Thermodynamic Analysis of Organic Rankine Cycle using Different Working Fluids

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***Abstract*— ORC (Organic Rankine Cycle) has a potential of reducing consumption of fossil fuel and has many favorable characteristics to exploit low – temperature heat sources. In this paper a comprehensive thermodynamic analysis of ORC using R245fa, R123, R600, R11 working fluids driven by waste low – grade heat source is conducted. Attention is paid to the effects of system parameters such as the turbine inlet temperature and pressure on the characteristics of the system such as volumetric flow rate, exergy efficiency, thermal efficiency, percentage of heat recovery, Net power produced are studied and the other parameters such as effect of vapor quality and effect of internal heat exchanger on thermal efficiency where also considered for 1 MW of power output.**

Keywords—Organic Rankine Cycle, low grade heat source, thermal efficiency, volume flow rate, exergy efficiency.

I. INTRODUCTION

Around the world as much as 40% of the total fuel consumption is used for industrial and process heating. Of this, around one third is wasted . Low grade heat has generally been discarded by industry and has become an environmental concern because of thermal pollution. This wasted heat can be lost to the atmosphere at all stages of a process, through inefficient generation, transmission, or during final use of the energy. This has led to the search for technologies which not only reduce the burden on non-renewable sources of energy but also take steps toward a cleaner environment.

In this context, utilization of waste heat for electricity production becomes significantly point of interest. Also to the fact that use of water/steam as working fluid causes some problems[2] like Superheating is required to avoid the condensation inside the last stages of turbine, low pressure during the condensation process is required, multistage turbine is needed because of the high pressure ratio, High volume flow rate, High temperature heat sources are required.

In recent years, Organic Rankine Cycle (ORC) has become a field of intense research and appears as a promising technology for conversion of low grade heat into useful electricity. The waste heat can be of various origins: solar radiation [3], biomass combustion [4], geothermal energy [5], or waste heat from process industries [6].

Vijayaraghavan and Goswami [7], Badr et al [8], saleh et al [9] are some of the researchers who analyzed the characteristics of different working fluids in an ORC application. Tranche et al. [10] investigated comparatively the Performance of solar Organic Rankine cycle using various working fluids. Volume flow rate, mass flow rate as well as thermal efficiency are used for comparison. Hung et al [11] examine Rankine cycles using organic fluids which are categorized into three groups of wet, dry and isentropic fluids. They point out that dry fluids have disadvantages of reduction of net work due to superheated vapor at turbine exit, and wet fluids of the moisture content at turbine inlet, so isentropic fluids are to be preferred. Kim [12] investigated comparatively the thermodynamic performance of ORC with super heater for various working fluids including wet, dry and isentropic fluids. Liu et al. [6] discussed an analysis of the performance of organic Rankin cycles subjected to the influence of working fluids. They investigated the effects of various working fluids on the thermal efficiency. Hung et al. [11] analyzed parametrically and compared the efficiencies of ORCs using cryogens such as benzene, ammonia, R11, R12, R134a and R113 as working fluids. The results showed that for operation between isobaric curves, the system efficiency increased for wet fluids and decreased for dry fluids while the isentropic fluid achieved an approximately constant value for high turbine inlet temperatures, and isentropic fluids were most suitable for recovering low temperature waste heat.

Even though some of the persons have compared the ORC performance with different working fluids, in this paper, the thermodynamic performance of ORC investigated for various working fluids. The various thermodynamic characteristics of the ORC such as net work production, and volume flow rate as well as thermal efficiency are investigated in terms of the parameters such as turbine inlet pressure and temperature. Some of the new parameters such as heat recovery efficiency, vapor quality at turbine inlet, effect of internal heat exchanger which affects the performance where also investigated.

II. SYSTEM DESCRIPTION

Fig.1. shows the schematic diagram of the Organic Rankine Cycle. The basic ORC system contains a working fluid pump, an evaporator driven by low-grade waste heat, a turbine, a generator and a condenser. Working fluid with a low boiling point is pumped into the evaporator, where it is heated and vaporized by the low-grade waste heat. The high pressure vapor from the evaporator flows into the turbine, where the

vapor gets expanded and the work is produced; simultaneously, the turbine drives the generator and electric energy is generated. Then, the exhaust vapor from the expander is released into the condenser and condensed by the cooling water. The condensed working fluid is pumped back to the evaporator, and another new cycle begins.

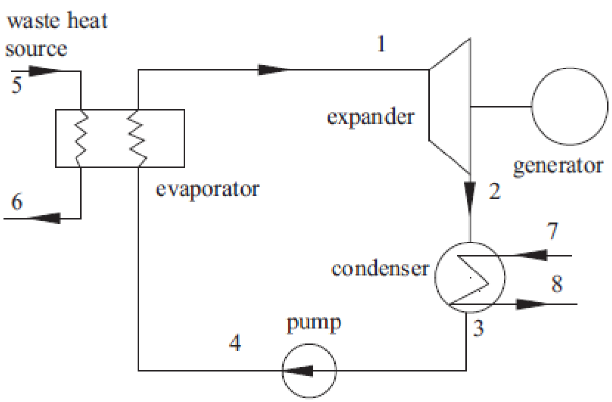

Fig.1. Schematic diagram of the ORC

The selection of working fluids is the critical to achieve high thermal efficiencies as well as optimum utilization of the available heat source and also involves various tradeoffs. The thermodynamic properties of working fluids will affect the system efficiencies, operating conditions and environmental impacts. Technically, the working fluids can be classified into three categories. Those are dry, isentropic, and wet depending on the slope of the cycle T-s diagram to be positive, infinite, and negative respectively.

The organic working fluids selected for this investigation are R245fa, R123, R600, and R11 with critical temperature ranking from 152 °C to 197.56 °C. Among the four working fluids, R11 is the isentropic fluid and remaining three fluids are dry fluids. Table 1 provides physical properties and table 2 represents safety and environmental data of considered working fluids.

TABLE I. PHYSICAL PROPERTIES OF WORKING FLUIDS

Working fluids	**Physical properties**		
	NBP(°C)	**T_C (°C)**	**Pc (bar)**
R123	27.8	183.68	38
R11	23.708	197.96	44.76
R245fa	15.14	154	36.51
R600	-0.5	152	36.96

NBP: NORMAL BOILING POINT

T_C = CRITICAL TEMPERATURE

P_C = CRITICAL PRESSURE

TABLE II. SAFETY AND ENVIRONMENTAL DATA

Working fluids	**Safety data**	**Environmental data**		**OEL**
	ASHARAE safety group	**ODP**	**GWP**	
R123	B1	0.01	90	50
R11	A1	1	4750	1000
R245fa	B1	0	1050	300
R600	A3	0	0	1000

ODP: OZONE DEPLETING PROPERTY

GWP: GLOBAL WARMING POTENTIAL

OEL: OCCUPATIONAL EXPOSURE LIMIT

III. THERMODYNAMIC MODELING

For the cycle performance modeling, various operating conditions are analyzed and compared for mentioned working fluids. The thermodynamic properties of working fluids and system performance are evaluated with a simulation tool Engineering Equation Solver, EES [13]. It is assumed that the pipe pressure drop and heat losses to the environment in the evaporator, condenser, expander and pump are neglected. The isentropic efficiency of turbine and pump are 85% and 75% respectively the condensing temperature kept at 30°C, the initial temperature at the turbine inlet was taken to be 100 °C and pressure corresponding to the saturation pressure of the particular selected working fluid at 100 °C. Fig.2. Shows the thermodynamic process of a basic ORC system can be illustrated in terms of a T-S diagram. More detailed processes of the ORC are as follows:

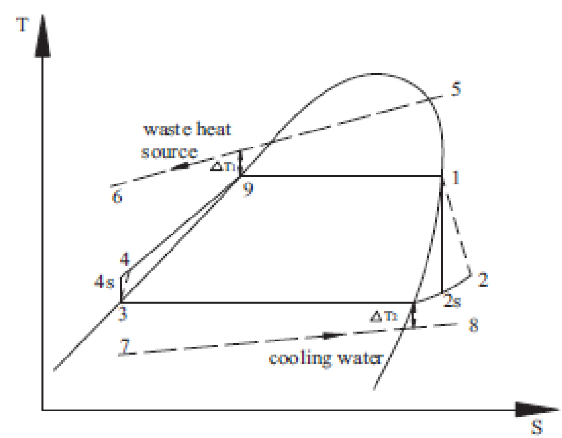

Fig. 2. TS Diagram of ORC

Process 4 to 1: This is an isobaric heating process in the evaporator. The low-grade waste heat source transfers the heat to the working fluid and then the working fluid is vaporized. The heat absorbed by the working fluid in the evaporator would be given by:

$$Q_{evp} = m_h(h_5 - h_6) = m_{wf}(h_1 - h_4) \quad (1)$$

Where, m_h and m_{wf} are the mass flow rate of the waste heat and working fluid, respectively. h_5, h_6, h_4 and h_1 are the specific enthalpies of the waste heat and working fluid at the inlet and exit of the evaporator, respectively.

Process 1 to 2: The high pressure vapor working fluid from the evaporator enters the expander, where the heat energy is converted into mechanical power. Then the mechanical power is converted into electric energy by the generator. The power generated by the turbine could be defined as:

$$w_t = m_{wf}(h_1 - h_{2s})\eta_s\eta_g \qquad (2)$$

Where, h_2 is the specific enthalpy of the working fluid at the outlet of the turbine, h_{2s} is the specific enthalpy of the working fluid at the outlet of the turbine in the ideal case. η_s and η_g are the turbine isentropic efficiency and generator efficiency, respectively.

Process 2 to 3: This is an isobaric heat rejection process in the condenser. The exhaust vapor at the outlet of the expander enters the condenser and releases the latent heat into the cooling water. The total heat released by the working fluid in the condenser could be expressed as:

$$Q_c = m_{wf}(h_2 - h_3) \qquad (3)$$

Where, h_3 is the specific enthalpy of the working fluid at the outlet of the condenser.

Process 3 to 4: In a real situation, this is a non-isentropic compression process in the pump. The power input by the pump could be expressed as:

$$W_p = \frac{m_{wf}(h_{4s}-h_3)}{\eta_p} = w_{wf}(h_4 - h_3) \qquad (4)$$

Where η_p is the isentropic efficiency of the pump h_{4s} and h_4 are the specific enthalpies of the working fluid at the outlet of the pump for the ideal and actual condition, respectively.

Thermal efficiency is the ratio between the net work outputs to the absorbed thermal energy in the evaporator.

$$\eta_{thermal} = \frac{(W_t - W_p)}{Q_{evp}} \qquad (5)$$

Volumetric flow rate at expander inlet can be calculated by dividing mass flow rate to the density at expander inlet.

$$V_{inlet} = \frac{m_{wf}}{\rho_{inlet}} \qquad (6)$$

The Net Work Out represents the difference between the expander power out and the power absorbed by the fluid circulation pump.

$$Net\ work\ out = W_{\text{turbine}} - W_{pump} \qquad (7)$$

Heat recovery efficiency is the ratio of maximum heat transfer capacity to the actual heat transfer capacity.

$$\eta_{heat\ recovery} = \frac{Q_{evpmax}}{Q_{evpactual}} \qquad (8)$$

According to D.Y. Wang the exergy efficiency evaluates the proximity of the real and Carnot cycle [14].

$$\eta_{II} = \frac{\eta_{thermal}}{\eta_{carnot}} = \frac{W_{net}}{Q_{evaporator}\times\left(1-\frac{T_L}{T_H}\right)} \qquad (9)$$

Where $T_{L=}$ Temperature of low temperature reservoir (°C)

$T_{H=}$ Temperature of high temperature reservoir (°C)

IV. RESULTS AND DISCUSSIONS

In this section the results of the analysis carried out on a modeled system using different working fluids are represented. Fig.3. shows the variation of the system thermal efficiency with the turbine inlet temperature. Basically, this figure shows the effect of superheating of the working fluid over the thermal efficiency of the cycle. It can be seen that the efficiency of the cycle for the evaluated organic fluids is a weak function of the turbine inlet temperature, because it remains approximately constant or slightly decreases with the increment of the turbine inlet temperature. It is because exergy losses increase with increases in inlet temperature.

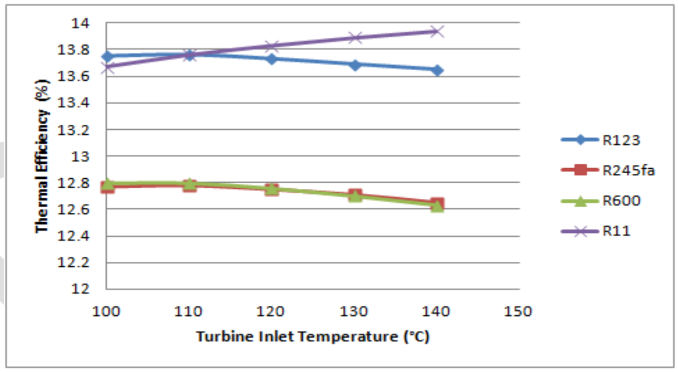

Fig.3. Temperature VS thermal efficiency

The volumetric flow rate is an important parameter in Organic Rankine Cycle design and component sizing. Fig.4. shows that the volumetric flow rate increases for all working fluids with increasing inlet temperature.

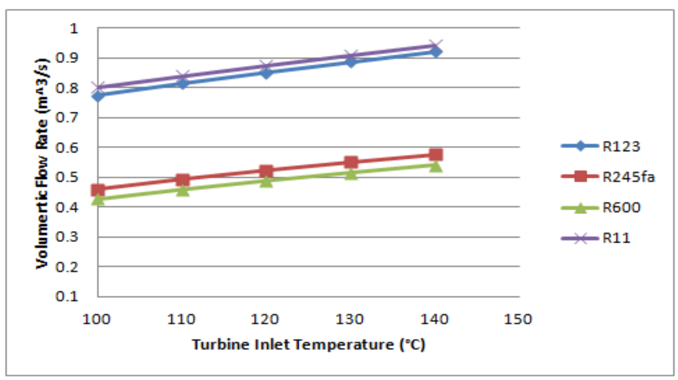

Fig.4. Temperature VS Volumetric flow rate

Exergy efficiency represents the ratio of the total exergy output to exergy input. From fig.5.it can be observed for all the fluids that exergy efficiency decreases with turbine inlet temperature because an increment in the system irreversibility yields a decrease in the system exergy efficiency.

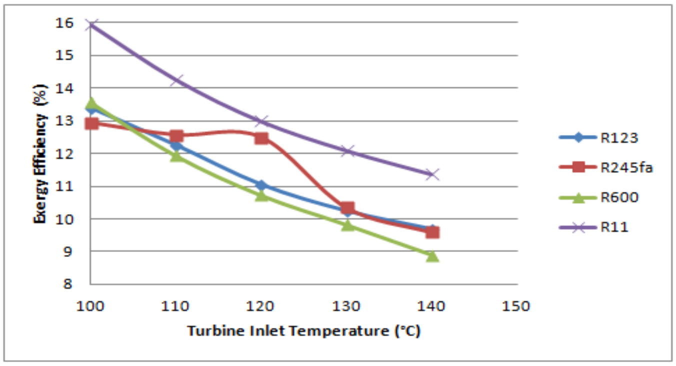

Fig.5. Temperature VS Exergy efficiency

Net Work of a working fluid shows how good the working fluid is to convert absorbed thermal energy from a certain heat source to useful mechanical work. Fig.6. shows that net work output increases with increases in turbine inlet temperature. R245fa shows the highest net work output with increase in temperature.

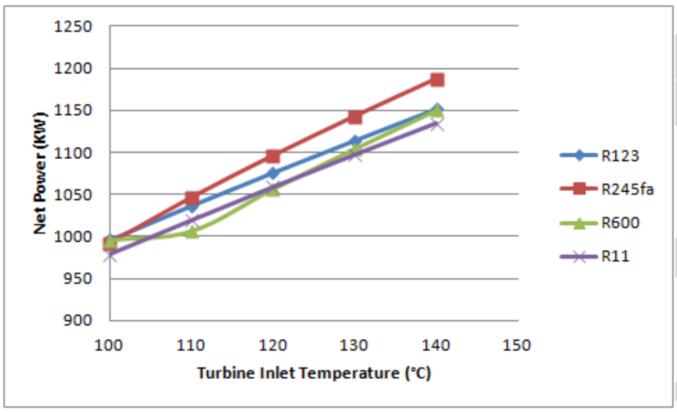

Fig.6. Temperature VS Net power

Fig.7. shows the effect of varying vapor quality of working fluid at the turbine inlet, it has been found that with increase in vapor quality of working fluids net power production increases and the thermal efficiency.

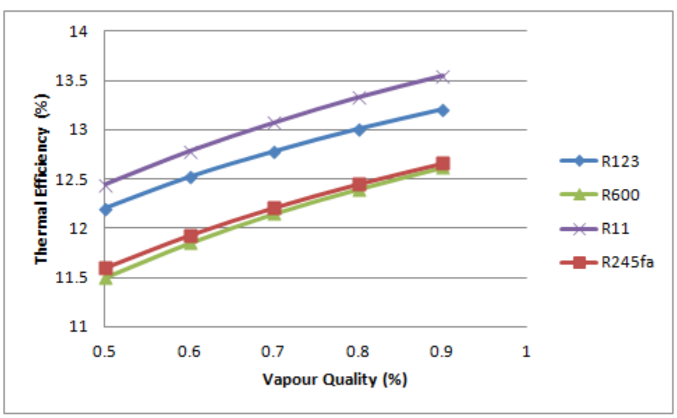

Fig.7. Vapor quality VS Thermal efficiency

Heat recovery efficiency is the ratio of maximum heat transfer capacity to the actual heat transfer capacity. Fig.8.shows the heat recovery efficiency in which with increase in turbine inlet temperature heat recovery starts to increase because temperature difference increases and hence the heat recovery from the hot fluid increases.

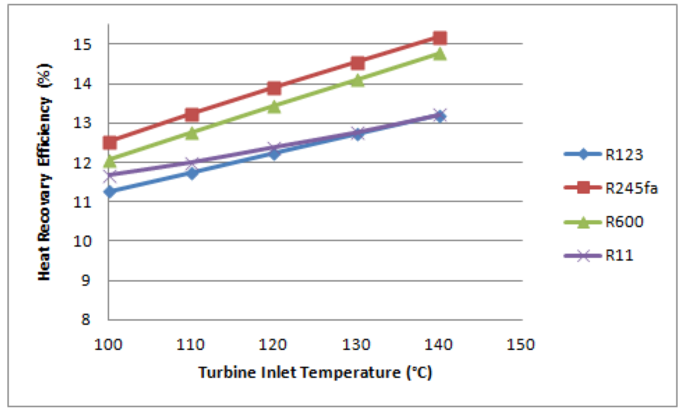

Fig.8. Temperature VS Heat recovery efficiency

Fig.9. illustrates the variation of the system thermal efficiency with the turbine inlet pressure while keeping the turbine inlet temperature at saturated conditions. For this case, the condenser temperature was kept constant at 30C. The results are consistent for all the fluids, because the system thermal efficiency increases with the increment of the turbine inlet pressure for all of them. This can be explained since with an increase in the inlet turbine pressure, both the net work and the evaporator heat increase. However, the percentage of increase of the net work is higher than that of increase of the evaporator heat. Therefore, the ratio of the net work to the evaporator heat increases with the turbine inlet pressure.

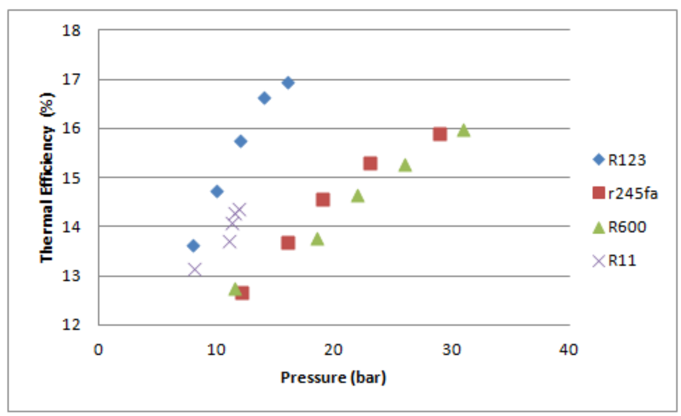

Fig.9. Pressure VS Thermal efficiency

When the working fluid leaves the turbine at a temperature higher than the temperature at the pump outlet, an internal heat exchanger can be introduced in the cycle to recover the thermal energy from the working fluid at expander outlet. Fig.10.shows the thermal efficiencies of different working fluids when internal heat exchanger is placed after turbine exhaust before the condenser, there is an increase in the thermal efficiency as internal heat exchanger raises the liquid the liquid temperature when it leaves the pump and the working fluid preheats before entering evaporator.

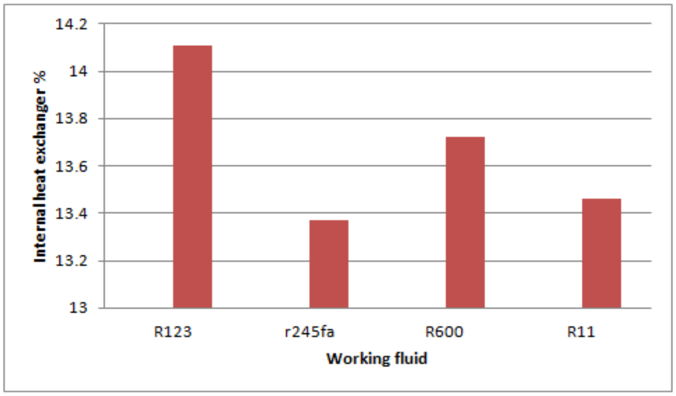

Fig.10. Working fluids VS Effect of IHE

CONCLUSION

In this paper, the performance of Organic Rankine Cycle with different working fluids has been thermodynamically analyzed. The main results are as follows.

- The fluid selection process is a trade-off between thermodynamic, environmental and safety properties.
- The selection of the optimal working fluid depends basically on the available heat source and the heat sink temperatures.
- The selected working fluid should have good thermodynamic properties like high thermal efficiency and Net work output.
- Isentropic fluids show higher efficiency than wet or dry fluids.
- Organic fluids need not be superheated as the cycle thermal efficiency remains constant when the inlet temperature of the turbine is increased.
- Heat recovery efficiency increases with increase in turbine inlet temperature.
- The volumetric flow rate and the working fluid viscosity should be as low as possible
- The working fluid must be saturated vapour before inlet to the turbine for maximum thermal efficiency.
- Introduction of IHE improves the thermal efficiencies of the cycle.
- GWP and ODP as low as possible.
- No working fluid is ideal.

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