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Dynamic response comparison of direct and indirect evaporation options in ORC systems for waste heat recovery

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Abstract

ORC is a mature technology that can be used for Waste Heat Recovery (WHR) of Internal Combustion (IC) Engines. Direct Evaporation of the organic fluid from the hot exhaust is an interesting option compared to the often preferred intermediary thermal oil loop choice due to its thermal efficiency potential and reduction of system footprint and weight. However, concerns due to dynamic variability of the hot source still hinder its consideration. In this paper, a comparison of the dynamic response of ORC evaporators with both indirect and direct evaporation is performed, under fluctuations of an IC engine exhaust according to relevant frequencies and amplitudes of a standard driving cycle. The results show the range of frequencies and amplitudes of hot source fluctuations for which direct evaporation is most feasible and the range for which special consideration must be taken.

Keywords: Waste Heat Recovery; Organic Rankine Cycle; Direct Evaporation

1. Introduction

A lot of attention has been paid recently to the development of technologies that can enable the efficient conversion of energy resources, among them, Waste Heat Recovery (WHR). Internal Combustion (IC) engines represent an interesting opportunity for WHR because typically approximately 60% of the fuel energy is unused as waste heat, half

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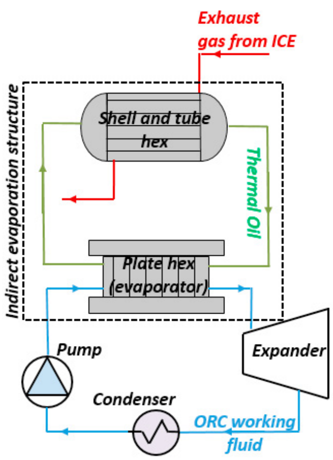
of which exits through the exhaust at relatively high temperatures of around 400 °C [1]. Organic Rankine Cycle (ORC) is one of the technologies that have been considered for the exploitation of waste heat due to its maturity, simplicity, flexibility and reliability [2]. ORC is a power cycle similar to the conventional Steam Rankine Cycle that utilizes an organic fluid that vaporizes at lower temperature than steam/water. The heat is transferred to the cycle via the evaporator where the organic fluid is vaporized. Due to its lower dynamics compared to the expander, the evaporator and the rest of the heat exchangers are the ORC components that dictate the dynamics of the whole system [3].

In engine waste heat recovery with ORC, the heat transfer to the ORC evaporator can be done directly from the exhaust gases [4] or indirectly via an intermediary thermal oil loop [5]. The advantages of the thermal oil loop is that it acts as a thermal buffer when high variability of the heat source is present such as in mobile applications. As a power and temperature stabilizer, it protects the working fluid from high temperatures that can lead to the chemical decomposition of organic fluids. This reduction in temperature however reduces the potential thermal efficiency of the system by bringing exergy destruction due to two heat transfer processes instead of one [6]. Another drawback of the intermediary thermal oil loop is the additional volume and weight of the oil and extra heat exchanger required which can hinder the feasibility of the system in volume or weight restricted applications such as the mobile ones. Direct evaporation from engine exhaust gases is an attractive option because it has the potentiality of decreasing the mass and footprint of the system which is critical in mobile applications, as well as increase the thermal efficiency by reducing the exergy losses due to heat transfer. However, the highest challenge to its implementation is the dynamic behavior during high variability of the exhaust thermal power.

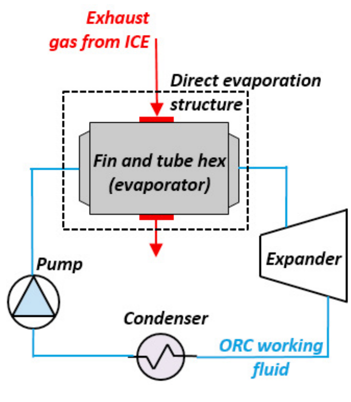
In this paper, a comparison of the dynamic behavior of ORC evaporator systems recovering heat from an IC engine is performed for the cases of indirect evaporation using a thermal oil loop and direct evaporation using two evaporators with different thermal inertias.

2. Systems description and models

In order to compare an ORC system suitable for the recovery of IC engine exhaust waste heat, an ORC system from the literature [5] that includes an intermediary thermal oil loop and R245fa as working fluid is considered. The exhaust heat is transferred to the oil in a shell and tube heat exchanger, while the oil transfers heat to the working fluid in a plate heat exchanger. This system is compared to a two direct evaporation options, using a single fin and tube heat exchanger with R245fa as working fluid as well. The two different systems are illustrated in Figure 1.



(a)



(b)

Fig,.1. a) Layout of ORC-WJHR system with indirect evaporation arrangement b) Layout of ORC-WHR system with direct evaporation arrangement

One of the direct evaporators geometry is chosen according to the standard methodology of heat exchanger design. This is called Direct Evaporator A. A second, alternative geometry selection for the heat exchanger is also proposed, signifying an extreme case of an evaporator with a large thermal inertia. This is called Direct Evaporator B. This geometry is selected as to maximize the volume of working fluid present and according to the methodology presented in a previous research by the authors [7]. The geometry of the heat exchangers is summarized in Table 1. The table also shows the weight and volume of the heat exchangers. It is to be highlighted that the direct evaporation arrangements A and B represent a reduction to 3% and 9% respectively of the heat exchangers weight and 11% and 31% of their volume considering the sum of both heat exchangers required for the indirect arrangement.

Table 1. Geometry and properties of heat exchangers considered. Direct Evaporator B corresponds to a high thermal inertia geometry

Geometric parameter	Indirect Oil HEX Shell and Tube	Indirect Evap. Plate	Direct Evap. A Fin and Tube	Direct Evap. B Fin and Tube
Number of tubes/plates [-]	400	40	16	18
Length of tubes/plates [m]	0.85	0.7	0.63	0.675
Tube inner diameter [m]	0.02	N/A	0.02	0.06
Plate width [m]	N/A	0.1	N/A	N/A
Wall thickness [mm]	2.5	1	2.5	2.5
Weight [kg]	468.65	38.11	14.07	47.12
Volume [m^3]	0.544	0.006	0.060	0.170

In order to compare the dynamic behavior of the different systems, detailed dynamic models are developed in the modeling language Modelica based on existing models of the TIL library [8] and simulated in the commercial software Dymola. The simulations are carried out assuming no pressure losses, and the heat exchange process from exhaust to organic fluid in isolation with fixed boundary conditions at the inlet of the exhaust and working fluid heat exchanger. These boundary conditions considered are the same as the literature [5] and are summarized in Table 2.

Table 2. Boundary conditions ORC system

Mass flow working fluid [kg/s]	0.17
Inlet temperature working fluid [°C]	20
Pressure working fluid [bar]	14
Initial super-heating working fluid [°C]	35
Mass flow exhaust (base case) [kg/s]	0.25
Temperature exhaust (base case [°C]	380

3. Results of the dynamic behavior comparison

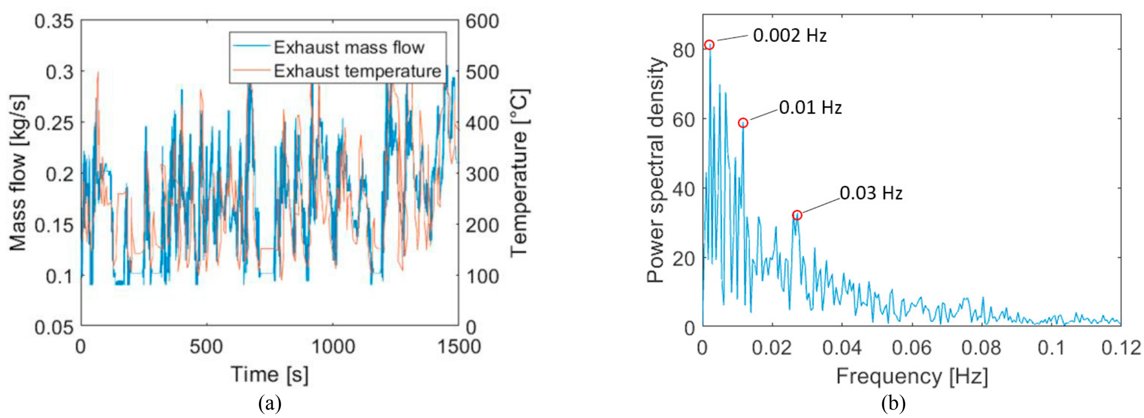


Fig. 2. a) Mass flow and temperature profile of the IC engine exhaust under the World Harmonized Driving Cycle from [9] b) Frequency

components of exhaust profile from Fast Fourier Transform spectral density

Fluctuations of the IC engine exhaust energy content in mobile applications depend on the particular driving conditions. The mass flow and temperature of the exhaust from an IC engine of a similar size are shown in Figure 2a, subjected to the World Harmonized Driving Cycle. From these graph, by means of Discrete Fourier Analysis, the principal frequency components can be extracted. This is shown in Figure 2b where some relevant frequency components of the mass flow fluctuations are highlighted. The temperature fluctuations, as seen in Figure 2a, show a similar pattern as the mass flow fluctuations.

3.1. Dynamic response comparison under relevant fluctuations and amplitudes

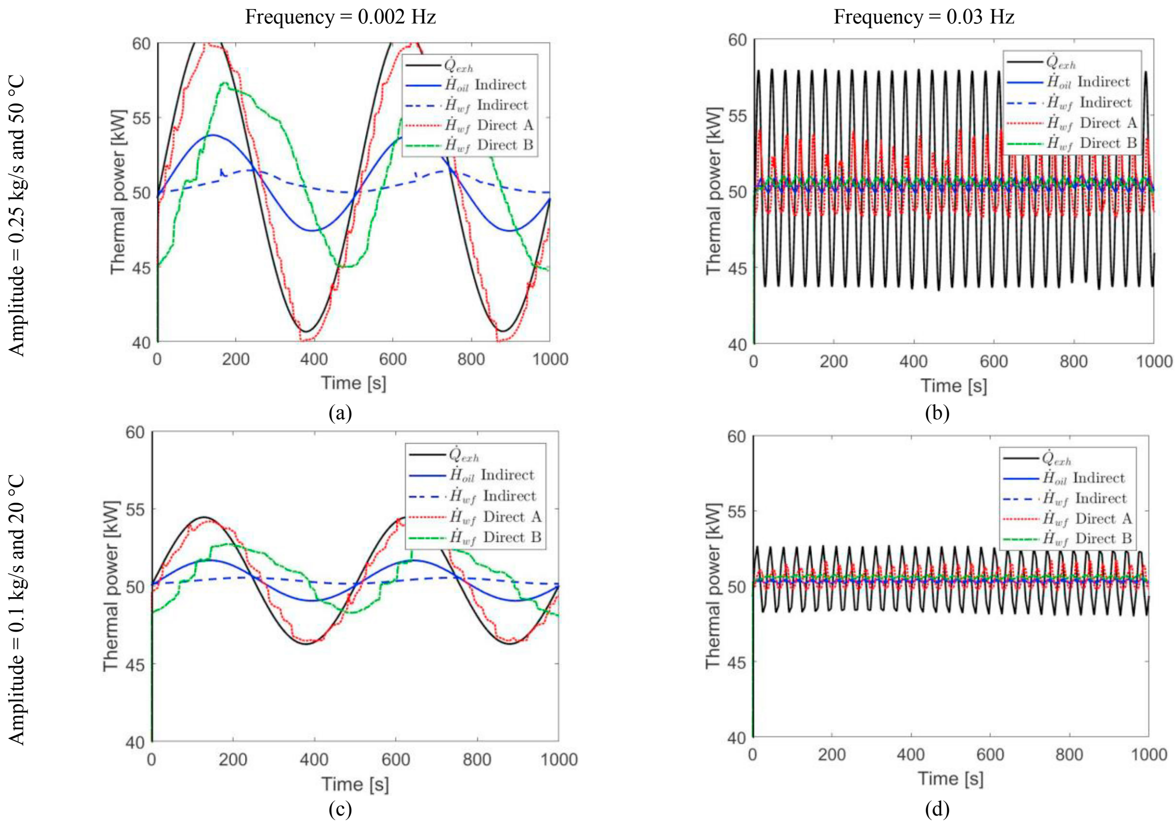


Fig. 3. Heat transferred from exhaust $\dot{Q}_{exh}$ and response of oil $\dot{H}_{oil}$ and working fluid enthalpy gain $\dot{H}_{wf}$ for two different frequencies and amplitudes of sinusoidal variation of the exhaust mass flow and temperature.

In order to study the dynamic response of the different ORC evaporation structures, the models were subjected to sinusoidal variations of the mass flow and temperature of the exhaust flow in the range of Figure 2. Figure 3 shows the dynamic response in the evaporator side for two different frequencies and amplitudes of fluctuation. Figures 3a and 3c represent a frequency of 0.002 Hz while Figures 3b and 3d a frequency of 0.03 Hz. Figures 3a and 3b are the response to an amplitude of 0.25 kg/s and °C and Figures 3c and 3d to an amplitude of 0.5 kg/s and 100 °C. The heat input from the exhaust, $\dot{Q}_{exh}$, and the enthalpy gain of the working fluid in the evaporator $\dot{H}_{wf}$ are defined as:

$$\dot{Q}_{exh}(t) = \dot{m}_{exh}(h_{exh,out} - h_{exh,in}) \quad (1)$$

$$\dot{H}_{wf}(t) = \dot{m}_{wf}(h_{wf,out} - h_{wf,in}) \quad (2)$$

Figure 4 shows the temperature of the working fluid at the outlet of the evaporator for the two different frequencies and an amplitude of 0.025 kg/s and 50 °C. It can be seen that for a frequency of 0.002 Hz the working fluid temperature at the outlet of the Direct Evaporator A falls at times below the saturation temperature which it is unacceptable for systems with turbo expanders. However direct evaporator B due to its high inertia still protects the expander from liquid droplets.

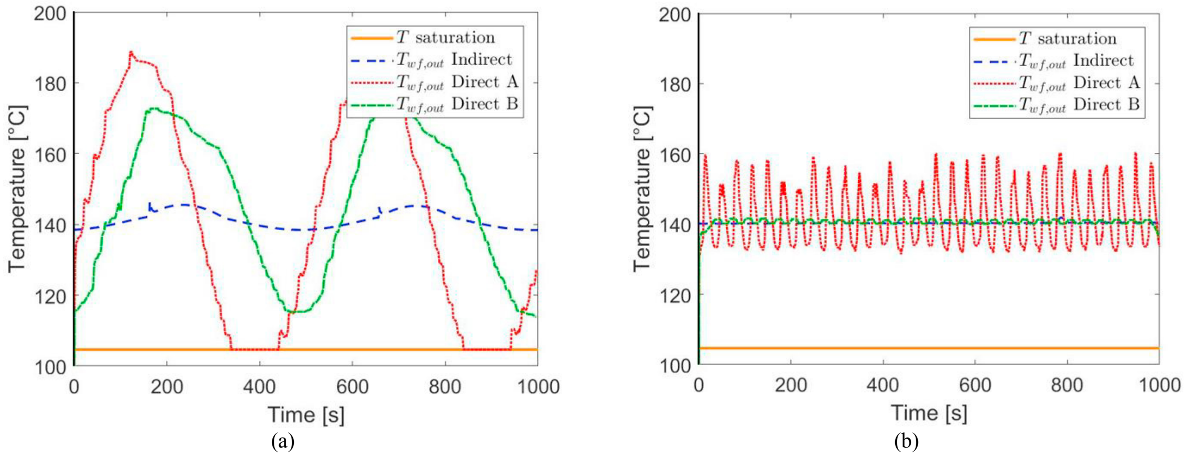


Fig. 4. Response of outlet temperature of working fluid $T_{wf,out}$ to fluctuations of exhaust mass flow and temperature for two different frequencies of sinusoidal variation. a) 0.002 Hz b) 0.03 Hz

3.2. Thermal power damping and amplitude ratio

As seen from Figure 3, the different ORC evaporation structures show different damping of the thermal power fluctuations present. The indirect evaporation with the thermal oil loop represents as expected the highest damping, while direct evaporation shows relatively much lesser thermal damping, though a high inertia direct evaporator can, to some extent, increase the thermal damping capability. To quantify this, an amplitude ratio AR of the fluctuations can be defined as:

$$AR = \frac{\max(\dot{H}_{wf}) - \min(\dot{H}_{wf})}{\max(\dot{Q}_{exh}) - \min(\dot{Q}_{exh})} \tag{3}$$

This amplitude ratio will depend on the amplitudes and frequencies of the heat source as well as the thermal inertia of the ORC evaporation structure. With the simplifying assumption of a constant, average, heat capacity Cp_{wf} of the working fluid in the vapor phase, a maximum amplitude ratio AR_{max} required can be defined as:

$$AR_{max} \leq \frac{\dot{m}_{wf} \cdot Cp_{wf} \cdot \Delta T_{max}}{\max(\dot{Q}_{exh}) - \min(\dot{Q}_{exh})} \tag{4}$$

The maximum temperature amplitude ΔT_{max} will depend on the saturation temperature as well as initial superheating degree of the working fluid at the outlet of the evaporator. Concerning the requisites that the fluid is fully vaporized at the outlet of the evaporator and also does not reach the decomposition temperature, ΔT_{max} is defined as:

$$\Delta T_{max} = \min[(T_{wf,out} - T_{wf,sat}), (T_{wf,decomp} - T_{wf,out})] \tag{5}$$

Figure 5a shows the variation of the amplitude ratio depending on the frequency of fluctuation for indirect and direct evaporation A and B according to average trends of the simulations. It also shows the AR_{max} required in the case of two different amplitudes of fluctuation, 20 and 8 kW. This shows how for some frequencies of fluctuation a direct evaporator with high thermal inertia can be enough to operate within safe limits, and for which frequencies and amplitudes of fluctuation other measures such as the implementation of an indirect evaporation structure or a robust flow control scheme may be needed to ensure the operation within acceptable operating points. The superheating at the outlet of the evaporator also plays an important role, because the minimum amplitude ratio AR_{max} will depend on it. Figure 5b shows this dependence according to the amplitude of heat input from the exhaust.

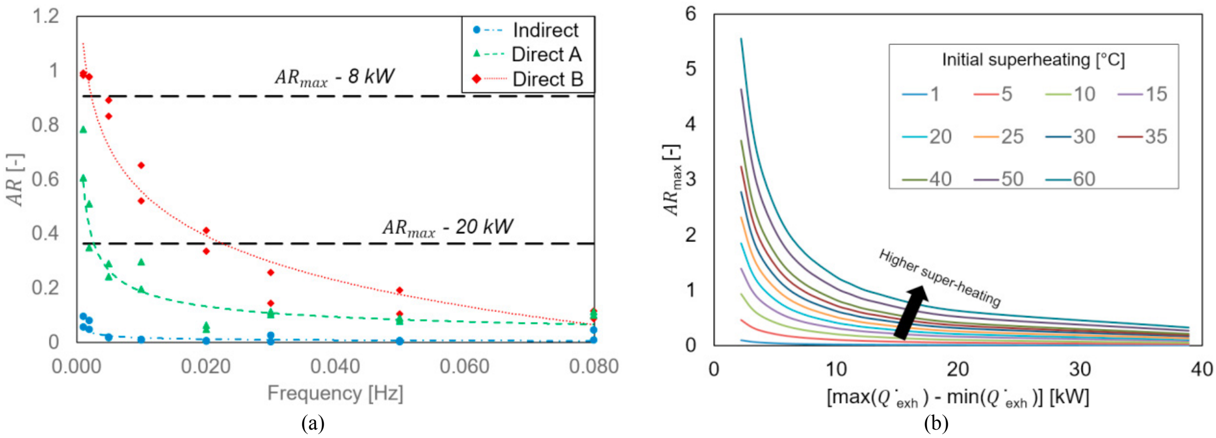


Fig. 5. a) Amplitude ratio of different evaporation structures according to different frequencies of exhaust fluctuation for an amplitude of 0.025 kg/s and 50 °C. b) Maximum amplitude ratio required as function of the amplitude of thermal power fluctuation for different values of initial super-heating

4. Conclusion

In this paper a comparison of the dynamic response between indirect and direct evaporation options for ORC has been performed. The systems were simulated under sinusoidal variation of an IC engine exhaust parameters according to relevant frequencies and amplitudes encountered in a standard driving cycle. The results show that, for this system and boundary conditions, for frequencies smaller than 0.03 Hz and amplitudes larger than 20 kW special measures such as a robust control must be taken when using a standard direct evaporator. However, with the use of high thermal inertia evaporator, this range can be reduced to frequencies smaller than 0.005 Hz considering the same amplitude. A high inertia direct evaporation arrangement would bring a dramatic reduction to 9% weight and 31% volume of the heat exchangers, as well as allow the potential to evaporate at higher temperatures and thus reduce heat transfer exergy losses and increase the system thermal efficiency.

Acknowledgements

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