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Modeling and control of Rankine based waste heat recovery systems for heavy duty trucks

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Abstract: This paper presents a control oriented model development for waste heat recovery Rankine based control systems in heavy duty trucks. Waste heat recovery systems, such as Rankine cycle, are promising solutions to improve the fuel efficiency of heavy duty engines. Due to the highly transient operating conditions, improving the control strategy of those systems is an important step to their integration into a vehicle. The system considered here is recovering heat from both EGR and exhaust in a serial arrangement and use a mixture of water and ethanol as working fluid. The paper focuses on a comparison of a classical PID controller which is the state of the art in the automotive industry and a nonlinear model based controller in a simulation environment. The nonlinear model based controller shows better performance than the PID one and ensures safe operation of the system.

Keywords: Waste Heat Recovery, Modeling, Control, PID.

1. INTRODUCTION

The idea of recovering waste heat and utilize it as another form of energy is not new. Most of the actual power plants use the principle of co or even tri-generation. In the automotive industry, the most applied form of waste heat recovery system is the turbocharger which converts the waste heat into aeraulic energy by compressing the fresh air via an exhaust driven turbine. Driven by future emissions legislations and increase in fuel prices, engine gas heat recovering has recently attracted a lot of interest. In the past few years, a high number of studies have shown the interest of energy recovery Rankine based systems for heavy duty trucks engine compounding [Sprouse et al. (2013), Espinosa (2011)]. Recent studies have brought a significant potential for such a system in a Heavy Duty (HD) vehicle which can lead to a decrease in fuel consumption of about 5% and reduce engine emissions. Yet many challenges have to be faced before the vehicle integration. The first challenge deals with the correct choice of fluids and system architecture [Mago et al. (2007), Grelet et al. (2014)] and shows that system simulation is a critical part of the development work. The use of water-alcohol mixture can bring some advantages in the power recuperation and overcome both disadvantages of these fluids: high freezing temperature of water and flammability of alcohol [Latz et al. (2012)]. In those blends, Water Ethanol is quite promising and is compliant with vehicle integration where both pure fluids are not. In comparison with stationary plant, where the system is designed to run at its nominal point, the vehicle integration has to face a second challenge such as the limited cooling capacity and the highly transient behaviour of the heat sources. To deal with that, an effective control strategy is really important to maximize power recuperation and ensure a safe operation of the system. Although many papers concerning Rankine components and system optimization for mobile application can be found in the literature [Seher et al. (2012), Mavridou et al. (2010)], only few of them deal with control development and operating strategy improvement [Peralez et al. (2013), Willems et al. (2012)]. One key variable to control is the working fluid temperature at the evaporator outlet / expander inlet since this temperature has a big impact on system performance [Quoilin et al. (2011)]. In a system perspective, it has to be as close as possible to the vapor saturation line to increase the system efficiency. An effective control of this temperature will allow to have longer recovery period by increasing the time where the expander is fed with vapour. This criterion can be achieved by reducing the standard deviation to the set point (SP). This paper is organized as follows. Section 2 presents the principle and the studied system. Section 3 approaches the modelling methodology and the resulting partial differential equation (PDE) system. Section 4 shows the two implemented controllers when section 5 compares their performances.

2. PRINCIPLES AND STUDIED SYSTEM

All the variables used in the following are explicitly defined in tables 2, 3 and 4 of the appendix.

2.1. Rankine Process

Rankine cycle is a widely used power generation cycle to turn heat into mechanical or electrical power. First the working fluid is pumped from a tank at the condensing pressure to the evaporator at the evaporating pressure. Then the pressurized working fluid is pre-heated, vaporized and superheated in one or several heat exchangers (HEX), also known as boilers. These HEX are linked to the heat source. The superheated vapor expands from evaporating pressure to

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condensing pressure in an expansion device converting the pressure and enthalpy drop into mechanical work. Finally the expanded vapour condenses through a condenser releasing heat into the heat sink (e.g. ambient air) and returns to the working fluid reservoir. In this process the changes of states in both the pump and the expander are irreversible and increase the fluid entropy to a certain extent.

2.2. Studied System

The Waste Heat Recovery System (WHRS) is compounded on a turbocharged 6 cylinder 11L 320kW HD engine using exhaust gas recirculation (EGR) and a selective catalyst reduction system (SCR) to reduce the NO_x emissions. The studied Rankine cycle is recovering heat from both EGR and Exhaust applying a serial configuration of two heat exchangers. Figure 1 shows a schematic of the studied system. The mass flow rate through the two boilers is controlled by the pump speed. The expansion machine is a turbine, which has a higher power density than volumetric expanders [Seher et al. (2012), Lemort et al. (2013)]. The working fluid is then condensed through an indirect condenser fed by coolant. Moreover the cycle is equipped with two bypass valves one located in the exhaust stream to control the amount of energy introduced in the system and a second in front of the expansion device to prevent liquid to enter in the turbine and avoid blade erosion caused by liquid droplets into a high speed rotor.

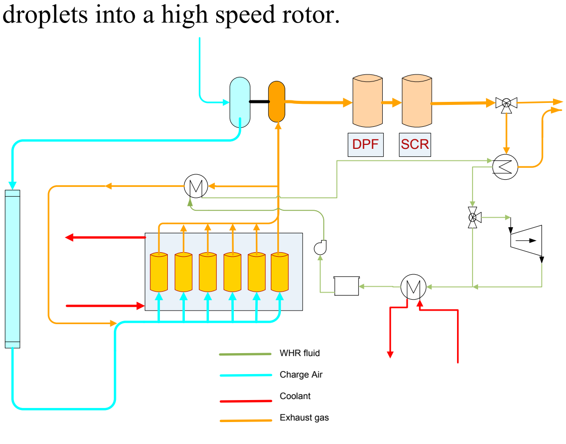

Fig 1 Studied system

The chosen working fluid is a predefined mixture of Ethanol and Water which gives the best compromise concerning performance and reduced flammability.

3. HEAT EXCHANGER CONTROL ORIENTED MODELING

The system dynamic is controlled by the HEX behaviour (i.e. evaporators and condenser) and models of these components are developed to dynamically predict temperature and enthalpy of transfer and working fluid at the outlet of each boiler. This is critical when coming to control design to ensure a safe operation and a proper operation of the EGR function. Safe operation means that the fluid is completely vaporized when entering the turbine in order not to destroy it. For the EGR, the aim is to have low gas temperature at the outlet of the boiler, in order not to disturb the emissions control strategy and the internal combustion process.

3.1. Model Assumptions

Several assumptions have been done to simplify the problem in a great extent. They are usually admitted when coming to heat exchanger modelling [Feru et al. (2013), Vaja (2009)]:

- The transfer fluid is always considered in single phase i.e. no condensation in the EGR/exhaust gases is taken into account.
- The conductive heat fluxes are neglected since the predominant phenomenon is the convection.
- The pressure drops on each fluids side (transfer and working fluids) are not considered.
- Both boilers are represented by a straight pipe in pipe counterflow heat exchanger, similarly to Vaja (2009), divided into n lumped sub-volumes in the longitudinal direction.
- Fluid properties are evaluated at the outlet of each sub-volume i.e. linear profile is considered between inlet and outlet of each node.
- Pressure dynamics is neglected since its time scale is very small considered to the HEX time scale.
- Working fluid mass flow rate is supposed constant along the heat exchanger.

3.2. Governing Equations

Since the mass is assumed constant the continuity equation is neglected and the model is only based on the energy conservation for the working fluid *(1)*, the gas *(2)* and the energy balance at the internal *(3)* and external wall *(4)*.

$$\frac{\partial \dot{m}_{wf} h_{wf}}{\partial z} - \dot{Q}_{\mathrm{conv\,wf\,int}} = \rho_{wf} V_{wf} \frac{\partial h_{wf}}{\partial t}. \tag{1}$$

$$\frac{\partial \dot{m}_g\, c_{p\,g}(T_g) T_g}{\partial z} - \dot{Q}_{\mathrm{conv\,g\,int}} - \dot{Q}_{\mathrm{conv\,g\,ext}} = \rho_g V_g\, c_{p\,g}(T_g) \frac{\partial T_g}{\partial t}. \tag{2}$$

$$\dot{Q}_{\mathrm{conv\,g\,int}} + \dot{Q}_{\mathrm{conv\,wf\,int}} = \rho_{\mathrm{wall}} V_{\mathrm{wall\,int}} c_{p_{\mathrm{wall}}} \frac{\partial T_{\mathrm{wall\,int}}}{\partial t}. \tag{3}$$

$$\dot{Q}_{\mathrm{conv\,g\,ext}} + \dot{Q}_{\mathrm{conv\,amb\,ext}} = \rho_{\mathrm{wall}} V_{\mathrm{wall\,ext}} c_{p_{\mathrm{wall}}} \frac{\partial T_{\mathrm{wall\,ext}}}{\partial t}, \tag{4}$$

with

$$\dot{Q}_{\mathrm{conv\,j\,k}} = \alpha_j A_{exch\,j\,k} (T_j - T_{wall\,k}), \tag{5}$$

where $j = g, wf, amb$ and $k = int, ext$.

Furthermore, to complete the system we need boundary and initial conditions. Time-dependent boundary conditions are used in z=0 and z=L:

$$\dot{m}_{\mathrm{wf}}(t,0) = \dot{m}_{\mathrm{wf}\,0}(t), \quad h_{wf}(t,0) = h_{wf\,0}(t),$$
$$\dot{m}_{\mathrm{g}}(t,L) = \dot{m}_{\mathrm{g}\,\mathrm{L}}(t), \quad T_g(t,L) = T_{g\,L}(t).$$

The initial conditions for the gas and wall temperatures and working fluid enthalpy are given by:

$$T_g(0,z) = T_{g\,init}(z), \quad T_{wall\,int}(0,z) = T_{wall\,int\,init}(z),$$
$$h_{wf}(0,z) = h_{wf\,init}(z). \quad T_{wall\,ext}(0,z) = T_{wall\,ext\,init}(z),$$

The manipulated variable (MV) is the working fluid mass flow $\dot{m}_{\mathrm{wf}\,0}(t)$, whereas the controlled variable (CV) is the working fluid enthalpy $h_{wf}(t,L)$. However the enthalpy is impossible to measure directly we have to use pressure and temperature measurement to compute it.

3.3. Heat Transfer

To model the convection from the transfer fluid to the pipe walls and from the internal pipe to the working fluid a heat transfer coefficient (α) is needed. The convection from a