HEAT STORAGE ORC SYSTEM FOR VEHICLE ICE EXHAUST HEAT RECOVERY

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ABSTRACT

A new Organic Rankine Cycle (ORC) system for vehicle internal combustion engine (ICE) waste heat recovery (WHR) is presented in this paper, which can effectively reduce the influence of exhaust gas fluctuation caused by vehicle driving cycle. An evaporator with heat storage material inside is used in this system to make it possible that the ORC expander working under more stable condition, and the conceptual scheme of the heat-storage ORC system is presented. The dynamic model is established, and a heavy diesel engine exhaust experimental data is used as heat source in the performance simulation. The heat resistance effect and the heat capacity effect are the main effects of heat storage ORC system, and the influence of these two effects are analyzed and discussed, meanwhile, the system designing criteria is presented. Compared with conventional system, the fluctuation of ORC evaporator outlet temperature can be decreased by 50 % and the system overall efficiency can be increased from 6 % to 7 %.

1. INTRODUCTION

Interest in WHR of vehicle ICE has grown dramatically recently, as CO2 emission regulations are becoming stricter. Around 60-70 % of the fuel energy is lost in the exhaust and coolant (Endo *et al*,2007), and many researchers recognize WHR is one of the most potential methods to improve ICE efficiency. Furthermore, the technologies for WHR of vehicle ICE can also be used to save energy in the industry.

The four main sources of waste heat in a vehicle are the exhaust, coolant, exhaust gas recirculation (EGR) cooler and charge air cooler (Teng *et al*, 2007). Exhaust has the highest temperature with the largest mass flow rate heat sources among them, so exhaust heat recovery (EHR) is the most attractive part of WHR. Researchers have proposed many methods for EHR, such as, Rankine Cycle, Organic Rankine Cycle (ORC), Brayton Cycle. By comparison, ORC system has the best performance (Bailey, 1985). Then a series of model and experimental researches of ORC system for vehicle WHR show that brake specific fuel consumption BSFC can be improved by up to 15-30% under the ideal working condition (Morgan *et al*, 1973; Poulin *et al*, 1984; Chammas & Clodic, 2005). On the other side, it means that thermal efficiency and net output power of the engine can be improved by 18-40%.

Selection of the working fluid and calculation of steady system performance used to be the primary focus of this research field (Hung *et al*, 1997; Sprouse &, Depcik, 2013; Roy *et al*, 2010). Recently, researchers paid more attention to the heat fluctuation caused by the vehicle varying working condition, because fluctuation could significantly influence the performance of ORC system in the application. The fluctuation not only can reduce the efficiency, but also can break the components of ORC system, especially when the expander is a turbine. If the turbine inlet working condition fluctuation is too large, it will work in low-efficient range at most of the time. What's worse, if the working fluid is unsaturated at the turbine inlet, the droplet will break the blade.

Using the active control is a method to deal with the influence of the heat fluctuation, and keep the system in a safe working condition range. In the last decade, some active control strategies have been researched (Quoilin *et al*, 2011; Horst *et al*, 2013; Pralez *et al*, 2013). Horst *et al* (2013) controls the ORC system by four actuators, which are the pump rotation speed, the expander speed and two valves. Although the active control method can make the system under a safety working condition, the fluctuation of the turbine inlet temperature and pressure can't be reduced. As a result, the turbine will work in the low-efficient range at most of the time.

This paper proposes a heat-storage ORC system to deal with the heat fluctuation instead of using complex active control system. In the HS evaporator, the heat of exhaust is absorbed by heat storage material firstly, and then the heat will be conducted from heat storage material to the working fluid. The heat fluctuation will be reduced by the heat storage material, therefore the ORC system can work in a relative steady condition. This heat-storage ORC system makes it feasible that using a fixed-speed motor to drive pump, at the same time the turbine will work around the design point with a high efficiency.

The description of this system is proposed in Section 2, where introduces the conceptual scheme of the new system as well as the key components. Then this work establishes the system dynamic model in Section 3. Later, the simulated performance of the new ORC system and discussions are shown in Section 4. Section 5 presents the conclusion and the summary of the whole work.

2. SYSTEM DESCRIPTION & METHODOLOGY

Figure 1 presents the conceptual scheme of the heat-storage ORC system. The heat-storage evaporator is the key component, which adds a layer of heat-storage material between the heat source and working fluid. Figure 2 shows the principle diagram of heat-storage evaporator. There are two homocentric pipes separating the heat storage material from exhaust and working fluid instead of the only one pipe in the conventional evaporator. The expander is a turbine expander connected with a high speed generator. The safety valve avoids the unsaturated vapor go into the turbine. The working process of this system is shown as follows.

Working fluid is pressed in the pump, and then it evaporates in the evaporator. After that, the working fluid will be expanded in the expander and generate electricity. At last, working fluid will be condensed back to the fluid reservoir. When the temperature of heat storage material becomes higher than the Start-Temperature, the working fluid pump start to work and the safety valve will be closed. When the engine stops working, the safety valve will be opened, as well as the pump is stopped. If the evaporator outlet temperature is lower than the Safety-Temperature, the safety valve will be opened, but the pump won't be stopped. Adding the heat storage material greatly reduces the times of opening and closing the safety which makes this system more feasible.

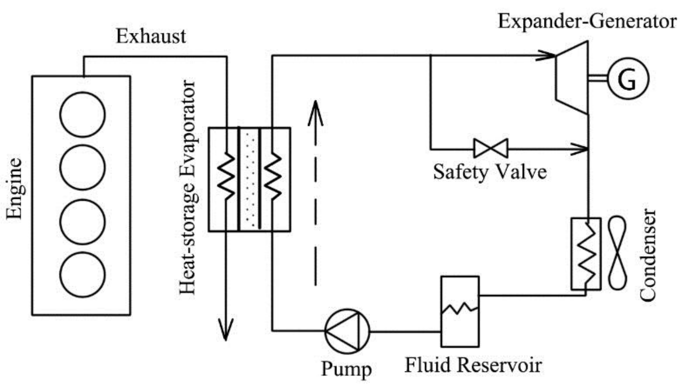

Figure 1: Conceptual scheme of heat-storage ORC system

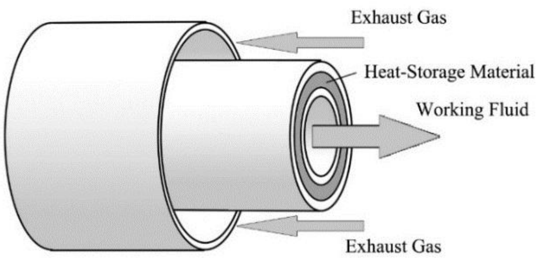

Figure 2: Principle diagram of heat-storage evaporator

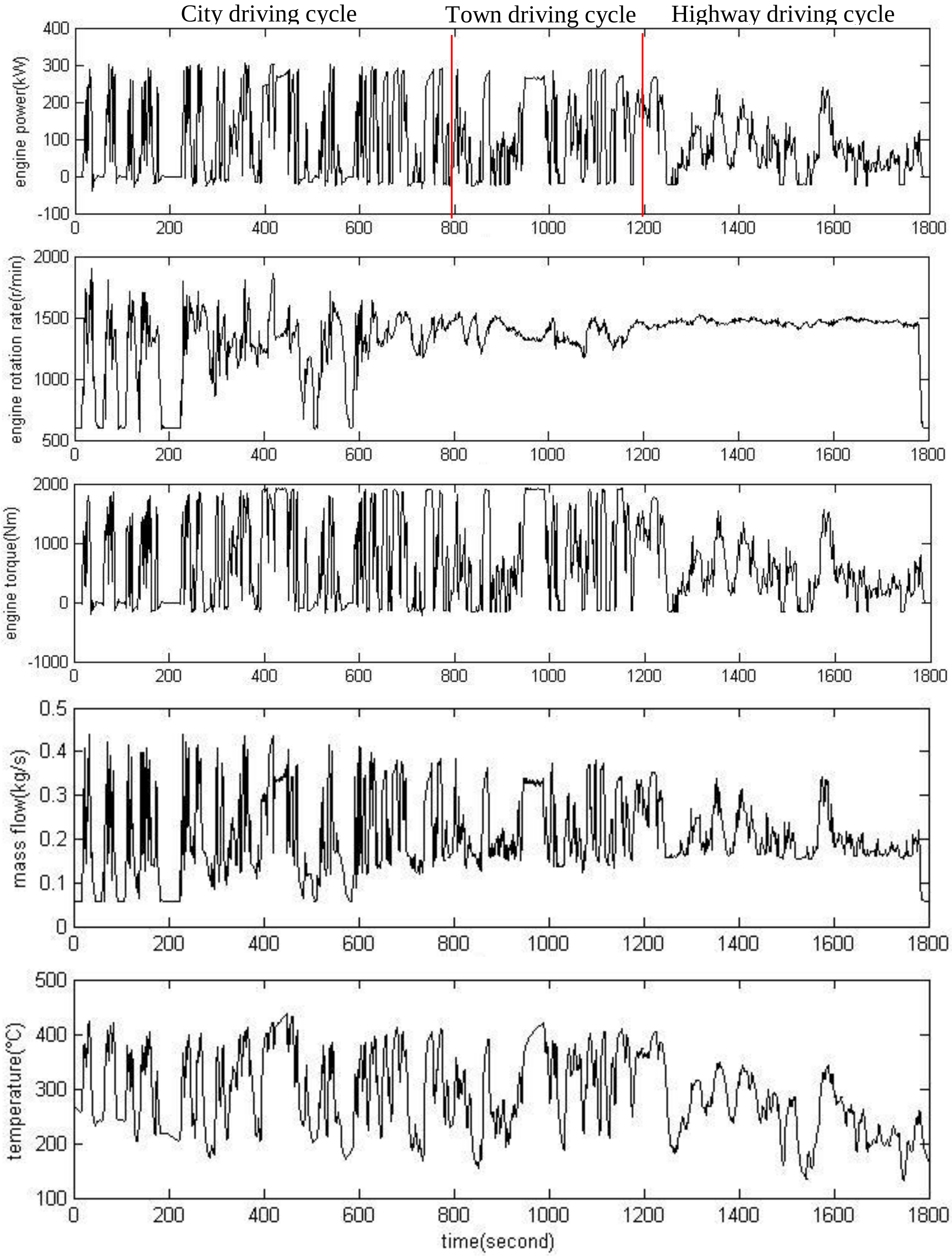

Figure 3: Vehicle diesel engine exhaust experimental data in ETC cycle

R245fa is selected as the working fluid of ORC system, as Malavolta *et al* (2010) researches that R245fa is a suitable working fluid to recover the heat in exhaust. This paper focuses on the influence of heat storage material, so the special research of working fluid selection is simplified.

Table 1: Reference properties of the heat storage material

Parameters	Value	Parameters	Value
Density	1667 kg/m^3	Thermal conductivity	0.8 W/(m*K)
Heat capacity	1.6 kJ/kg		

Table 2: Main parameters of the diesel engine

Parameters	Value	Parameters	Value
Numbers of cylinders	6	Rated power	300 kW
Displacement	12 L	Rated rotation speed	1900 r/min
Compression ratio	17	Maximum torque	1900 Nm

The selection of heat storage material is the key problem for this new ORC system. At the same time, there are hundreds of inorganic salts can be chosen, which include nitrite salts, nitrate salts, carbonate salts and so on. Herrmann U *et al* (2002) , Reilly HE& Kolb WJ (2001) and Zalba B *et al* (2003) have summarized the properties of the inorganic salts in their researches, it shows that the density is in the range of 1400-2200 kg/m^3 , the thermal conductivity is in the range of 0.5-2 W/(m*K) and the heat capacity is in the range of 1.5-1.8 kJ/kg. This work doesn’t select a special material, while the reference properties is selected in the reliable range, in Table 1, because the purpose of this work is researching the performance of the HS ORC system and presenting the criterion for the heat storage material selection. A special heat storage material will be selected in the future.

The waste heat recovery performance of the heat-storage ORC system for a heavy duty diesel engine operating under ETC driving cycle was investigated. The main parameters of the diesel engine is shown in Table 2. ETC cycle is the standard test for vehicle engine transient performance evaluation. It consists of three parts, city driving cycle, town driving cycle, and highway driving cycle, which include the typical working conditions of the engine in China. The engine rotation rate and torque must follow the text requirement, which is shown in Figure 3. It also shows the experimental data of the exhaust temperature and mass flow rate of the engine operating under ETC cycle. The heat source of the evaporator is determined by the exhaust experimental data.

3. DYNAMIC MODELING

3.1 Heat-storage Evaporator

Figure 4 shows the simplified model of the evaporator. Finite difference mothed is used in this model, and the evaporator is simplified as a 1D pipe with the constant shape of transverse section. The loss of pressure is ignored and the pump is working in a steady condition, so the pressure of exhaust and working fluid is assumed as a constant in the whole flow field. Both of the exhaust and working fluid are considered as inviscid fluid. According to these assumptions, the energy conservation equation (Eq. 1) is considered.

$$\frac{DU}{Dt} = \dot{Q} + \dot{W} \tag{1}$$

The simplified form based on the above assumptions is given as follows:

$$\int_v (\rho \frac{\partial H}{\partial t} - \rho \cdot u \frac{\partial H}{\partial x}) \mathrm{d}v = \int_A \dot{q} \cdot \mathrm{d}A \tag{2}$$

One order upwind scheme is used for discretization of exhaust and working fluid flow field:

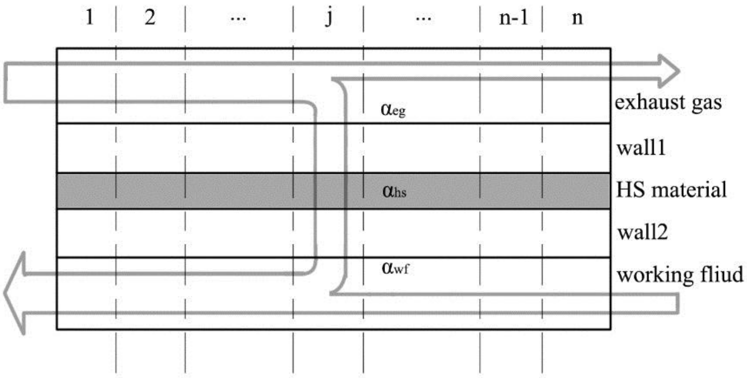

Figure 4: Discrete model of evaporator

Table 3: Main parameters of the HS-evaporator

Parameters	**Value**	**Parameters**	**Value**
Area	3 m^2	α_{exh}	1000 W/m^2K
V_{exh}	0.04 m^3	$\alpha_{wf,liquid}$	260 W/m^2K
V_{wf}	0.009 m^3	$\alpha_{wf,evapor}$	360 W/m^2K
M_{wall1}	10 kg	n	40
M_{wall2}	10 kg	M_{sto}	5 kg
Cv_{wall}	4600 J/kg	α_{sto}	800 W/m^2K

$$\overline{\rho_{exh,j}}V_{exh,i}\frac{\partial H_{exh,j}}{\partial t}=\dot{M}_{exh}\frac{H_{exh,j+1}-H_{exh,j}}{2}-A_i\dot{q}_{exh,j} \tag{3}$$

$$\overline{\rho_{wf,j}}V_{wf,i}\frac{\partial H_{wf,j}}{\partial t}=-\dot{M}_{wf,j}\frac{H_{wf,j}-H_{wf,j-1}}{2}+A_i\dot{q}_{wf,j} \tag{4}$$

The energy conservation equation of pipe wall and heat-storage material is shown as follows. The heat transfer between different respective elements is ignored.

$$M_{sto,i}\cdot\frac{\mathrm{d}E_{sto,j}}{\mathrm{d}t}=A_i(\dot{q}_{sto1,j}-\dot{q}_{sto2,j}) \tag{5}$$

$$Cv_{wall1}M_{wall1,i}\cdot\frac{\mathrm{d}T_{wall1,j}}{\mathrm{d}t}=A_i(\dot{q}_{exh,j}-\dot{q}_{sto1,j}) \tag{6}$$

$$Cv_{wall2}M_{wall2,i}\cdot\frac{\mathrm{d}T_{wall2,j}}{\mathrm{d}t}=A_i(\dot{q}_{sto2,j}-\dot{q}_{wf,j}) \tag{7}$$

The heat transfer coefficient is calculated by:

$$\dot{q}_{exh,j}=\alpha_{exh}(T_{exh,j}-T_{wall1,j}) \tag{8}$$

$$\dot{q}_{sto1,j}=\alpha_{sto}(T_{wall1,j}-T_{sto,j}) \tag{9}$$

$$\dot{q}_{sto2,j}=\alpha_{sto}(T_{sto,j}-T_{wall2,j}) \tag{10}$$

$$\dot{q}_{wf,j}=\alpha_{wf,j}(T_{wall2,j}-T_{wf,j}) \tag{11}$$

The heat transfer coefficient of exhaust side and heat storage material side is considered as a constant, while that of working fluid side depends on the state. Liquid and gas coefficients are constant respective (Quoilin *et al*, 2011), and the two-phase coefficient is calculated by Eq.12 (Horst *et al*, 2013).

$$\frac{\alpha_{wf,ev}}{\alpha_{wf,bub}} = \left((1-x)^{0.01} \left((1-x)^{1.5} + 1.9x^{0.6} \left(\frac{\rho_{wf,bub}}{\rho_{wf,dew}} \right)^{0.35} \right)^{-2.2} + x^{0.01} \left(\frac{\alpha_{wf,dew}}{\alpha_{wf,bub}} \left(1 + 8(1-x)^{0.7} \left(\frac{\rho_{wf,bub}}{\rho_{wf,dew}} \right)^{0.67} \right) \right)^{-2} \right)^{-0.5} \tag{12}$$

Other main parameters are shown in Table 3, the heat transfer coefficients of R245fa and exhaust are referenced to Quoilin *et al*,(2011).

Software REFPROP® is used to provide the properties of R245fa, and software Matlab@ is used for ORC modeling.

3.2 Other components

The turbine working condition is relatively steady, as the mass flow rate and pressure are constants. The inlet temperature is the only unsteady parameter which makes a limited difference in the turbine performance. The isentropic efficiency just changes less than 2% in the range of the temperature (350-450 K) calculated by the turbine design results of Fiaschi *et al* (2015). As a result, the turbine isentropic efficiency is considered as a constant: $\eta_{turbine} = 0.75$.

The pump of the heat-storage system is under a steady working condition, which means the pressure ratio and volume flow rate are constants. Therefore, the isentropic efficiency is considered as a constant: $\eta_{pump} = 0.7$.

The condenser is an air-cooled condenser installed in the front of the vehicle, so the cooling capacity is good enough to make sure the outlet temperature close to the ambient temperature. The ambient temperature in this work is 25 ℃, and the outlet temperature is set to 40 ℃.

4. PERFORMANCE SIMULATION & DISCUSSION

Several cases have been simulated for comparing the heat-storage ORC system with the conventional system. Two main effects are found, which are caused by adding the heat storage material, and they are the heat resistance effect and heat capacity effect. Heat resistance effect is bad for the system performance, while the heat capacity effect is favorable. The details are shown in this section.

4.1 Heat resistance effect

It is easy to comprehend that adding heat storage material in the evaporator pipe (Figure 2) will resist the heat transfer. In conventional system, thermal energy is conducted by the route: exhaust-pipe-working fluid, while in this system it is conducted by the route: exhaust-pipe1-heat storage material-pipe2-working fluid. Obviously there are two more heat transfer steps, which increase the overall heat resistance of the system. The heat transfer coefficient between pipe and heat-storage material is the key parameter of the heat resistance effect, which depends on its thickness and heat conductivity.

The heat resistance effect will reduce the heat transfer capability of the evaporator. A steady simulation is conducted and the influence of the heat transfer coefficient is shown in Figure 5, where the exhaust temperature is 330 ℃ and the exhaust mass flow rate is 0.15 kg/s. It can be found that the superheating temperature at evaporator outlet decreases as the heat transfer coefficient decreases. It also means that if we keep the identical superheating temperature, the mass flow rate will be smaller as the heat transfer coefficient decreases.

The heat resistance effect will reduce the heat fluctuation of the exhaust, which is shown in Figure 6. It is the result of the unsteady simulation, when the heat source of this system is 200 seconds among the ETC cycle which can represent the characters of the ICE exhaust. The result is calculated in two steps. A steady simulation is finished at first, and then it will be used as the initial condition of the unsteady simulation. In the steady simulation, the heat source is the mean value of the exhaust temperature and mass flow rate along the 200 seconds, and an iterative computation is done to make sure that the evaporator outlet superheating temperature is about 50 K, so the mass flow rates of the curves are different. In the unsteady simulation, the working fluid mass flow rate is a constant respectively, and the exhaust temperature and mass flow rate are unsteady. The evaporator outlet superheating temperature is shown in the Figure 6, as a result, heat fluctuation is reduced as the heat transfer coefficient going down.

The system performance depends on the heat transfer capability and the fluctuation of the evaporator outlet superheating temperature. A smaller the fluctuation means that the working fluid at the evaporator outlet has smaller possibilities to be unsaturated, as a result that the working fluid mass flow rate can be set large. If the fluctuation is large, the working fluid at the evaporator outlet will be unsaturated vapor some time, at that time, the safety valve will be opened to protect the turbine from droplet harming and the output power will be null. The heat transfer capability influences the working fluid mass flow rate straightly as is shown in Figure 5, which is discussed above.

Figure 7 shows the system performance changes as the working fluid mass flow rate increases. The vertical coordinate represents the overall efficiency, which is a calculation of net power output divided by the total thermal energy in the exhaust during that time, and an identical trend is found in the average net output power and the engine power improvement. When the mass flow rate is small, the working fluid at the evaporator outlet is saturated vapor during almost all the time, so the performance is better as the mass flow rate increases. While the mass flow rate is large, the unsaturated time is longer which makes the performance worse as the mass flow rate increases. As a result, there is a performance maximum in the middle mass flow rate. The curves represent the ORC system with different heat transfer coefficients, and it is shown that the maximum performance decreases as the heat transfer coefficient decreases. So the heat resistance effect is a bad effect for the system performance.

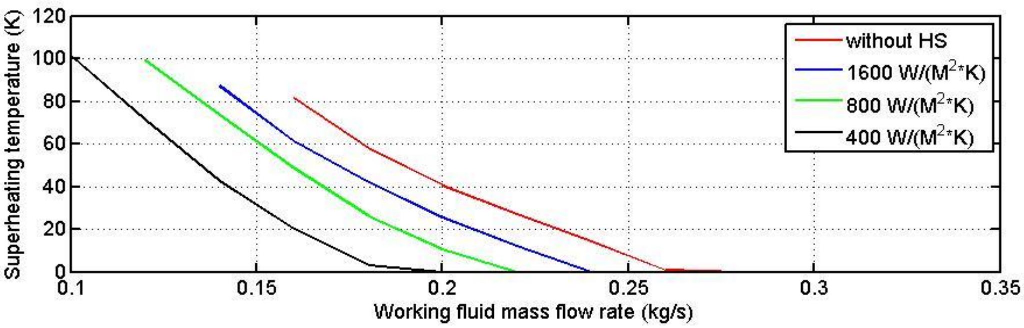

Figure 5: Superheating temperature at the evaporator outlet vs. working fluid mass flow rate

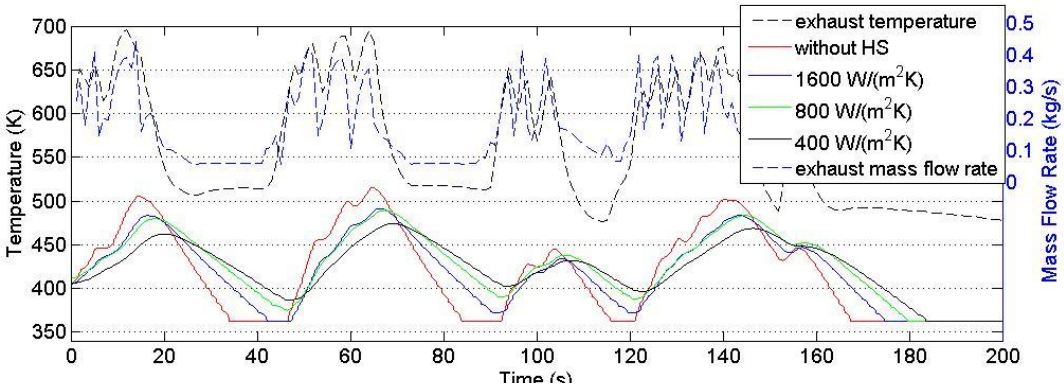

Figure 6: Fluctuation of the evaporator outlet temperature

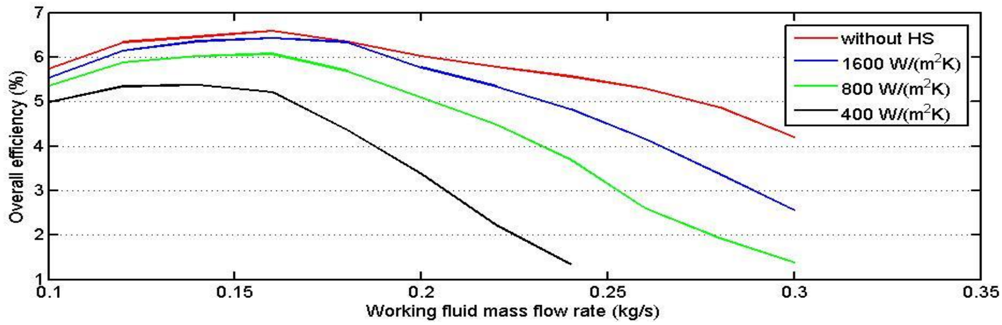

Figure 7: System overall efficiency vs. working fluid mass flow rate

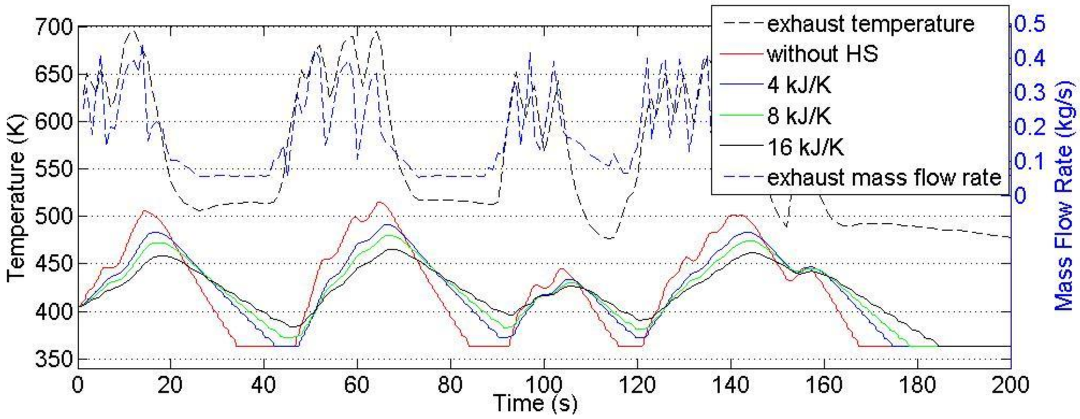

Figure 8: Fluctuation of the evaporator outlet temperature

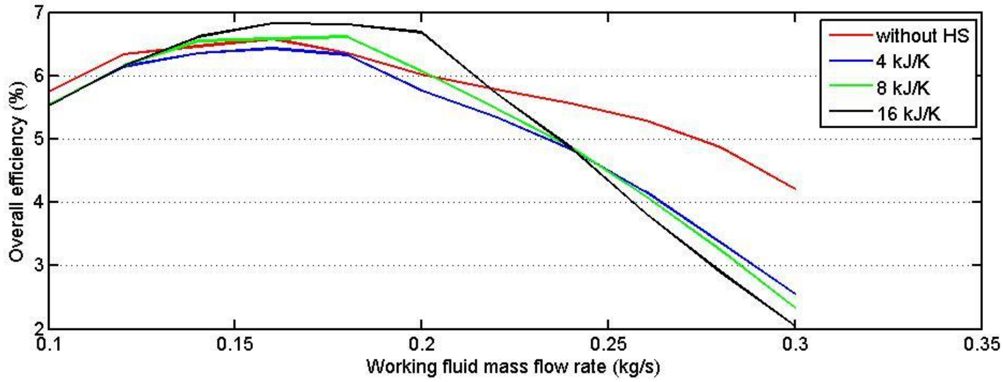

Figure 9: System overall efficiency vs. working fluid mass flow rate

4.2 Heat capacity effect

The heat capacity effect is produced by the heat capacity of the heat-storage material, while it is unrelated to the heat transfer coefficient, however, it also can reduce the fluctuation which is shown in Figure 8. In these cases, the heat transfer coefficient is set to 1600 W/(m^2K), and different curves represent the systems with different total heat capacities of the heat storage material. It shows that the fluctuation is reduced as the heat capacity decreases. It is worth noting that the heat capacity effect is in helpful to improve the system overall efficiency, which is shown in Figure 9. The overall efficiency can be improved from 6 % to 7 % in these calculation cases, and the larger total heat capacity is, the better the system performance is.

4.3 System performance discussion

The two effect's influence has been conducted in above sections, and the conclusion is valuable for selecting the heat storage material and designing the evaporator construction.

The heat transfer coefficient depends on the thickness and the heat conductivity of the heat storage material:

$$\alpha_{sto} = 2\frac{\lambda_{sto}}{h_{sto}} \tag{13}$$

The total heat capacity depends on the total volume and the volumetric heat capacity of the material:

$$C_{sto} = c_{v,sto} \cdot V_{sto} \tag{14}$$

The thickness and volume is the parameter of the evaporator construction, and the relationship between them is fixed if the heat transfer area is fixed:

$$A = \frac{V_{sto}}{h_{sto}} \tag{15}$$

It shows that there is a trade-off between the heat resistance effect and the heat capacity effect, which is related to the thickness of the heat storage material. It means that a good heat storage evaporator can be designed by optimizing the heat storage material thickness. At the same time, it should be noted that the thickness of the heat storage material shouldn't be too large, because it leads to the increasing of weight and volume of the evaporator, which are limited in the vehicle.

On the other hand, a conclusion can be drawn that the system performance will be improved when selecting a heat storage material with both of the larger heat conductivity and the larger volumetric heat capacity. Except the conventional heat storage material which store the heat by temperature rising, the phase change heat storage material and the chemical heat storage material have great potential in the heat-storage ORC system. M. Medrano *et al* (2010), A. Gil *et al* (2010) and M.M. Kenisarin (2010) summarized the phase change heat storage materials, their works showed that nearly any melting temperature can be realized in the range of 100-400 ℃ by adjusting the component of inorganic salt mixtures. If the phase change reacts along the temperature range instead of at a special temperature, it will be more valuable for this system.

5. CONCLUSIONS

The heat storage ORC system for the vehicle ICE EHR reduces the heat fluctuation by 50%, at the same time, it improves the overall efficiency of the ORC system. In the cases of this paper the overall efficiency can be increased from 6 % to 7 %. Furthermore, this system has the potential to reduce the heat fluctuation more and improve the system performance more by selecting a better heat storage material.

Two effects of the system are proposed and analyzed in this paper. Both of them have the ability to reduce the heat fluctuation, but the heat resistance effect is unfavorable for the system performance while the heat capacity effect is advantageous for it. If the latter is greater than the former, the system efficiency will be improved. As a result, it is better to increase the heat transfer coefficient as well as the total heat capacity during designing the heat storage ORC system evaporator. At the same time, it should be noted that the thickness of the heat storage material shouldn't be too large because of the limitation of weight and space in the vehicle.

The selection of the heat storage material is important for the system performance, the material with larger heat conductivity and larger volumetric heat capacity are the better one. The latent heat storage material has the potential, among which the phase change heat storage material is much more feasible.

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