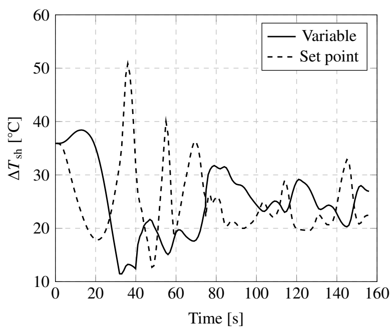

Fig. 6. ΔT_{sh}, test case B

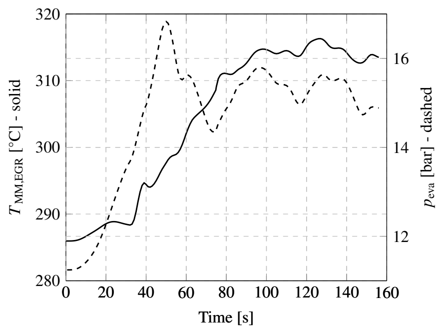

Fig. 7. $T_{EGR,wall}$ and p_{eva}, test case B

higher performance stems from the greater conversion efficiency of the cycle obtained through recuperation and the higher temperature of the thermal source, which prevails over the decrease in thermal energy recovered from the exhaust gases.

3. such an improvement comes, however, at the cost of requiring a more sophisticate control system. The control of the two evaporators in parallel appears a rather complicated task, since the control bandwidth is limited by the presence of RHP transmission zeros, at a crossover frequency around 0.01 rad s^{-1}, which is much slower than the harmonic content of disturbances in real driving cycles. Therefore, since the disturbance is faster than the process, its rejection and the set-point tracking are not satisfactory. A simple centralized control system consisting of a static decoupler and two PI loops is then not enough to guarantee acceptable dynamic performance when the WHR unit operates under real driving conditions.

A key outcome of the present study is that simple control strategies are not effective enough to cope with this kind of systems. To solve the control performance issues discussed in the previous section, it seems necessary to adopt dynamic feed-forward control action, as well as fully centralized optimal control, which is the subject of further research. Another option is to review the process design (e.g., the heat exchangers mass and volumes) in order to obtain a more favourable dynamic behaviour. This can be achieved by making the system response faster and by attenuating the non-minimum phase behaviour, while keeping (or possibly improving) static performance and costs. This requires a comprehensive optimization-based approach to the system design which represents another interesting research topic for the future.

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