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Temperature Control of Evaporators in Automotive Waste Heat Recovery Systems

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Abstract

This paper presents a control strategy for the steam generation process in automotive waste heat recovery systems that are based on the subcritical Rankine cycle. The central question is how to regulate the flow of water into the evaporator such that dry steam is generated at its outlet, subject to large variations in the heat input. Tight control of this process increases the amount of recovered energy while ensuring safe system operation. The method consists of inversion-based feedforward combined with output feedback on the temperature of the evaporator, which is estimated using exhaust gas measurements. As this method does not require a high fidelity evaporator model, it is easy to implement. It is demonstrated on an experimental setup, where the exhaust flow is imitated by electrically heated air. On an automotive driving cycle, steam was generated reliably with a superheating temperature of 10–20 [K].

\textsuperscript{1} Introduction

Over 97\% of the actively publishing climate scientist are convinced that global warming is caused by human activities [1]. The majority of them agree that the mechanism behind global warming is the greenhouse effect. The transportation sector account for 14\% of the emitted greenhouse gas globally. Waste Heat Recovery (WHR) systems increase the fuel economy of an Internal Combustion Engine (ICE), and thereby decreasing its CO\textsubscript{2} emissions, by converting a portion of its waste heat into useful energy. Reported fuel economy improvements range from 1 to 6\% [2]. A candidate working principle for automotive WHR is the subcritical Rankine Cycle (RC) [3]. In such a system, water is converted into steam by extracting thermal energy from the exhaust gas flow. Subsequently, the steam is fed to an expander that generates mechanical work. Finally, the cycle is closed by condensing the steam back into water. As the exhaust gas flow of an ICE has a reasonably low temperature, RC based WHR systems run at low evaporation pressures,
typically up to 15 [bar]. The low pressure RC is prone to flow boiling instabilities [4], adding to the challenges of control design.

In a RC based WHR system, steam generation is widely regarded to be the most challenging sub-process to control. Among the challenges are the large variations in the heat input, slow dynamics of the evaporator wall and non-linearities related to the boiling process. However, it is crucial that the produced steam is superheated as droplets cause wear in the expander. Advanced control strategies for automotive WHR systems with organic working fluids are presented in [5–7]. Experimental studies into the low pressure RC report great challenges regarding the design of stabilizing and robust controllers. In [8], these stability problems are due to high input-output sensitivities, varying time delays and unknown temperatures inside the evaporator. In [9], non-linearities and pressure-temperature coupling were pointed out to be the main causes. Feedback control of the superheating temperature on itself could not stabilize the system in [10] due to time delays. Similar issues were encountered in the low pressure RC of this study, leading to the avoidance of direct feedback control on the superheating temperature altogether. Instead, even more so as in [11], the emphasis is laid on the temperature of the evaporator itself.

This paper has three main contributions. First, the dominant behavior of the evaporator is modeled using partly linear dynamics, enabling the application of linear model based control design techniques. The parameters of this model are identified directly from simple experiments, such that a high fidelity model of the evaporator is not required. Second, it is shown that the superheating temperature can be controlled indirectly with the temperature of the evaporator itself. This is especially useful when the steam generation process is unstable or during system starts, as then there are no consistent steam measurements available. Third, the experimental demonstration is performed extensively, including a cold system start and a wide range of heat input frequencies. The paper is organized as follows. Section 2 presents the control synthesis model that is identified in Section 3. Section 4 presents the controller which is experimentally demonstrated in Section 5. Finally, conclusions are drawn in Section 6.

2. Evaporator Modeling

This section presents the evaporator model that is used for control design. More specifically, it has the following purposes. First, to describe the evolution of the evaporator temperature subject to variations in the heat input and water inflow. Second, to couple the evaporator temperature to the superheating temperature at the outlet of the evaporator. Fig. 1 (a) shows a schematic representation of a once-through, counter-flow evaporator and depicts the measurements that are assumed to be available during this study. The \( T \) stands for temperature, \( p \) for pressure and \( m \) for mass flow. The subscript \( g \) represents the exhaust gas and \( f \) the fluid side at either the inlet \( in \) or the outlet \( out \). The ambient temperature is denoted by \( T_{amb} \). The uniform evaporator wall temperature \( T_{evap} \) is not directly measurable but plays a central role in this study. The mass flow of the water flowing into the evaporator \( \dot{m}_f \) is assumed to be controlled.

The thermal inertia of the evaporator wall is the dominant dynamic effect as the flow dynamics of both the gas and working fluid are much faster [12], and therefore are assumed to be in steady state. The inertia effect is captured by the following energy balance equation:

\[
c_{he}\frac{dT_{evap}}{dt} = \dot{Q}_g - \dot{Q}_f - \dot{Q}_l,
\] (1)
where \( c_{hc} \) is the lumped heat capacity, \( Q_g \) the heat flow from the gas to the evaporator, \( Q_f \) the heat flow from the evaporator to the working fluid and \( Q_l \) the heat loss. The evaporator temperature \( T_{evap} \) is referred to as the system state, as its evolution is governed by (1). The heat loss is assumed to be dominated by natural convection, represented by:

\[
\dot{Q}_l = c_l(T_{evap} - T_{amb}),
\]  

(2)

were \( c_l \) is the heat loss parameter. The flow \( \dot{Q}_g \) is dominated by forced convection and is modeled as proposed in [6]:

\[
\dot{Q}_g = c_p c_g \dot{m}_g \left( \frac{T_{g,in} + T_{g,out}}{2} - T_{evap} \right),
\]

(3)

implicitly assuming a linear decrease in gas temperature over the evaporator. Here \( c_p \) is the known specific heat of the gas and \( c_g \) the parameter related to the heat transfer of the gas side. The outlet gas temperature \( T_{g,out} \) is not regarded to be a valid model input as it depends on the system state \( T_{evap} \). In order to eliminate \( T_{g,out} \) from (3), a second equation is introduced:

\[
\dot{Q}_g = c_p \dot{m}_g (T_{g,in} - T_{g,out}).
\]

(4)

Equating (3) to (4) results in:

\[
T_{g,out} = \frac{\dot{m}_g^0 \left( \frac{T_{g,in}^0 - \frac{1}{2} c_g}{\dot{m}_g^0 + \frac{1}{2} c_g} + c_g T_{evap} \right)}{c_g},
\]

(5)

which can be substituted into (4) to arrive at:

\[
\dot{Q}_g = \frac{c_p \dot{m}_g (T_{g,in} - T_{evap})}{\dot{m}_g^0 + \frac{1}{2} c_g}.
\]

(6)

Note that this equation only depends on the measurable characterization of the incoming exhaust gas flow \( \{\dot{m}_g, T_{g,in}\} \) and the current state \( T_{evap} \), and therefore is a valid model input. However, (6) renders the ordinary differential equation (1) non-linear in these incoming exhaust gas flow measurements. Equation (5) can alternatively be written into:

\[
T_{evap} = \frac{T_{g,out} (\dot{m}_g^0 + \frac{1}{2} c_g) - T_{g,in} (\dot{m}_g^0 - \frac{1}{2} c_g)}{c_g},
\]

(7)

enabling an online, measurement based estimation of the evaporator temperature, which is very useful for control purposes. Finally, an equation for the heat flow to the working fluid is derived. It is based on the assumption that the system is in normal operation, e.g., cold water is converted into superheated steam. In low pressure RC’s, the enthalpy difference over the evaporator only varies slightly and therefore is assumed to be the known constant \( \Delta h_{f,0} \), resulting in:

\[
\dot{Q}_f = \dot{m}_f \Delta h_f, \quad \Delta h_f := (h_{f,in} - h_{f,out}) \approx \Delta h_{f,0},
\]

(8)

rendering the model (1) linear in the control input \( \dot{m}_f \), which is very convenient for control design. As none of the unknown model parameters \( \{c_{hc}, c_l, c_g\} \) are related to steam production, they can be identified by fitting (1)(2)(5) and (6) to a dry heat-up experiment, e.g., \( \dot{Q}_f = 0 \) [kW].

Although the dynamical model does not depend on the steam conditions, it still needs to be linked to the superheating temperature as this is the main control objective. This is accomplished by the following empirical relation:

\[
\dot{Q}_{f,ss} = c_{10} T_{evap} - c_{01} T_{f,ss} - c_{00}, \quad \{c_{10}, c_{01}, c_{00}\} > 0,
\]

(9)

meaning that, in steady state, more steam can be produced if the evaporator temperature is higher or when the required superheating temperature \( T_{f,ss} \) is lower. The parameters \( \{c_{10}, c_{01}, c_{00}\} \) are obtained by performing steady state steam generation experiments representative for normal operation. Substitution of (2)(6) and (9) into the steady state of the energy balance (1), so \( \dot{T}_{evap} = 0 \) [K/s], results in:

\[
T_{evap}^{ss} = \left( c_{01} T_{f,ss}^{ref} + \frac{\dot{m}_g c_p c_g T_{g,in}}{\dot{m}_g^0 + \frac{1}{2} c_g} + c_{00} + c_l T_{amb} \right) \left( c_{10} + \frac{\dot{m}_g c_p c_g}{\dot{m}_g^0 + \frac{1}{2} c_g} + c_l \right)^{-1},
\]

(10)

describing the evaporator temperature which will approximately result in a certain superheating temperature in steady state, given the current heat input. Fig. 1 (b) provides a schematic interpretation of (10).
3. System Identification

Fig. 2 (a) shows the experimental demonstrator, available at Eindhoven University of Technology, that is used in this study. The evaporator consists of 5 parallel spirals made of stainless steel tubes with an inner diameter of 6 [mm] and an outer diameter of 10 [mm]. The diameter of a single spiral is 60 [mm] and its length is 50 [cm]. A controlled flow of electrically heated air, with a maximum capacity of 150 [kg/h], imitates the exhaust of an ICE. Demineralized water is pumped through the evaporator by a positive displacement pump. The mass flow of the working fluid is measured and controlled with a gear flow meter. The steam is expanded over a valve, imitating a turbine nozzle.

The model presented in Section 2 is fitted to a heat-up and cool-down experiment of the evaporator, where the mass flow of the air was 80 [kg/h] and no steam was generated ($\dot{m}_f = 0$ [kg/h]). The top plot of Fig. 2 (b) shows a similar experiment that is used for validation. Here the evaporator was heated-up and cooled-down dry with a mass flow of air of 30 [kg/h] and subsequently 130 [kg/h]. The middle plot of Fig. 2 (b) shows the prediction error $\Delta T_{g,\text{out}} = T_{g,\text{meas}} - T_{g,\text{out}}$ for this experiment. When the inlet gas temperature was in steady state, the Root Mean Square (RMS) prediction error is 4 [K]. Overall, the RMS error is 7 [K] and the maximum prediction error is 18 [K]. Therefore, it is concluded that the dominant dynamics of this particular evaporator can be represented by a lumped mass model and that (7) gives an estimation of the evaporator temperature.

The bottom plot of Fig. 2 (b) shows that the estimation of the evaporator temperature (7) exhibits non-physical behavior during strong transients. This is mainly a result of neglecting the residence time of the air inside the evaporator. A Kalman filter [13], shown in Fig. 3 (a), improves the prediction of the evaporator temperature by forcing it to behave like the identified model. The general working principle is as follows. The heat flow from the gas $Q_g$ (4) and to the working fluid $Q_f$ (8) are back measured, where the enthalpy difference of the working fluid is determined using IAPWS IF-97 lookup tables. The Kalman filter computes the heat balance (1) and outputs a weighted average of this calculation and the measurement based estimation (7). The weighted average is used as system state for the next sample time. As can be seen in the bottom plot of Fig. 2 (b), the Kalman filter is able to mitigate the non-physical behavior (7) while still having fast convergence to (7) when the strong transient resides.

Fig. 3 (b) shows the fit of the empirical equation (9), relating the evaporator temperature to the superheating temperature in steady state. The RMS error of the fit is only 0.07 kW. Therefore, it is expected that the superheating temperature can be accurately controlled by tracking the evaporator temperature.

Table 1. Parameters of the experimental demonstration.

<table>
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<tr>
<th>Parameter</th>
<th>$c_{hc}$</th>
<th>$c_l$</th>
<th>$c_g$</th>
<th>$c_p$</th>
<th>$\Delta h_{f,0}$</th>
<th>$T_{\text{thresh}}$</th>
<th>$c_{l0}$</th>
<th>$c_{10}$</th>
<th>$c_{l1}$</th>
<th>$K_p$</th>
<th>$\omega_c$</th>
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<td>19.0</td>
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<td>3.59·10^{-3}</td>
<td>0.05</td>
</tr>
<tr>
<td>Unit</td>
<td>J/K</td>
<td>W/K</td>
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<td>J/kgK</td>
<td>kg/kg</td>
<td>W/K</td>
<td>W/K</td>
<td>kg/sK</td>
<td>rad/s</td>
<td>rad/s</td>
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</table>

Fig. 2. (a) Schematic representation of the experimental setup; (b) Model validation (top, middle) and Kalman filtering effect (bottom).
4. Control Synthesis

In a RC based automotive WHR system, the purpose of the evaporator controller is to reliably generate superheated steam, even when the heat input varies intensively. Tight control of this process allows for lower superheating temperatures, maximizing the recovered thermal energy [14]. According to (10), controlling the superheating temperature can be achieved by tracking a specific evaporator temperature. This enables the control problem to be reduced to tracking of $T_{\text{evap}}$ and rejection of disturbances in $\dot{Q}_g$ for the linear system (1)(2) and (8), given by:

$$c_h T_{\text{evap}} = -c_l T_{\text{evap}} + Q_g - \Delta h_{f,0} \dot{m}_f + c_l T_{\text{amb}},$$

(11)

and since it is linear, it can be written into the transfer function $G(s)$, with $s$ being the Laplace variable, given by:

$$T_{\text{evap}} = G(s) [\dot{m}_f \quad Q_g \quad T_{\text{amb}}]^\top$$

where $G(s) = \frac{1}{c_h s + c_l} \left[ \begin{array}{ccc} \Delta h_0 & 1 & c_l \end{array} \right]$.

(12)

Note that (6) is deliberately not substituted for $\dot{Q}_g$ as this would render (11) non-linear. Transfer functions enable frequency domain analysis by replacing $s$ with the complex frequency $j\omega$. When for example, the transfer function $H(s)$ is exited by the sinusoidal input $u(t) = \sin(\omega t)$, the output of $H(s)$ becomes $y(t) = |H(j\omega)| \sin(\omega t + \angle H(j\omega))$. In this study, only the magnitude $|H(j\omega)|$ is of interest as this provides the amplification of signals within the system and theoretical stability is not an issue. A far more elaborate and rigorous explanation of frequency domain analysis and control synthesis is provided in [15]. The fact that the evaporator control problem is now in a linear framework enables the application of many well established control techniques. In this study, closed loop shaping is proposed.

Fig. 4 (a) shows the control structure used in this study. It consists of an inversion-based feedforward and a dynamic output feedback controller. The goal of these controllers is to track the reference evaporator temperature $T_{\text{evap}}$ up to the crossover frequency $\omega_c$. However, the control input $\dot{m}_f$ should not contain significant energy for frequencies over $\omega_n$, as fast changes in the fluid pump cause the superheating temperature to behave erratically.

The inversion-based feedforward controller $C_{FF}(s)$ provides approximately the correct control input $u_{FF}$, based on knowledge of the system and the incoming reference and disturbance signals. This is achieved by swapping the roles of $T_{\text{evap}}$ ($T_{\text{evap}}^{\text{ref}}$) and $\dot{m}_f$ ($u_{FF}$) in (12), resulting in:

$$u_{FF} = C_{FF}(s) \left[ T_{\text{evap}}^{\text{ref}} \quad Q_g \quad T_{\text{amb}} \right]^\top$$

where $C_{FF}(s) = \frac{\Delta h_0^{-1}}{\omega_n^{-1} s + 1} \left[ \begin{array}{ccc} -c_h s - c_l & 1 & c_l \end{array} \right]$.

(13)

where the low pass filter, with time constant $\omega_n$, is added to smoothen the control input. For online implementation, (6) is used to calculate $\dot{Q}_f$ in (13). As the output of the feedforward controller will not be perfect, the control input is corrected by a feedback signal $u_{FB}$. In process control, a proportional-integral controller, given by:

$$u_{FB} = C_{FB}(s) e$$

where $C_{FB}(s) = K_p \left( \frac{1}{\omega_c^{-1} s + 1} \right)$.

(14)
achieves satisfactory performance [15]. Here $K_p$ is the feedback gain and $\omega_c^{-1}$ the integration time. For implementation, the estimation of the evaporator temperature of the Kalman filter is used as controlled output, see Fig. 3 (a). The reference evaporator temperature $T_{evap}^{ref}$ is generated by low passing (10) with crossover frequency $\omega_c^{-1}$, in order not to harm the steady state assumption in (10) too much, see Fig. 4 (a). The total control input is given by $\dot{m}_f = u_{FF} + u_{FB}$.

Fig. 4 (a) Control structure combining inversion-based feedforward with dynamic output feedback. (b) Frequency response of both sensitivity (15) and process sensitivity (16) functions.

Tuning of the control parameters $\{\omega_u, \omega_c, K_p\}$ is performed by shaping two closed loop transfer functions. The control loop is closed by applying the connections shown of Fig. 4 (a) using $G(s)$ (12), $C_{FF}(s)$ (13) and $C_{FB}(s)$ (14). The signals of interest are the tracking error $e$ and the control input $\dot{m}_f$. Deriving the corresponding transfer functions is a straightforward procedure. However, due to space limitations it is not performed here. Furthermore, the effect of the ambient temperature is omitted as its impact is limited due to proper insulation of the evaporator. To shorten notation, the numbers in the subscripts refer to the entries in the corresponding transfer function matrices. The sensitivity function $S(s)$ is given by:

$$ e = S(s) [T_{evap}^{ref} \tilde{Q}_g]^T $$

where

$$ S(s) = \begin{bmatrix} \frac{1 - G_1(s)C_{FF,1}(s)}{1 + C_{FB}(s)G_1(s)} & -\frac{G_2(s) + G_1(s)C_{FF,2}(s)}{1 + C_{FB}(s)G_1(s)} \end{bmatrix}. $$

(15)

and the process sensitivity function $PS(s)$ by:

$$ \dot{m}_f = PS(s) [T_{evap}^{ref} \tilde{Q}_g]^T $$

where

$$ PS(s) = C_{FB}(s)S(s) + \begin{bmatrix} C_{FF,1}(s) & C_{FF,2}(s) \end{bmatrix}. $$

(16)

Fig. 4 (b) shows the magnitude of the frequency response of these transfer functions. As expected, the magnitude of $S(j\omega)$ is small below $\omega_c$ and the magnitude of $PS(j\omega)$ is small over $\omega_u$. The actual tuning of $\omega_c$ and $\omega_u$ is performed experimentally by closely examining the behavior of the superheating temperature during transients in the heat input. Generally, $\omega_c$ should be increased to achieve a constant superheating temperature and $\omega_u$ should be reduced to improve the stability of the flow boiling process. Note that the disturbances between $\omega_c$ and $\omega_u$ pose the challenge as both the error and the control input are high. The fact that the magnitude of $S(j\omega)$ decreases after $\omega_u$ is a result of the physics of the evaporator. The parameter $K_p$ should be reduced until the feedback signal $\dot{m}_F$ starts to show oscillatory behavior. High values of $K_p$ will cause problems during systems starts, as here the error $e$ will be rather large.

The method presented in this section has the following key features. First, the feedback signal is always available, allowing for controlled systems starts and increased robustness. Note that feedback control of the superheating temperature is only possible during normal and correct system operation. Getting the system to run correctly only by feedforward is not trivial, especially due to the wide range of initial evaporator temperatures that can occur. Second, since the evaporator temperature does not exhibit saturation, erratic or non-minimum phase behavior, the feedback gain can be much higher. Finally, the decision making process for systems starts and shutdowns becomes straightforward. When the evaporator temperature exceeds a certain threshold value, e.g., $\tilde{T}_{evap} > T_{threshold}$. Table 1 provides the parameters used in this study.
5. Experimental Results

The control method presented in Section 4 is demonstrated on the driving cycle shown in blue in the top two plots of Fig 5. The cycle is based on a dynanometer test of an Euro4 2.0 diesel passenger car running the NEDC [16]. It is altered in order to satisfy the constraints of the experimental setup. Therefore, the goal of the validation experiment is not to show feasibility of automotive WHR but rather to show the performance of the control strategy. In part 1 of the driving cycle, the evaporator is started cold on the NEDC slowed down three times. Subsequently, the cycle is stretched two times and run backwards, in order to create a fluent transition to the non-time-scaled cycle in part 3. The second plot of Fig. 5 shows the air temperature measurements and the reference and estimated evaporator temperature. The third plot shows the measured working fluid mass flow and the outputs of both the feedforward and feedback controllers. The fourth plot shows the superheating temperature and the last plot of Fig. 5 the evaporation pressure.

In the first 500 [s] of the experiment, the pump was disengaged in order to heat up the evaporator in a minimal amount of time. The controllers were automatically engaged when the estimated evaporator temperature exceeded the threshold value. It took approximately 100 [s] for steam generation to initiate. In the next 1200 [s], the evaporator is brought to correct operation by active interference of the feedback controller, while the tubing behind the evaporator was heated up by the steam. The dip in the superheating temperature at 1700 [s] is regarded to be still related to the system start. The dip at 2500 [s] however, occurs due the transition from the urban cycles to the extra urban cycle. By the time part 1 of the experiment had ended, the feedback contribution to the control signal had diminished, demonstrating the quality of the feedforward controller. Part 2 of the experiment contains disturbances in the medium frequency range. Again, by transitioning between the extra urban and urban drive cycle, some dips in the superheating temperature occurred. During part 2, the superheating temperature was above 5 [K] for 98% of the time and between 10 and 20 [K]...
for 79% of the time. Finally, part 3 of the experiment shows that high frequent disturbances are naturally rejected by the system resulting in less variations in the superheating temperature then in part 2. However, shutting down the heat input, right after full load, proved to be too challenging as the superheating temperature went to 0 [K] for 1 minute. The fact that the system recovered from this dip is an excellent example of the robustness of the control strategy. In the 2 hour experiment, the system delivered for over 75% of the time steam with a superheating temperature above 5 [K] and for 55% of the time between 10 and 20 [K].

6. Conclusion

This paper presents and demonstrates a novel control strategy for evaporators in automotive WHR systems. As this method does not rely on a measurement of the superheating temperature, it omits the stability issues reported in literature. Furthermore, it is shown that the dominant dynamics of a prototype evaporator were captured sufficiently well by a lumped mass evaporator model. Additionally, this model provides a means to estimate the evaporator temperature from exhaust gas measurements. By relating the evaporator temperature to the superheating temperature, the control problem is reduced to tracking and disturbance rejection of a linear system, enabling the application of well established control design techniques. In this case, closed loop shaping is used to tune the parameters of a dynamic feedback and inversion-based feedforward controller. The control strategy handles system starts and shutdowns naturally by monitoring the evaporator temperature. Therefore, no minimum flow is required resulting in shorter start-up times and no evaporator flooding at shutdown. The derived inversion-based feedforward controller can be easily obtained as no simulation-type evaporator model is required. The controller was able to run the low pressure steam generation process reliably at a low superheating temperature, maximizing the amount of heat recovered.

Future recommendations are to extend the approach for a more realistic demonstrator, including an ICE, off-the-shelf evaporator and possibly an organic working fluid.

References