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Dynamic Model of a Multi-Evaporator Organic Rankine Cycle for Exhaust Heat Recovery in Automotive Applications

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Abstract: This paper presents a dynamic model of an organic Rankine cycle (ORC) for exhaust heat recovery (EHR) in automotive applications. EHR converts the thermal energy of the exhaust gas into mechanical power. Dynamic components like evaporator, condenser, etc. are modeled based on physical conservation laws given as partial differential equations (PDE). The system involves two parallel evaporators coupled via an open t-piece which is handled numerically efficiently by introduction of algebraic constraints. An integral method improves the evaluation of the fluid properties which partly contain discontinuities. They can have negative effects on accuracy and simulation performance if evaluated inappropriately. The model is implemented by signal-based programming suitable for simulation in MATLAB/SIMULINK. This enables convenient software-in-the-loop (SIL) engineering since many control unit producers provide rapid prototype programming via MATLAB/SIMULINK. A simplified model is proposed for the increase of robustness and simulation performance or for application of controller design methods. Simulation shows both the entire ORC system compared measurements from a DAIMLER prototype truck, and comparison between complex and simplified model.

Keywords: organic Rankine cycle, fluid dynamics, finite volume method, modeling

1. INTRODUCTION

EHR systems in heavy duty vehicles convert the thermal energy of the exhaust gas, which is roughly 30% of the primary fuel energy, into mechanical power via an ORC. If the mechanical power is feed back to the powertrain EHR reduces the overall fuel consumption. Software development for the control of such complex systems requires an essential time for testing if done in the vehicle. Therefore, SIL simulation environment is usually established which requires a model which meets the trade-off between accuracy and computation effort which dictates the simulation speed.

Concerning the ORC system, there exist a number of publications about modeling of evaporators, condensers, steam machines and ORCs in general (Moran and Shapiro (2009)). Besides static models, suitable for steady state processes and components development (Quoilin and Lemort (2011), Dai et al. (2009), Wang et al. (2011)), the dynamic models put most of the effort on the adequate modeling of the dynamics in the countercurrent heat exchanger (Pettit et al. (1998)). This is reasonable since they contain the most dominant time constants. The phase change of the working fluid is especially challenging as it involves strong couplings between energy and pressure as well as large ranges of flow velocities. The governing equations are based on the mass and energy conservation for the working fluid with phase change and energy conservation for the other fluid, often gas, and the wall material between these two fluids (Pangborn et al. (2015)). Some applications also require the dynamic conservation of momentum, in especially if the propagation of pressure waves cannot be neglected. These conservation laws are in form of nonlinear PDEs, for which a spatial integration over finite volumes is the usual procedure for the derivation of an ODE system (Bendapudi et al. (2008)).

The choice of these volumes is the main difference between the following two modeling approaches. The first, called finite volume (FV) approach, divides the whole working fluid volume of e.g. the evaporator into many small ones with fixed size (Bendapudi et al. (2004), Wei et al. (2008), Jensen (March 2003)). The second, called moving boundary (MB) approach, divides into maximal three volumes, each containing all of the liquid, two-phase and vaporized fluid, respectively (Willatzen et al. (1998), Bonilla et al. (2015), McKinley and Alleyne (2008), Luong and Tsao (2014)). Their intermediate boundaries are able to move in spatial direction as the zones shrink or grow. Here, however, assumptions for the spatial profile of the specific enthalpy over a large range are necessary, for which usually linear profiles are used. Hence, its drawback compared to the FV approach is lower accuracy and worse represen-

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tation of the transport delays of the fluid, e.g. when a temperature wave is propagated through the evaporator. The low number of dynamic states in the MB model is advantageous compared to the FV model. This paper uses the FV approach as it has better behavior for the desired simulation purposes.

A Software-in-the-loop simulation of the ORC requires the implementation of the entire closed system in order to include all couplings. This requires accurate conservation of fluid mass. Due to approximation errors, e.g. in the evaluation of the fluid properties with help of look-up tables, there occurs a slight gain or loss of mass over time in our implementation, which is undesirable. This is solved pragmatically with a mass correction and will be further discussed in subsection 2.4.

In the system setup under consideration there exist two parallel evaporator coupled via an open t-piece. Feru et al. (2014), You et al. (2010) and Elliott and Rasmussen (2008), for example, show the modeling of parallel evaporators. Expansion valves usually decouple the pressure in the single evaporators. Shah et al. (2003) models the coupling by insertion of a small volume element. It has the advantage that the individual evaporator models remain unchanged and do not have algebraically coupled pressure values. However the stiffness is increased. The method proposed in this paper is applicable to parallel evaporators modeled either with the MB or the FV approach and are based on the equalization of all evaporator pressures. For the prototype system under investigation this does not mean a restriction.

Another proposal of this paper concerns the evaluation of the fluid properties, e.g. $\frac{\partial}{\partial h}\rho$. Usually they are evaluated at the center points of the finite volumes or calculated as the average of the evaluation at their boundaries. Quoilin et al. (2014) and referenced articles investigates different evaluation methods and their impact on accuracy and robustness of the simulation. The spatial discretization must not be too rough as the partial derivatives of the density, $\frac{\partial}{\partial h}\rho$ and $\frac{\partial}{\partial p}\rho$, contain discontinuities at the transition from liquid to two-phase zone. On the one hand this might cause stability problems, on the other hand loss of accuracy can originate from it. As a remedy this paper proposes an integral method that involves the precomputation of partially integrated lookup tables which smoothens the discontinuities a lot, hence allowing a smaller number of finite volumes.

The paper is organized as follows: after modeling of the individual ORC elements in section 2, which also describes the method for evaluation of fluid properties and coupling of the parallel evaporators, the closed system is build. Section 3 proposes a simplification approach for more robust and faster simulation or for application of controller design methods. After discussing simulation results in section 4 conclusion are finally drawn in section 5

2. MATHEMATICAL MODELING

This section is divided into four subsections: at first a general system overview is given, then the individual components are modeled with algebraic and dynamic mathematical equations. After interconnection of the subsystems via their input/output ports, a method for improved evaluation of fluid properties containing discontinuities is presented.

2.1 System Setup

The ORC in the automotive application under consideration consists of four main components: pump, two evaporators using two heat sources, expander/nozzle and condenser (see figure 1). Additionally, vapor pipes and a receiver are integrated for better approximation of the total fluid volume of the real system.

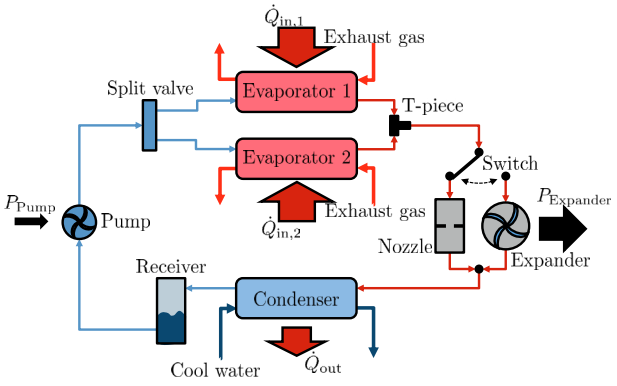

Fig. 1. Schematic configuration of the organic rankine cylce components.

2.2 Component Models

Components of the ORC model are divided in two classes: dynamic and static models. The two evaporators, the condenser as well as the pipes and the receiver are modeled dynamically with differential equations while the pump, split valve and expansion device are considered static, described by algebraic equations. The reason is that the transport of mass and energy in the dynamic components are much slower than fast pressure and mass transport dynamics in pump, split valve and expansion device.

Pump The static equation for the pump computes the pump mass flow rate and the specific enthalpy increase given the pressure difference as well as the input enthalpy, the rotational speed n_{pump}, the pump volume V_{pump} and leakage coefficient $k_{\text{leak,pump}}$:

$$\begin{aligned}\dot{m}_{\text{pump}} =&\rho(h_{\text{pump,in}}, p_{\text{low}}) \cdot n_{\text{pump}} \cdot V_{\text{pump}} \\ &- k_{\text{leak,pump}} \cdot \rho(h_{\text{pump,in}}, p_{\text{low}}) \cdot (p_{\text{high}} - p_{\text{low}}),\end{aligned} \tag{1}$$

$$\begin{aligned}h_{\text{pump,out}} =&h_{\text{pump,in}} \\ &+ \frac{1}{\eta_{\text{pump,is}}} \cdot (h_{\text{pump,out,is}} - h_{\text{pump,in}})\end{aligned} \tag{2}$$

with $h_{\text{pump,out,is}} = h\left(s\left(h_{\text{pump,in}}, p_{\text{low}}\right), p_{\text{high}}\right)$ being the specific enthalpy at the pump outlet if isentropic process is assumed (s denotes the specific entropy). The pump power is $P_{\text{pump}} = \frac{1}{\eta_{\text{pump,el}}} \cdot \dot{m}_{\text{pump}} \cdot (h_{\text{pump,out}} - h_{\text{pump,in}})$, with the electric efficiency $\eta_{\text{pump,el}}$.

Behind the pump a split valve separates the mass flow rate into two flows, each to one of the evaporators. The valve is assumed to be fast enough so that the duty cycle (DC) of the pulse-width-modulated (PWM) voltage