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COMPARATIVE ANALYSES BETWEEN ORGANIC RANKINE CYCLE (ORC) AND ALTERNATIVES USING WET GASES AS LOW-GRADE HEAT SOURCES

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ABSTRACT

Low-grade heat from industrial exhaust gases can be recovered for electricity generation by various thermodynamic cycles. Previous studies assumed that only sensible heat is available for the heat recovery. However, in fact, the moisture content is commonly high and as the moisture condenses, a significant amount of latent heat is released. To gain further understanding of the effect of the moisture, we carried out comparative analyses among a basic organic Rankine cycle (ORC) and alternative cycles, including organic flash cycle (OFC), Kalina cycle (KCS) and transcritical ORC (T-ORC). Furthermore, the performance of an advanced dual-pressure ORC (DPORC) was evaluated under a wet gas scenario. The results show that by taking into account the moisture condensation heat, the net power output per kg/s of heat source flow increases significantly for all cycles examined although the exergy efficiency, known as an indicator for system irreversibility, tends to drop. Regardless of the level of moisture content in the heat source, ORC and T-ORC always have higher performance than OFC and KCS. In a case study using a moisture content of 0.1 in the heat source, a DPORC can achieve a performance enhancement by 58% compared with a simple ORC.

Keywords: *Kalina cycle (KCS), Organic flash cycle (OFC), Transcritical ORC (T-ORC), Dual-pressure ORC (DPORC), Heat recovery*

NONMENCLATURE

Abbreviations	
KCS	Kalina cycle system
OFC	Organic flash cycle
ORC	Organic Rankine cycle
T-ORC	Transcritical ORC
DPORC	Dual-pressure ORC
Symbols	
$\dot{E}_{ex}$	Exergy flow (kW)
$\dot{I}$	Exergy loss (kW)
$\dot{Q}$	Heat transfer rate (kW)
u	Moisture content (-)
$\dot{W}$	Work or power generation (kW)

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1. INTRODUCTION

Exhaust gases from various industrial processes (up to 200 to 250 °C) are widely regarded as potential low-grade heat sources for the conversion of waste heat into electricity [1]. Due to the incompatibility of conventional steam power cycle for low temperature applications, many other thermodynamic cycles have been studied to make the best use of the waste heat. Organic Rankine Cycle (ORC) [2] has received much attention due to its proven high performance in a wide range of working conditions, as well as its simple configuration, straightforward maintenance and feasibility of system scale-down [3]. Limitations of ORC have also been noticed, for example, the pinching problem in the heat transfer process, which can introduce significant irreversibility [4]. Other thermodynamic cycles have therefore been proposed, such as organic flash cycle (OFC) [5], Kalina cycle system (KCS) [5] and transcritical ORC (T-ORC) [6].

In previous studies, it is generally assumed that only sensible heat is available for recovery. However, industrial exhaust gases commonly contain large amount of moisture in the vapor form in various processes, such as flashing, washing, cleaning and drying. As the water vapor eventually condenses into liquid, considerable latent heat is released. In the present study, we conducted comparative performance analyses between ORC and other alternative cycles, such as OFC, KCS, T-ORC and DPORC, taking into account the presence of moisture in the heat source.

2. METHODOLOGY

2.1 Characteristics of Wet Gases

Hot humid air is a good representative of wet heat sources in various industrial processes. Initially, as heat is released, the heat source temperature has a linear reduction, like a standard dry heat source. Once the dew point is reached, however, the temperature profile starts turning flat, indicating that the temperature reduction is no longer proportional to the heat release, as shown in Fig. 1. It is evident that even at a relatively low moisture content, e.g. u = 0.1, the characteristics of the heat source already differs greatly from a dry scenario, showing a significant increase in available heat and only moderate change in temperature due to moisture condensation.

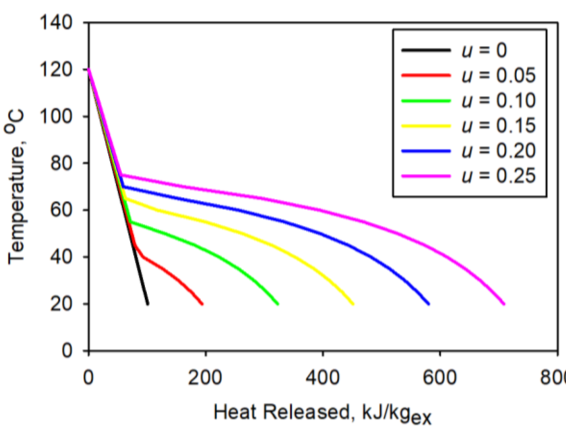

Fig. 1. Temperature profiles of a wet heat source at different moisture contents and a heat source temperature of 120 °C

2.2 Modelling

The ORC, OFC, T-ORC, KCS and DPORC systems are presented in Fig. 2. R245fa (1,1,1,3,3-penta-fluoro-propane) is selected as the base working fluid and has been used in ORC, OFC and DPORC systems. A T-ORC has the same system configuration as the basic ORC although the working fluid develops into a supercritical state in the evaporator. Carbon dioxide has been used for achieving transcritical operation with heat source at 120 °C. A KCS uses ammonia-water mixture pair as the working fluid to enable partial evaporation in the evaporator.

Table 1 lists the parameters used in the modelling. The following assumptions are made to simplify the model:

- All cycles operate in a steady-state condition;
- Heat and frictional losses in the systems are neglected; and
- Effects of the fluid kinetic and gravity are negligible.

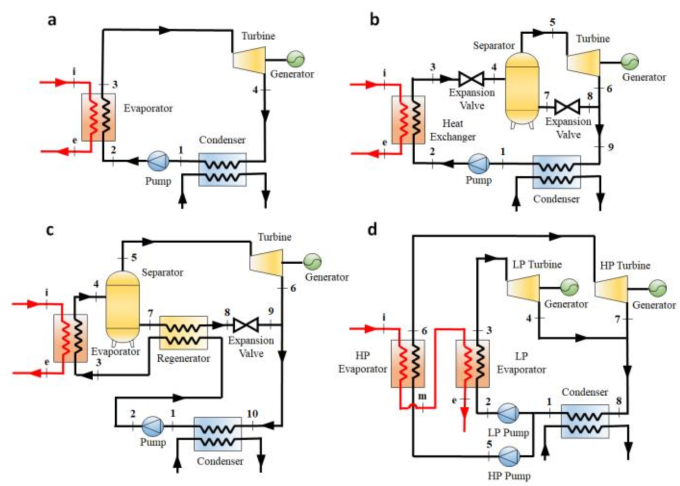

Fig. 2. System configurations of ORC **(a)**, T-ORC **(a)**, OFC **(b)**, KCS **(c)** and DPORC **(d)**

Table 1. Parameters used in the modelling analyses

Parameter	Value
Exhaust gas inlet temperature, $T_{ex,in}$ (°C)	120
Exhaust gas flow rate, $\dot{m}_{ex}$ (kg/s)	1
Exhaust gas moisture content, u (-)	0 to 0.25
Turbine isentropic efficiency, $\eta_{is,turbine}$ (%)	80%
Pump isentropic efficiency, $\eta_{is,pump}$ (%)	80%
Condensing temperature, T_{cond} (°C)	25
Pinch point temperature, ΔT_{pinch} (K)	8
Regenerator effectiveness, ε_{reg} (-)	0.9
Ambient pressure, p_o (atm)	1
Ambient temperature, T_o (°C)	20

Exergy loss of each system component has been examined according to the equations summarized in Table 2. The first law efficiency, or thermal efficiency of each cycle, is defined below:

$$\eta_{thm} = (\dot{W}_{turbine} - \dot{W}_{pump})/\dot{Q} \qquad (1)$$

where $\dot{Q}$ is the heat transfer rate in the evaporator, kW; $\dot{W}_{turbine}$ is the power generation of the turbine, kW; and $\dot{W}_{pump}$ is the power consumed by the pump, kW.

The second law efficiency, or exergy efficiency, is given by the following expression:

$$\eta_{exg} = \frac{\dot{E}_{ex,in} - \sum \dot{I} - \dot{E}_{ex,out}}{\dot{E}_{ex,in}} \quad (2)$$

where $\dot{E}_{ex,in}$ is the total available exergy flow in the heat source, kW; $\dot{E}_{ex,out}$ is the remaining exergy flow as the heat source stream is discharged from the system, kW; and $\dot{I}$ is the exergy loss of each component in the system, kW.

Table 2. Exergy loss expressions for different components

Component	Expression
Condenser	$\dot{I} = \dot{m}_h(e_{in,h} - e_{out,h})$
Expansion valve	$\dot{I} = \dot{m}T_0(s_{out} - s_{in})$
Evaporator/heat exchanger/regenerator	$\dot{I} = T_0[\dot{m}_h(s_{h,in} - s_{h,out}) + \dot{m}_c(s_{c,in} - s_{c,out})]$
Pump/turbine	$\dot{I} = \dot{m}T_0(s_{out} - s_{in})$

2.3 Model Verification and Optimization

Verification of the models has been conducted by comparing the computational results with data published in the literature [7-9]. The good agreements indicate high-level of reliability of the current models.

Optimization has been carried out for each cycle for obtaining the highest cycle performance. For basic ORC, the evaporation temperature is the only independent variable. For OFC, both heat exchanger outlet temperature and turbine inlet temperature are independent variables and required to be optimized. For KCS, the two independent variables are turbine inlet pressure and vapour quality at the evaporator outlet. For T-ORC, the independent variables are the turbine inlet pressure and temperature. For DPORC, the independent variables are the evaporation temperatures at the high-pressure and low-pressure evaporators.

3. RESULTS AND DISCUSSION

3.1 ORC

The net power output of the ORC has been evaluated at different moisture contents as shown in Fig. 3. With the presence of moisture, the net power output at lower evaporation temperature increases significantly. For u = 0, the highest power output is achieved at an evaporation temperature of 70 °C. For u > 0, the optimized evaporation temperature is far lower, indicating that for a wet heat source scenario, the turbine is required to operate at a much smaller pressure ratio to gain the highest net power output.

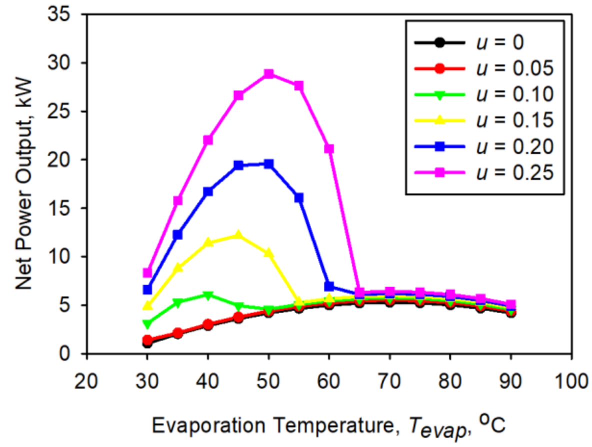

Fig. 3. Net power output of a basic ORC at different evaporation temperatures with the moisture content ranging from 0 to 0.25

The first law and second law efficiencies of ORC at different evaporation temperatures are presented in Fig. 4. The first law efficiency curves at different moisture contents overlap with each other since it is only a function of the evaporation temperature. The second law efficiency is affected by both the evaporation temperature and the moisture content. It tends to be lower as the moisture is present although the corresponding net power output is significantly higher.

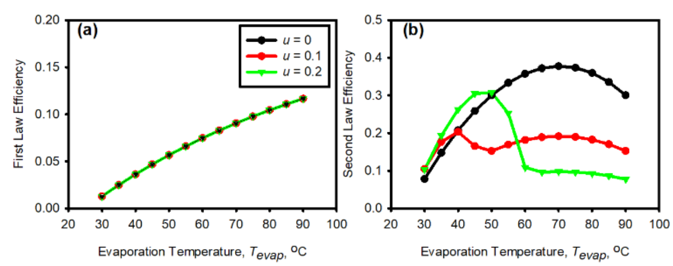

Fig. 4. First law **(a)** and second law **(b)** efficiencies of an ORC at different evaporation temperatures

3.2 Alternative Cycles

OFC, T-ORC and KCS have also been studied at both wet and dry heat source conditions. The performance of each cycle has been optimised and compared with the basic ORC. Figure 5 shows the cycle comparison in exergy loss for each component, as well as the net power output for wet and dry heat source scenarios.

At u = 0, ORC and T-ORC cycles have the highest performance. OFC has a significantly reduced evaporator loss and unused exergy (remaining exergy in the heat source flow discharged from the system); however, it creates a considerable loss in the expansion valve (EV), resulting in less net power generation compared with ORC and T-ORC. KCS reduces the evaporator loss through an improved temperature matching. In contrast, the use of a regenerator increases the liquid feeding temperature to the evaporator and reduces its ability for heat absorption.

At u = 0.2, T-ORC is still among the highest performing cycles, although the loss in the evaporator increases due to the non-linear heat source temperature profile, which has also caused the OFC to perform worse. The performance of KCS is not much affected by the moisture content; nevertheless, the heat source temperature level appears to be too low for KCS to perform.

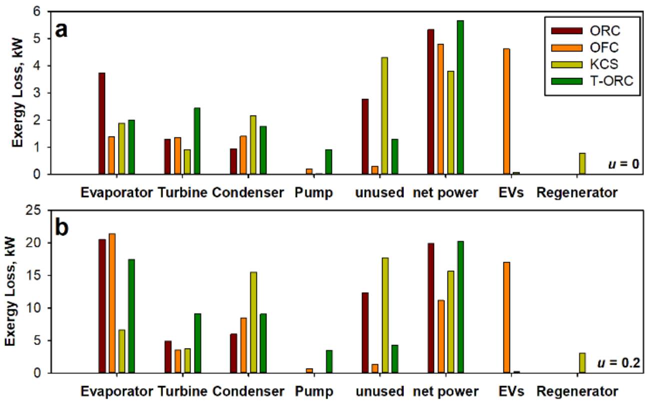

Fig. 5. Comparison of the component exergy for different cycles at dry **(a)** and wet **(b)** heat source conditions

3.3 DPORC

DPORC has been proposed as an advanced cycle for improved performance at wet heat source condition. Figure 6 shows the optimised net power output at different moisture contents, indicating that an enhancement as high as 58% can be achieved at u = 0.1. The gain from DPORC will, however, reduce as the moisture content is further increased or decreased.

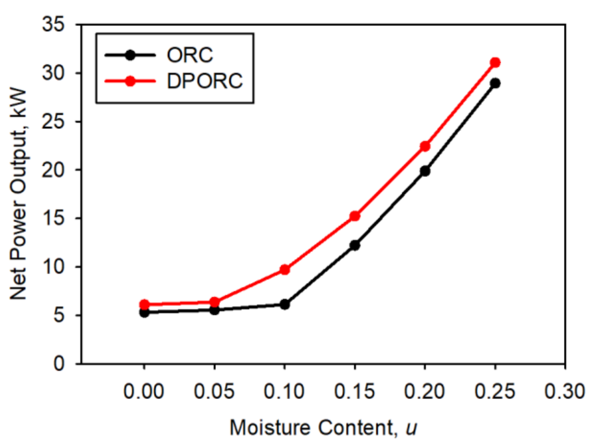

Fig. 6. Net power output of ORC and DPORC at different moisture contents

4. CONCLUSIONS

In summary, with the presence of moisture in the heat source, the ORC net power output increases significantly although the system exergy efficiency reduces. The system needs to operate at a much lower evaporation temperature to achieve the optimal performance. Moisture content has mostly negative or negligible effects on OFC, T-ORC and KCS, reducing their abilities to compete against ORC. An DPORC shows a performance enhancement as high as 58% compared with a basic ORC. The highest enhancement is obtained at u = 0.1 and further increase or decrease in u will reduce the gain.

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