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IFAC-PapersOnLine 49-11 (2016) 625–632

# Design of a Nonlinear, Dynamic Feedforward Part for the Evaporator Control of an Organic Rankine Cycle in Heavy Duty Vehicles

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**Abstract:** A model-based two degree of freedom (2DOF) controller for the evaporator outlet condition in an organic Rankine cycle (ORC) system is presented. The ORC is used in a heavy duty vehicle to convert exhaust gas heat into mechanical power which increases the fuel efficiency. The evaporator outlet condition of the working fluid is a crucial variable for the ORC efficiency which thus has to be controlled very accurately. However, the exhaust gas thermal energy, which serves as heat source to the evaporator, is highly transient depending on the actual engine load. Therefore, excellent disturbance rejection is required with help of an efficient feedforward part. Its novel derivation is based on an exact inversion of a nonlinear, dynamic evaporator model, obtained by the finite volume method that is reduced subsequently. The resulting feedforward controller is therefore very close to the exact inverse dynamic model. Furthermore, online parameter adaption enhances the steady state accuracy of the feedforward part and compensates for model uncertainties. The improvements by the feedforward controller are shown in simulations and its applicability on the vehicle control unit is demonstrated by measurements on a Daimler prototype truck under real traffic conditions.



*Keywords:* ORC, fluid dynamics, distributed-parameter systems, finite volume method, disturbance rejection, 2DOF

## 1. INTRODUCTION

In times of rising oil prices, shrinking resources and increasing traffic volume it becomes more and more important to raise the efficiency of vehicles with internal combustion engines. A large fraction of traffic consists of heavy duty vehicles which are essentially for the transportation of goods. However, they need large engines with high energy consumption, so, making them more efficient has the perspective to save a lot of fuel.
In recent years many new ideas have been investigated and realized in consumer market products. Some of which are hybrid engine strategies, predictive cruise control and combustion optimization. Even though these techniques brought considerable improvements a huge fraction of engine power still remains unused as waste heat in the hot exhaust gases released to the environment. In summary they contain about 35% of the primary fuel energy needed in the truck. This is source for the idea of converting the exhaust gas heat into mechanical power, which can be profitably done with an exhaust heat recovery (EHR) system that uses an organic Rankine cycle (ORC). It uses an organic working fluid, which is suitable for mobile applications with medium grade heat. An deeper overview about ORCs is given, for instance, in the review articles Sprouse III and Depcik (2013) and Lecompte et al. (2015). Compared to stationary ORC systems running at steady state conditions, the exhaust gas heat in an automotive application are governed by the highly transient engine load. That requires advanced control algorithms which guarantee safety regulations and good tracking performance at once, even at the presence of severely fluctuating disturbances. The derivation of approved model-based control algorithms, however, require a good dynamical representation of the essential system behavior. The main challenges lie in the evaporator and condenser dynamics, respectively, as they contain complex thermodynamical processes which include phase change of the working fluid. In recent publications two main approaches can be found for the dynamic modeling of heat exchanger with 2 phase flow: the moving boundary (MB) approach and the finite volume (FV) method. MB models divide the working fluid side into up to three volumes with moving intermediate boundaries and averaged cell properties (see Jensen et al. (2002), Luong (2013), Cuevas et al. (2009), or Shah et al. (2003)). This result in a model with just few states. If inlet or outlet condition changes between single phase and 2-phase fluid, switching MB models are required. They


Peer review under responsibility of International Federation of Automatic Control.
10.1016/j.ifacol.2016.08.091

can change the number of volume cells as demonstrated in McKinley and Alleyne (2008) or Bonilla et al. (2015)).
In the contrary, the FV method uses a finer spatial discretization with fixed boundaries (Bendapudi et al. (2004),Cruz et al. (2013)). They can achieve high accuracy by distributed evaluation of flow and heat transfer properties, and better approximation of the spatial temperature profiles. Comparisons of MB and FV models are found e.g. in Bendapudi et al. (2004) or Wei et al. (2008).
Apart from the mentioned parametric models the ORC can also be described by black box models obtained via system identification methods. Their advantages is the possibly good approximation of the dynamics and the opportunity for online identification. However, these models give less understanding of the system behavior and are less flexible towards changes in the process which could be easily reacted to by intuitive parameter adaption in the FV and MB approach.
Many applications are available for air conditioning systems (e.g. Gupta and Rasmussen (2008) or Elliott and Rasmussen (2015)), however, an increasing number of publications can be found for waste heat recovery systems operated under transient heat input (Peralez et al. (2012), Quoilin et al. (2011)), as they occur in automotive applications. Investigations has already been carried out on controller design for the outlet conditions of evaporators and condensers (see Peralez et al. (2013), Feru et al. (2013)).
The paper is organized as follows. After a short description of the system setup the mathematical model is presented. Then the design of the feedforward controller is conducted including the parameter adaption implementation. The improved performance is demonstrated with simulation and measurement results. Finally, a conclusion summarizes the work and gives further prospects for improvements and future work.

## 2. SYSTEM SETUP

The EHR system consists of the classical thermodynamic organic Rankine cycle which converts heat into mechanical power. A working fluid is transported clockwise through the four processes of the closed system which are illustrated in figure 1:

- 1-2: liquid transport and adiabatic pressure rise by the pump
- 2-3: isobaric evaporation at high pressure
- 3-4: adiabatic expansion in the expander machine or nozzle
- 4-1: isobaric condensation at low pressure.

The heat input comes from the hot engine exhaust gases which cool down in the evaporator, which is in countercurrent configuration. The condensation of the working fluid must be realized with help of the cooling water system of the engine or, alternatively, with a dedicated cooling cycle. The crucial property of the Rankine cycle is that the expander delivers much more power than needed by the feed pump. However, the maximal achievable efficiency of the present Rankine cycle under the given boundary conditions is limited to ≈ 15%, governed by thermodynamic laws.
The mechanical power produced by the reciprocating expander machine can either be used to directly support the

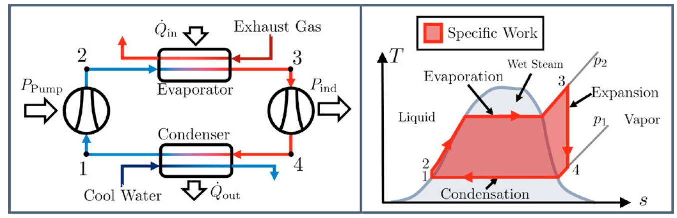


Fig. 1. Processes of the Rankine cycle (left) and corresponding T-s-diagram (right).

crankshaft or to generate electrical power in a generator. For the process (3-4) there is the option to switch between expander machine and nozzle through a bypass path. In case of warmup or inappropriate working fluid conditions at the evaporator outlet the nozzle is chosen in order to protect the expander.

## 3. MATHEMATICAL MODEL

For the control of the evaporator outlet condition only the high pressure part (from pump to expander/nozzle) of the Rankine cycle (see figure 1)) is of interest because the feedback through the low pressure part (from expansion outlet to pump inlet) can be neglected for our purpose. This can be justified by the fact that the pump is speed controlled and the expander is of positive displacement type, as opposed to turbo machines which are of dynamic type.

### *3.1 Pump*

As the pump has an internal speed controller and the fluid at its inlet is assumed to be completely liquid it can be handled as an ideal mass flow supplier, independent of the pressure ratio. Therefore, the mass flow rate is considered as the control input of the evaporator model.
A model for the little energy rise of the fluid through the pump is not further regarded here since measurements of temperature and pressure behind the pump are available. Presuming subcooled liquid, pressure and temperature can be converted to specific enthalpy with help of property tables. The losses due to transportation in the pipe from pump to evaporator are neglected. The specific enthalpy at the evaporator inlet cannot be controlled and hence has to be accepted as a disturbance input.

### *3.2 Expander/Nozzle*

The structure of the mathematical evaporator model of subsection 3.3 will require an input of the outlet mass flow which is, especially in transient condition, different from the inlet mass flow and is necessary for correct modeling of pressure dynamics. Either nozzle or expander can be chosen which differ in mass flow models.
In case of the nozzle the standard formula

$$\dot{m}_{\mathrm{Noz}} = C_{\mathrm{Noz}} \sqrt{2 \cdot \rho_{\mathrm{in}} \cdot (p_{\mathrm{in}} - p_{\mathrm{out}})} \qquad (1)$$

is chosen, where $\rho_{\mathrm{in}} = \rho(h_{\mathrm{in}}, p_{\mathrm{in}})$, $p_{\mathrm{in}}$ and $p_{\mathrm{out}}$ are the inlet and outlet conditions, respectively. $C_{\mathrm{Noz}}$ is a constant parameter proportional to the cross-sectional area of the nozzle that can be obtained by parameter identification. The dependency on the low pressure $p_{\mathrm{out}}$ is visible here but with the assumption that $p_{\mathrm{out}}$ varies only within small