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WET WORKING FLUIDS FOR REGENERATIVE ORC WITH VARYING HEAT SOURCE TEMPERATURE

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ABSTRACT

In the current work, the analysis of regenerative organic Rankine cycle (ORC) using wet fluids is performed. For the purpose to convert waste energy to useful work from three different heat sources, eleven wet organic fluids were selected divided into three groups according to their critical temperatures ranging from 66°C to 132°C. Results show that the working fluids with the lowest critical temperature to heat source inlet temperature ratio will generate the maximal net power output, namely R125, R1270 and R152a. Superheating affects the system performance negatively, since the net power output decreases as the superheat increases.

Keywords: Organic Rankine cycle, wet fluid, regenerative, waste heat recovery.

1. INTRODUCTION

The harmful effect of fossil fuels on the environment urges to develop effective and clean energy technologies. As it is stated in [1] nearly 20–50% of industrial energy usage is exhausted in the form of flue gasses. Recently, organic Rankine cycles (ORCs) have been used to recover waste heat and convert it into useful work to generate electricity. ORC attracts more attention due to its simplicity and reliability and have been thoroughly investigated in the literature, for example in [2-4]. The performance of ORCs is affected by heat source conditions, working fluid selection, heat sink conditions, cycle configuration and components, and process variables. According to [5], selecting an appropriate working fluid is not an easy task because the variation of heat source temperature ranges and the wide number of substances that can be used as working fluids. In general, there is three types of working fluid: a dry fluid, a wet fluid and an isentropic fluid. Hung et al.[1] assumed that isentropic fluids are most appropriate for recovering low-grade waste heat. Another research made by Hung et al. [6] , reported that wet fluids with very steep saturated vapor curves have a better performance in energy conversion efficiencies than dry fluids. It reported that the recuperative basic ORC performs better than the non-recuperative regenerative ORCs from both energetic and economic aspects but the recuperative regenerative ORCs are better than the recuperative basic ORC from thermodynamic point of view [7]. It is not possible to efficiently implement an internal heat exchanger to regenerative ORCs since the working fluid at the expander's outlet could be in the two-phase region. Li Gang [8] investigate in detail the performance of baseline ORC, reheat ORC, ORC with internal heat exchanger and a regenerative ORC, but he selected only dry or isentropic fluids. The present work aims to examine the performance of wet fluid and discuss the feasibility of them in the ORC, since there exists not enough information about them. To achieve this goal, eleven organic fluids have been adopted and the effect of superheat degree has been investigated.

2. ANALYSIS OF REGENERATIVE ORC SYSTEM

2.1 System description

The basic components of a regenerative ORC system consist of two pumps, evaporator, turbine, feed heater and condenser. The system layout of the regenerative ORC is shown in Fig.1.

Vapor enters the turbine at the evaporator pressure and expands to an intermediate pressure. Some vapor is

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extracted and sent to the feed heater, while the remaining vapor continues to expand to the condenser pressure. The vapor leaves the condenser as a saturated liquid at the condenser pressure. The working fluid enters the pump, where it is compressed to the feed heater pressure and is directed to the feed heater, where it mixes with the vapor extracted from the turbine. The working fluid leaves the heater as a saturated liquid. The working fluid compressed again to the turbine inlet pressure by second pump.

Three case studies are analyzed in the present work where the hot source temperatures are 90°C, 120°C and 150°C with mass flowrate of 14 kg/s, with liquid water at 20°C as cooling medium.

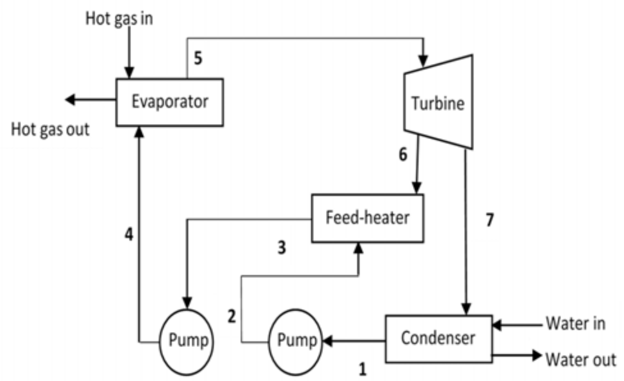

Fig 1 Schematic diagram of regenerative ORC.

The following assumptions are employed in the analyses of the regenerative ORC system:

a) The system is under steady-state.
b) Pressure drops are neglected through evaporator, condenser and pipelines.
c) The intermediate pressure (the vapor extraction pressure) is $P_{\text{int}} = P_{con} + \frac{1}{3}(P_{eva} - P_{con})$ [8].
d) The parameters of operating conditions are listed in Table 1.

The mass balance of each control volume can be expressed as:

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \tag{1}$$

The energy balance equation can be express as:

$$\sum_i E_i + \dot{Q} = \sum E_0 + \dot{W} \tag{2}$$

The fraction of the flow rate that flows into the feed-heater is given by

$$X_1 = \frac{h_3 - h_2}{h_6 - h_2} \tag{3}$$

The work consumed by both pumps

$$W_{P1} = m_{wf}\,(1 - X_1)(h_2 - h_1) \tag{4}$$

$$W_{P2} = m_{wf}\,(h_4 - h_3) \tag{5}$$

The turbine power is calculated by

$$W_{Tur} = m_{wf}\left[(h_5 - h_6) + (1 - X_1)(h_6 - h_7)\right] \tag{6}$$

The net output power of the system is defined as:

$$W_{net} = W_{Tur} - W_{P1} - W_{P2} \tag{7}$$

The heat absorbed by the working fluid in the evaporator is given by:

$$Q_{in} = m_{wf}\,(h_5 - h_4) \tag{8}$$

Table 1 Main input parameters for the regenerative ORC system

Heat source temperature (°C)	90/120/150
Heat source mass flow rate (kg s^{-1})	14
Cp of the hot gas (kJ kg^{-1} K^{-1})	1.1
Degree of superheat (°C)	5
Temperature difference in evaporator (°C)	8
Temperature difference in condenser (°C)	5
Turbine isentropic efficiency	0.85
Pump isentropic efficiency	0.8
Condenser temperature (°C)	30
Environmental temperature (°C)	20

2.2 Working fluid selection

A good choice of working fluid has a significant impact on the improvement of cycle performance because the working fluid affects the system efficiency, the system size and stability. As demonstrated in Fig. 2 working fluids can be categorized into three groups (dry, isentropic and wet) according to the slope of the saturated vapor curve in the *T-s* diagram. Wet fluids need superheating before entering the expander to avoid the fluid from crossing the two-phase region during expansion, while isentropic and dry fluids do not need

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superheat, since they leave the expander as saturated or superheated vapor. The selected working fluids are presented in Table 2.

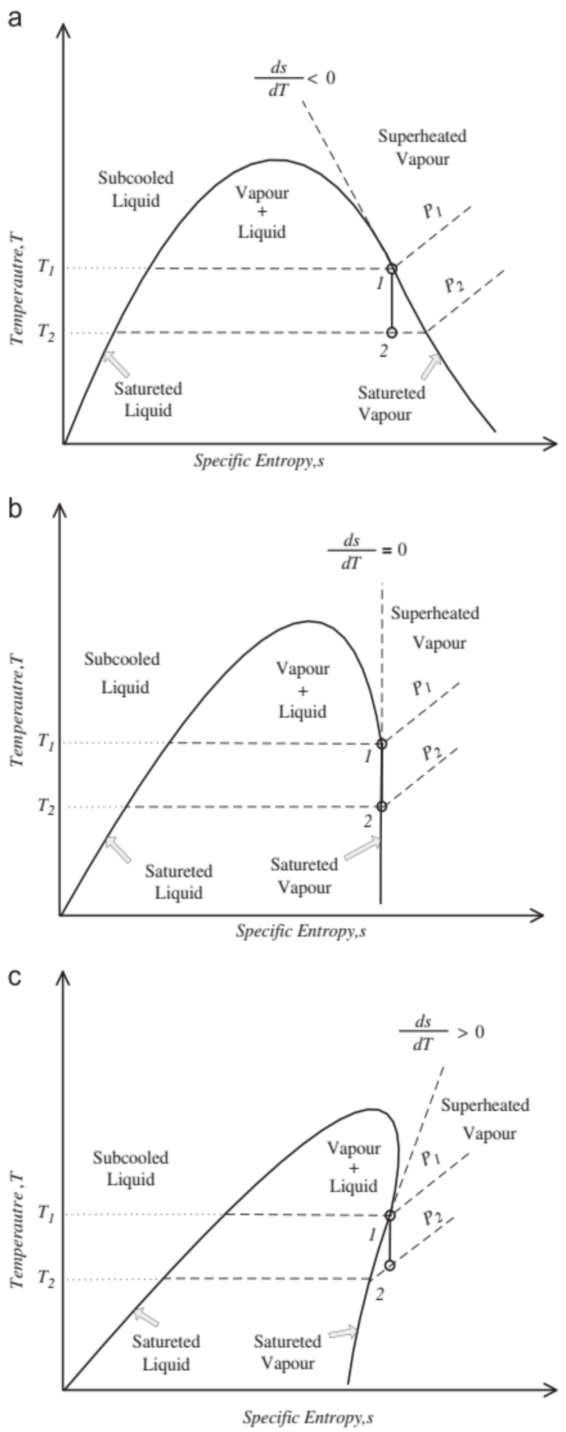

Fig 2 *T-s* diagram for fluids (a) wet, (b) isentropic and (c) dry [5].

Table 2 Properties of selected working fluids.

Working fluid	T_{hot}	T_c (°C)	$\frac{T_c}{T_{hot}}$	P_c (MPa)
R-125	90°C	66.02	0.73	3.62
R-143a		72.71	0.81	3.76
R-32		78.11	0.87	5.78
R-1270	120°C	92.42	0.77	4.66
R-22		96.15	0.80	4.99
R-290		96.68	0.81	4.25
R-134a		101.06	0.84	4.06
R-152a	150°C	113.26	0.75	4.52
Cyclopropane		125.15	0.83	5.58
Propyne		129.23	0.86	5.63
Ammonia		132.25	0.88	11.33

3. RESULT AND DISCUSSION

The investigated eleven working fluids are divided into three groups according to their critical temperatures ranging from 66.02°C to 132.25°C. Fig. 3 presents the variations of the net power output with the evaporating pressure for various heat source temperatures. For the heat source temperature of 90°C three working fluid are selected including R125, R143a and R32, net power output value increases and then decreases after reaching its maximum value except R125 which keep increases until it reaches its maximum value. When the heat source temperature is 120°C four working fluid are selected: R1270, R22, R290 and R134a. For R22, R290 and R134a the net power output is raising first then decreasing after attaining its maximum value except R1270 which keep increasing until it reaches its maximum value. Concerning the heat source temperature of 150°C, four working fluids are selected: cyclopropane, propyne and ammonia increases then decreases, while R152a keep increasing until they reach its maximum. Fluids R125, R1270 and R152a have the lowest value of T_C/T_{hot} in their groups, therefore it can be concluded that the working fluid with the lowest value of T_C/T_{hot} will produce the highest net power output.

Fig. 4 illustrates the variations of the net power output with the degree of superheat for various heat source temperature. The evaporating pressure is set constant corresponding to ($T_C - 10$). The net power output decreases with the increase of superheat degree for all working fluid except R32 which slightly decreases then increases. This can be explained by the following: as the degree of superheat increases the mass flow rate decreases while the enthalpy difference in expander increases but the decrease of mass flow rate is faster

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than the increase of enthalpy difference, thus the net power output decreases.

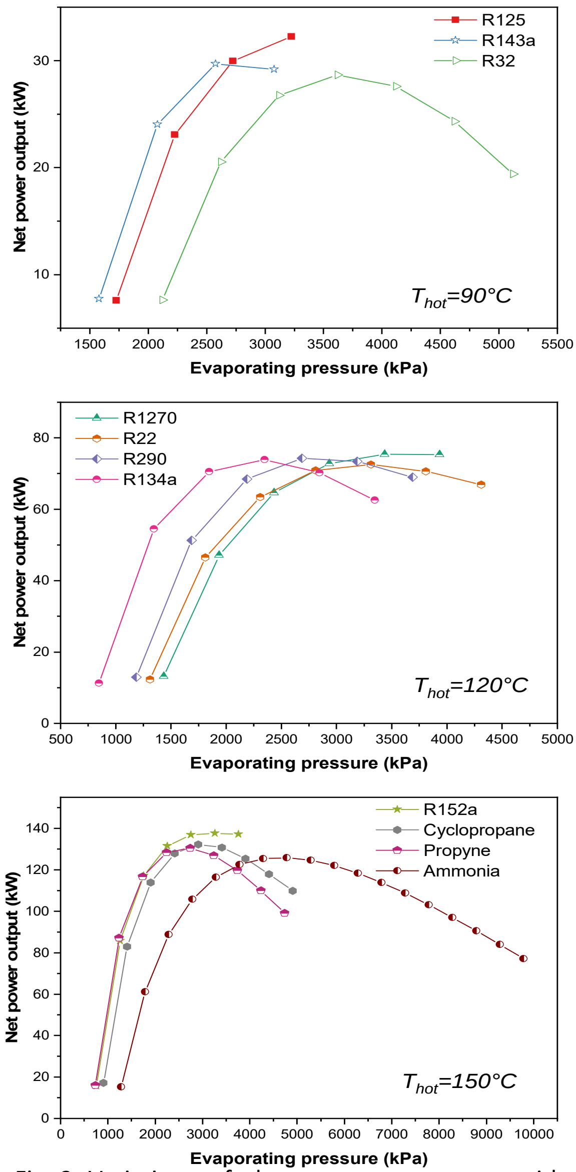

Fig 3 Variations of the net power output with evaporator pressure.

Fig. 5 presents the *T-s* diagrams for best fluids in each waste heat temperature case when the superheat degree was 5°C. Depending on the boundary conditions of the ORC, superheated vapor at the expander's outlet could be achieved with a wet fluid, this is the case for R125, while for R1270 and R152a fluids the expander's outlet occurs in the two-phase region, existence of liquid inside expander can damage expander blades and it also reduces the isentropic efficiency of the expander. This can be avoided by increasing the superheat degree.

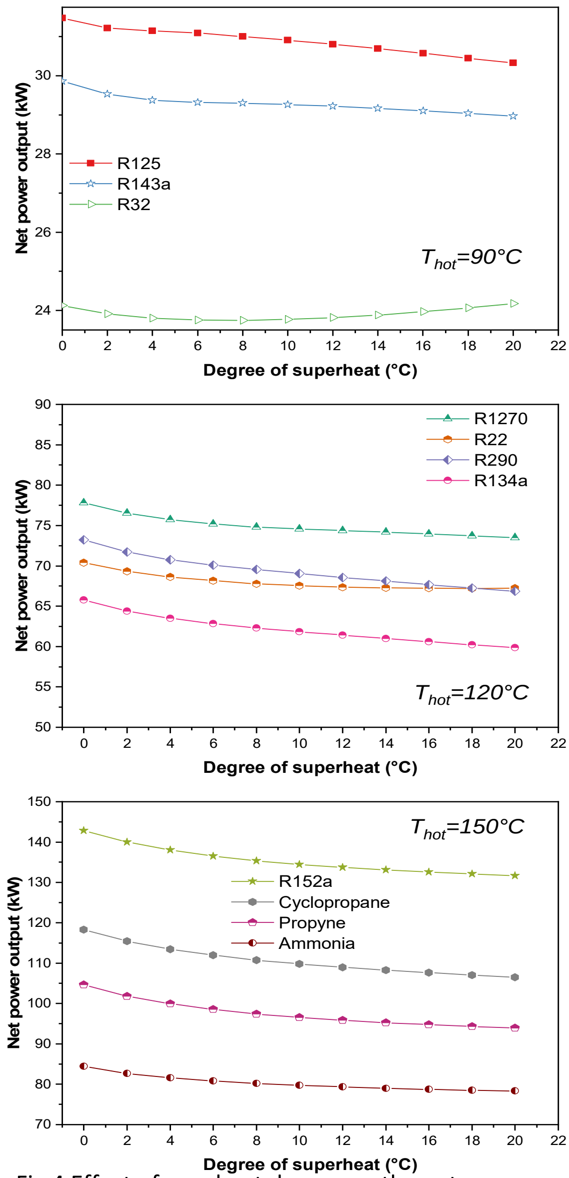

Fig 4 Effect of supeheat degree on the net power output.

4. CONCLUSION

The present paper investigates the performance of regenerative ORC using wet working fluids for low waste heat source temperature. the selected eleven working fluids are divided into three groups according to their critical temperatures ranging from 66.02°C to 132.25°C. As T_C / T_{hot} the evaporation temperature increases, the net power output increases than decreases with few exceptions (R125, R1270 and R152a).

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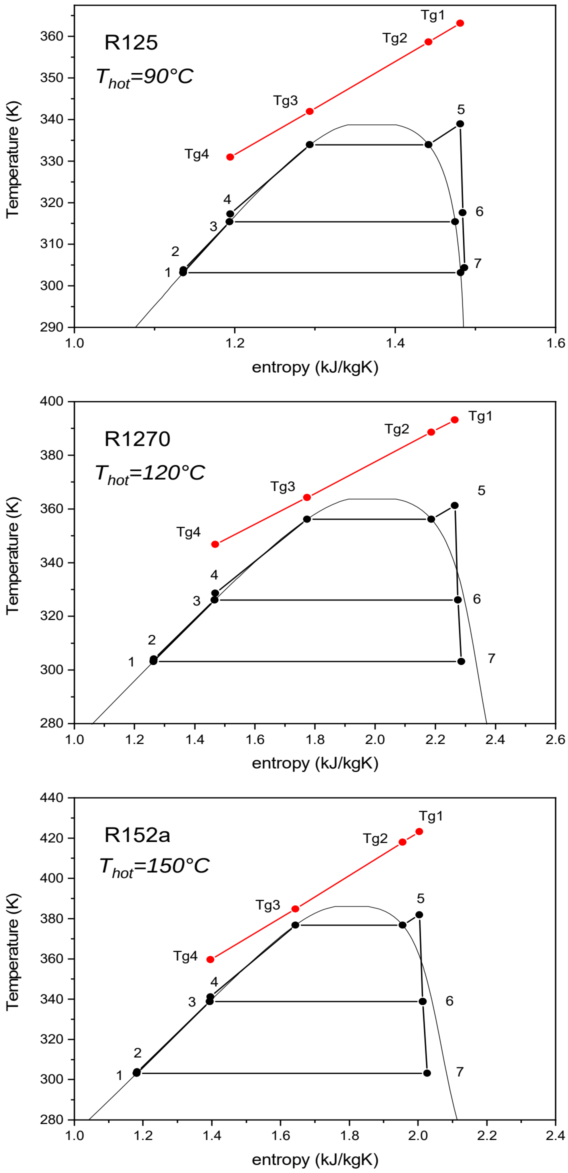

Fig 5 *T-s* diagrams for best fluids in each case.

The working fluid with lowest ratio will generate the maximal net power output. R125, R1270 and R152a showed to be the best working fluid in their temperature range. The superheat degree has a negative impact on the net power output. In addition, superheated vapor at the expander's outlet could be achieved with a wet fluid, this is the case for R125, while for R1270 and R152a fluids the expander's outlet occurs in the two-phase region. Existence of liquid inside the expander can damage expander blades and it also reduces the isentropic efficiency of the expander; this can be avoided by increasing the superheat degree.

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