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Model-based control of exhaust heat recovery in a heavy-duty vehicle

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ABSTRACT

For an exhaust heat recovery (EHR) system, this paper presents a model-based controller that is operated in a common automotive electronic control unit (ECU) and tested in a heavy-duty vehicle on the road. The EHR system is based on an Organic Rankine cycle (ORC) and improves the fuel efficiency. The sensitive vapor quality at the expander inlet is regulated by the pump using a 2-degree-of-freedom design. The feedforward part employs an inverse of a modified Moving-Boundary model, combined with online adaption of parameters. The feedback path utilizes gain-scheduling and LQR based on a Finite-Volume model. The controller performance is compared in simulation and experiment using a real ORC in a heavy-duty vehicle (HDV).

1. Introduction

Frequent discussions about emissions of road traffic draw attention on the fuel consumption of cars and trucks and urge automotive companies to improve their fuel efficiency. Besides aerodynamics or engine/powertrain optimizations, engine downsizing, electrification of auxiliary systems or hybridization, waste heat recovery (Singh, 2015) has been identified as a potential solution. It is motivated by the fact that the largest fraction of fuel energy is lost in the heat flow via exhaust gas and cooling, which can temporarily sum up to about 50% (Görsmann, 2015), depending on the engine load. Utilization of this huge amount of waste heat is an obvious idea and can be accomplished with various technologies (Saidur, Rezaei, Muzammil, Hassan, Paria, & Hasanuzzaman, 2012). A turbo compounder (Aghaali & Ångström, 2015) can utilize the remaining pressure difference between exhaust manifold and environment. This increases the back-pressure and pumping losses of the engine. The pure heat in the exhaust gas can be converted into usable electrical or mechanical power by thermoelectric generators (Orr, Akbarzadeh, Mochizuki, & Singh, 2016) or Rankine processes (Sprouse & Depcik, 2013). In the latter the exhaust heat vaporizes a working fluid which subsequently produces mechanical power by expansion in an expander (see Fig. 1). The process promises a good cost to efficiency ratio for the amount of heat flows in heavy-duty vehicles (HDVs) (Wang, Zhang, Peng, & Shu, 2011). The classical Rankine process (Moran & Shapiro, 2009) with water as working fluid is common in power plants. With medium heat source temperature, as with exhaust gases, organic working fluids are preferred (Bao & Zhao, 2013). The idea to utilize the organic Rankine cycle (ORC) in the automotive sector has already been proposed several decades ago (Tona & Peralez, 2015) but has attracted increasing interest over the past few years: e. g. at TNO Automotive (Willems, Kupper, & Cloudt, 2012), Cummins (Nelson, 2008), Volvo (Howell, Gibble, & Tun, 2011) Honda (Ibaraki, et al. 2007) or BMW (Freymann, Strobl, & Obieglo, 2008).

With the implementation of ORCs in vehicles, new challenges emerge, like the desire to use cheap and robust components while preserving a good thermal efficiency. An absolutely necessary step to place the technology on the market is the design of control algorithms that have a good performance but still can be operated on a standard automotive electronic control unit (ECU). Therefore, this paper focuses on a model-based control technique that is implementable in a common automotive ECU, which is restricting in terms of computational performance and memory.

1.1. ORC system

A large number of publications deal with ORC design, dimensioning, working fluids and estimation of the potential (e. g. Grelet, Reiche, Lemort, Nadri, & Dufour, 2016; Katsanos, Hountalas, & Pariotis, 2012). The ORC considered in this paper is implemented in a HDV, uses ethanol as working fluid and consists of the four processes depicted in Fig. 1:

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Nomenclature

V	volume
A	heat transfer area at wall
α	heat transfer coefficient at wall
$\tilde{\alpha}$	scaled heat transfer coefficient at wall
$\hat{\alpha}$	static heat transfer coefficient at wall
$\dot{Q}$	heat flow
P	power
h	specific enthalpy
$h_{F,l}$	specific enthalpy of saturated liquid
$h_{F,g}$	specific enthalpy of saturated vapor
p	spatially constant pressure
T	temperature
$T_{F,Boiling}$	working fluid boiling temperature
$\bar{T}$	mean temperature
$\boldsymbol{T}$	vector of temperatures
ρ	density
$\dot{m}$	mass flow rate
L_1, L_2	normalized zone lengths (MB model)
$\boldsymbol{M}$	mass matrix
s_α	scaling factor
m	mass
c_p	specific heat capacity at p =const.
η	efficiency
β, γ	factors
Σ, $\hat{\Sigma}$	LTI system, observer for system Σ
τ	time constant
q, R	weights in LQR synthesis
$\boldsymbol{K}_{SF}$	vector of state feedback gains
φ	shaping functions
M	performance number
Θ	Heaviside function
Δ	difference
L	open-loop system (for the Nyquist plot)
z	normalized position coordinate
t	time coordinate
s, ω	Laplace variable, frequency
$\boldsymbol{A}, \boldsymbol{B}, \boldsymbol{C}, D, E$	system matrices
$\boldsymbol{x}$, $\Delta\boldsymbol{x}$, $\Delta\hat{\boldsymbol{x}}$	state vector, local state vector, estimated local state vector
Δw, $\Delta\hat{w}$	local additive output disturbance, estimated local additive output disturbance
u, Δu	input, local input
y, Δy, $\Delta\hat{y}$	output, local output, estimated local output
N	number of finite volumes (FV model)
k, m, n	indices

Subscripts

F	working fluid
ExG	exhaust gas
W	wall
Exp	expander
amb	ambient
in	evaporator inlet
out	evaporator outlet
d	desired
sim	simulation
meas	measurement
el	electrical
vol	volumetric
FF	feedforward
Fdb	feedback
aug	augmented
OL	open-loop
SH	superheating

Abbreviations

ORC	Organic Rankine cycle
HDV	Heavy duty vehicle
FV	Finite-Volume
MB	Moving-Boundary
ECU	Electronic control unit
2DOF	Two-degree-of-freedom
PID	Proportional–integral-derivative
LTI	Linear time invariant
LQR	Linear–quadratic regulator
LQR	Linear–quadratic-integrator control
SISO	Single input single output
GS	Gain scheduling
OP	Operating point
PM	Phase margin
IMC	Internal model control

1–2: Pumping the working fluid onto a high pressure level.
2–3: Evaporation and superheating using the exhaust gas downstream of the aftertreatment system as heat source ($T_{ExG,in} \approx$ 200 to 400 °C).
3–4: Expansion of vapor produces the mechanical power.
4–1: Condensation with cooling water as a heat sink.

The mass flow rate through the pump is set continuously by an electrical drive that accepts rotational speed as input. An expander (see e. g. Lemort, Quoilin, Cuevas, and Lebrun, 2009 for details) is enabled if the vapor quality is adequate and power could be produced. Otherwise a valve switches to the bypass path where a fixed throttle ensures separation of high and low pressure. The expander is directly coupled with a speed-controlled generator. In order to provide flexibility for the research and development process, the controller should also be suitable for the operation with a turbine instead of an expander. The present prototype-vehicle is a parallel-hybrid vehicle that allows storing the produced electrical energy in a battery if not needed immediately. Despite the large exhaust gas heat $\dot{Q}_{ExG}$, the low ORC efficiency (Xie & Yang, 2013) limits the power yield. The thermal efficiency (Moran & Shapiro, 2009) for the ideal ORC can reach up to about 17.5% but the real value is considerably lower due to imperfect processes.

The prototype setup contains temperature and pressure measurement equipment at any point ①–④ in Fig. 1, and a volume flow sensor is included between pump and evaporator. The mass flow rate of the exhaust gas as well as its temperature at inlet and outlet are required for control and obtained by an engine lookup table and temperature sensors, respectively.

The dynamic simulation and control of the evaporator outlet condition rely on a realistic model that captures the transient behavior well. Authors widely use mass and energy conservation laws, which are mostly discretized with the Finite-Volume (FV) or the Moving-Boundary (MB) approach (Bendapudi, Braun, & Groll, 2008; Cruz, Munoz, Bencomo, & Moya, 2013; Jensen, 2003). In this paper, they are utilized for the design of the feedback and feedforward paths, respectively.

1.2. ORC control

Fluctuating engine load produces strongly varying exhaust heat flows $\dot{Q}_{ExG}$, indicated in Fig. 2, which is largely caused by its varying mass flow rate. Xie and Yang, (2013) demonstrate that the cycle inefficiency during every day traffic operation is largely caused by the switching between expander and bypass mode, due to an insufficient vapor quality. Therefore, high energy recovery and the satisfaction of