PERFORMANCE OPTIMIZATION OF A CONSTANT POWER SUBCRITICAL ORGANIC RANKINE CYCLE FOR LOW GRADE HEAT SOURCE WITH VARYING TEMPERATURE

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ABSTRACT

Thermodynamic analysis is carried out for a ORC with working fluid R134a. R134a is chosen based on its suitability for conversion of heat available from low grade sources, typically from sources like geothermal springs. The ORC considered in this study is subcritical. Optimization is carried out iteratively to calculate the efficiency of organic Rankine cycle at various combinations of evaporator pressure, turbine inlet temperature and to find the maximum ORC efficiency under the constraint that the ORC should always deliver a constant net power output of 1MW. Consequently the mass flow rates of $\dot{m}_{ORC}$, $\dot{m}_{HTF}$ at various combinations of $[P_{Evap}, T_5]$ are evaluated. The maximum efficiency of ORC is found to be 8.857% at a P_{Evap} of 3.6530 (MPa) and 453 K of T_5 . The corresponding values of $\dot{m}_{ORC}$ and $\dot{m}_{HTF}$ are 38.201 Kg/s and 34.669 Kg/s respectively.

KEYWORDS: ORC, R134a, Low Grade Heat Source & Pinch Point

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1. INTRODUCTION

A promising heat-power conversion technology for low and medium temperature (<400 °C) heat sources is the organic Rankine cycle (ORC), which is based on the Rankine cycle principle and organic fluids with low boiling points as working fluids [1]. Low -grade waste heat has usually been discarded by industry due to lack of effective recovery methods and has become an environmental issue due to thermal pollution. In a conventional power plant, one-third of the fuel energy is converted to electricity and the remainder of two-third as wasted and dumped to the atmosphere. It causes global warming and environmental unrest. A significant portion of industrial energy input is rejected as waste heat highlighting the importance of waste heat recovery techniques to enhance the power cycle heat use and energy conversion efficiency.

Reconstruction of the waste heat has become popular recently. Bottom-up electricity generation is the promising option as it consumes virtually no extra fuel. In addition to the waste heat recovery system, a non-condensation steam turbine can be used for a steam cycle in a topping cycle [2]. The heat source for such cycles happens to be the exhaust heat coming from the topping cycles or from the heat sinks of topping cycles [3]. The schematic sketch of organic Rankine cycle is shown in

Figure 1 where the heat source can be low grade energy reservoirs like geothermal hot springs, solar energy or waste heat from the thermodynamic process [4]. The effective utilization of the primary energy resources like coal, oil and gas is possible by attaching another heat-power conversion cycle like Kalina cycle, Goswamy cycle and organic Rankine cycle as bottoming cycle to that of topping cycles [5][6].

Present study will not discuss about any particular heat source to the organic Rankine cycle it would be renewable energy resources like thermal hot springs, solar energy or waste heat from any thermodynamic process, it is constrained to evaluation of the thermodynamic performance of the conventional subcritical organic Rankine cycle which producing continuously 1MW power output under design conditions. The selection of the heat transfer fluid to absorbs heat from the thermal reservoirs is rely on stability and thermo physical properties at low and medium temperature energy resources [7]. It can be understood that most widely used heat transfer fluids are molten salts, thermal VP-1 etc. [8].

Most of studies have focussed on the suitability of the heat transfer fluid for the existing low and medium heat sources to absorb the maximum energy from them with the help of kalina cycle, Goswamy cycle and organic Rankine cycle etc. [9]. Installation of such power plants in the present study are suitable where the renewable energy resources and waste heat sources are abundantly available i.e. the tropical regions or sun belts and industrial sectors. In the present study, there is no discussions about the performance of the system which is effected by the characteristics of the heat transfer fluid [10]. For a given value of condenser pressure and pump efficiency, the work done or power obtained from a organic Rankine cycle is thermodynamically represented by the area enclosed with in the organic Rankine cycle and this keeps increasing with the increasing evaporator pressure as well as the inlet turbine temperature [11].

Figure 1: Schematic sketch of organic Rankine cycle

The nominal net power output from the system is limited to be 1MW indicating that the area under organic Rankine cycle in T-s chart as shown in **Error! Reference source not found.** is constant.

Figure 2: Rankine cycle on T-s chart

Under such design conditions, since we are trying to enhance the performance of the organic Rankine cycle, it is to be known that the performance of the system or thermodynamic efficiency depends on both the operating parameters are evaporator pressure as well as the turbine inlet temperature. The comparison of the present operating mode with the Carnot heat engine, it is presumed that the average temperature at which the heat transfer is taking place from the heat transfer fluid to the organic fluid has a great impact on the thermodynamic performance of the system. Since in a subcritical organic Rankine cycle this temperature is not constant due to different temperatures exist in the three sections of a boiler i.e. economiser, evaporator and super heater with temperature being constant in evaporator and variable in economiser and super heater, respectively [12].

In the present study, the evaporator pressure will effects the operation of the organic Rankine cycle under design conditions and hence the performance of the system also will effect that's why it is consider as the one of the operating parameter in the optimization process. Similarly, turbine inlet temperature also will effects the operation of the organic Rankine cycle under design conditions, even at a given evaporator pressure, the presence of turbine inlet temperature in the superheated region is mandatory, for enhancing thermal efficiency as it improves the average heat source temperature and to guarantee the safe operation of the turbine. The wet condition of Organic Rankine fluid in the turbine expansion process will cause significant damage of the turbine blades. Therefore, a promising organic Rankine cycle cannot sustain to have wet expansion in the turbine. Thus determination of the value of one of the operating parameter in the present study i.e. turbine inlet temperature should guarantee the dry vapor expansion in the turbine and to gives the maximum thermodynamic efficiency under constant power output of 1 MW from the heat-power conversion cycle or organic Rankine cycle

2.METHODOLOGY ADOPTED

Figure 1 represents the schematic layout of the organic Rankine cycle and from it we can see the pump followed by heat recovery steam generator consisting of three subsections namely economiser, evaporator and superheater followed by a turbine and a condenser. The corresponding thermodynamic T-s chart of the Rankine cycle is represented in Fig.2. Typically [13] the upper limit for the evaporator pressure is of the order of 90% of the critical pressure of the organic fluid i.e. in organic Rankine cycle, the lower limit of the evaporator pressure is about at least 100 KPa greater than the condenser pressure of the cycle and therefore the space between these two pressure lines happens to be the search space of

exploration to pick up the optimal evaporator pressure in conjunction with the turbine inlet temperature, so as to maximise the efficiency of the organic Rankine cycle [14]. The upper limit for the turbine inlet temperature is based on pinch point temperature difference prevailing at the exit of the boiler i.e. at the super heater [15]. Similarly, the lower limit for T_5 is to the expansion of the organic fluid in the turbine to be dry.

The present study evaluates ORC at different pinch point temperature difference of 20°C and 5°C . Iterative procedure adopted to choose appropriate values at evaporator pressure and turbine inlet temperature to enhance or maximise the efficiency. The most significant aspect of this iterative procedure is that the three important conditions are ensured as part of the procedure. Firstly, starting with an assumption that the satisfaction of the pinch point temperature difference at the boiler inlet i.e. at the economiser section, calculations are done iteratively to find the temperature at the exit of the boiler i.e. at the super heater section. If and in the iterative process, the total heat load subject to the condition of the satisfaction of constant power output from the organic Rankine cycle is also ensured thereafter the mandatory pinch point temperature difference at exit of the super heater is also to be complied with, failing which, the procedure resets the temperature to ensure the pinch point temperature difference at the super heater and the iterative procedure keeps continuing till all these three conditions i.e. the pinch point temperature difference at the boiler inlet, the pinch point temperature difference at the boiler exit as well as the total heat load satisfaction over the boiler are all ensured. The iterative procedure stops upon the satisfaction of all these criteria along with the constraint of fixed power output of 1MW from ORC. The thermodynamic parameters obtained under such conditions i.e. the values of T_2 , T_3 , T_4 , T_5 and T_6 are considered to evaluate the ORC efficiency. Therminal VP-1 oil has to be considered as HTF in this analysis, because it has good thermo-physical properties, environmentally friendly and economically feasible when compared to other heat transfer fluids for the assumed operating conditions of the present study [16] [20-21].

3. PROCESS DESCRIPTION

The schematic organic Rankine cycle is shown in the

Figure 1 and its thermodynamic representation in **Error! Reference source not found.**. At state 1, the saturated R134a is pressurised to state 2, a compressed liquid state with desired pressure with the help of feed pump (1 to 2 process). The subcooled liquid absorbs the heat in 3 sections which includes preheating of the compressed liquid in economiser (2-3 process), saturated liquid is evaporated to saturated vapor in the evaporator section (3-4 process) and super heating to super-heated vapor in the super heater section (4-5 process). The super-heated vapor expands in expander (5-6 process) and produce electric power through the connected generator. The exhaust expanded gas is cooled (6-7 process) and condensed in a condenser (7-1 process) to complete the cycle.

Table 1: Assumed operational conditions of ORC

Parameter	Symbol	Value
Net Power output /MW	P_{net}	1
Inlet temperature of heat transfer fluid $^{\circ}\text{C}$	$T_{\text{HTF, in}}$	200
Evaporator pinch point temperature difference $^{\circ}\text{C}$	$\Delta T_{\text{Evp, pp}}$	20
Condenser pinch point temperature difference $^{\circ}\text{C}$	$\Delta T_{\text{Con, pp}}$	10
Coolant inlet temperature $^{\circ}\text{C}$	$T_{\text{cool, in}}$	20
Coolant temperature at pinch point $^{\circ}\text{C}$	$T_{\text{cool, pp}}$	27
Condensation temperature $^{\circ}\text{C}$	T_{Con}	37
Efficiency of the turbine/%	η_T	80
Efficiency of generator/%	η_g	97
Efficiency of feed pump/%	η_p	75

Efficiency of motor/%	η_{motor}	75
Pressure head/m	H	10
Specific heat of HTF /KJ.kg ⁻¹ .K ⁻¹	C _{p,HTF}	2.319

The electric power produced from the generator is utilised to supply the motive power for the coolant pump and feed pump, and the remaining power is considered as the net power output. The net power output from the turbine can be calculated as

$$P_T = \eta_g \dot{m}_{wf} (h_5 - h_6) \quad (1)$$

where $\dot{m}_{wf}$ is mass flow per unit time of working fluid in ORC; turbine entry and exit enthalpies of the working fluid are h_5 and h_6 respectively.

The net power output consumed from coolant pump and feed pump are.

$$P_{cool,p} = \frac{\dot{m}_{wf} (h_7 - h_1)}{\eta_{motor} (h_{cool,pp} - h_{cool,in})} g H \quad (2)$$

$$P_p = \frac{\dot{m}_{wf} (h_2 - h_1)}{\eta_p} \quad (3)$$

where h_7 is the enthalpy of R134a at entrance of the condenser; h_1 and h_2 are the enthalpies of R134a at inlet and outlet of the feed pump respectively; $h_{cool,in}$ is enthalpy of the coolant at the inlet of the condenser; $h_{cool,pp}$ enthalpy of the coolant at pinch point in the condenser.

The net power output is given by

$$P_{net} = P_T - P_{cool,p} - P_p \quad (4)$$

where P_{net} is a fixed value and $\dot{m}_{wf}$ can be obtained from (4).

The heat transported by heat transfer fluid is completely supplied to the R134a(working fluid) in the evaporator. The energy balance equation for HRVG is

$$\dot{m}_{HTF} C_{p,HTF} (T_{HTF,in} - T_{HTF,out}) = \dot{m}_{wf} (h_5 - h_2) \quad (5)$$

The outlet temperature of heat transfer fluid in the evaporator should be calculated from pinch point technology[17]. In an organic Rankine cycle the evaporator pinch point occurs at the inlet of the working fluid (state 2 in Fig.2.) or at the starting of evaporation process (bubbling point, state 3 in Fig.2), the mass flow rate of the HTF can be evaluated as $\dot{m}_{HTF}$ [15].

The isentropic efficiency of the turbine and feed pump are

$$\eta_T = \frac{W_{act}}{W_{ideal}} = \frac{h_5 - h_6}{h_5 - h_{6s}} \quad (6)$$

$$\eta_p = \frac{W_{ideal}}{W_{act}} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (7)$$

where h_{6s} is isentropic enthalpy of R134a (working fluid) at the turbine outlet for an isentropic

expansion process, and h_{2s} is the isentropic enthalpy of the working fluid at outlet of the feed pump for an isentropic compression process.

The thermodynamic efficiency of the subcritical ORC can be evaluated as

$$\eta_{ORC} = \frac{P_{net}}{\dot{m}_{HTF} C_{p,HTF} (T_{HTF,in} - T_{HTF,out})} \quad (8)$$

As the nature of variation of η_{ORC} with respect to P_{evp} and T_5 is already understood to be positive i.e increasing with increase in P_{evp} and T_5 , the rate of variation therefore would be of interest. This gives us an understanding about the change in the thermodynamic performance of ORC, with a change in T_5 at any given P_{evp} . Similarly, the corresponding variation in the mass flow rate of ORC and HTF can be studied.

4. RESULTS AND DISCUSSIONS

Figure depicts the variation of η_{ORC} with respect to the evaporator's pressure and turbine inlet temperature, for a given T_{HTF} . It is observed that the efficiency increases with increasing pressure. Also at a given evaporator pressure, the η_{ORC} keeps increasing with increase in T_5 . Figure corresponds to the variation η_{ORC} with respect to P_{evp} and T_5 for the T_{HTF} value of 473 K where η_{ORC} is maximum. Similar trends are observed at other values of T_{HTF} . The value of η_{ORC} at various values of $T_{HTF,in}$ with the corresponding optimal values of P_{evp} and T_5 are given in Table 3 and **Error! Reference source not found.**

From Figure , it can be observed that, the maximum η_{ORC} is obtained at a (P_{evp} , T_5) combination of 3.653MPa and 453K, for a $T_{HTF,in}$ of 473K, with a corresponding maximum value of η_{ORC} of 8.857%. It can also be observed that, for a ΔT_{pp} of 5°C, there is not much difference in the η_{ORC} . A similar trend is observed at all other combination of P_{evp} and T_5 .

Table 2: Maximum ORC efficiencies at various [P_{Evap} , T_5]

P_{Evap} (MPa)	$\Delta T_{pp}=20$ ($T_{HTF,in}=200/190/180/170$)		$\Delta T_{pp}=5$ ($T_{HTF,in}=180/170/160/150$)	
	T_5 (K)	η_{ORC} (%)	T_5 (K)	η_{ORC} (%)
3.6530	453	8.857	448	8.841
3.2530	453	8.218	448	8.206
2.8530	453	7.451	448	7.439
2.4530	453	6.496	448	6.501
2.0530	453	5.302	448	5.307
1.6530	453	3.768	448	3.762
1.2530	453	1.678	448	1.676

Table 3 Maximum efficiency and corresponding optimal operating conditions at various $T_{HTF,in}$, [$\Delta T_{pp}=20$ °C]

$T_{HTF,in}$ (°C)	200	190	180	170
P_{Evap} (MPa)	3.6530	3.6530	3.6530	3.6530
T_5 (K)	453	443	433	423
η_{ORC}	8.857	8.823	8.755	8.684
$\dot{m}_{ORC}$ (Kg/s)	38.201	40.053	42.262	44.745
$\dot{m}_{HTF}$ (Kg/s)	34.669	37.468	40.893	44.962

4.1 Effects of Turbine Inlet Temperature and Evaporator Pressure

Irrespective of the HTF temperature, η_{ORC} has always increased with increase in P_{evp} . The same kind of behaviour cannot be ascribed for ORC cycles operating with other working fluids, as some studies [18][19][20] reported otherwise. The underlying reason for this difference is that, for certain fluids like Toulene, Cyclohexane etc, the saturation vapour curve displays a inflection point. This particular feature of such working fluids, coupled with the constraint of ensuring a dry expansion in turbine results in maximum values of η_{ORC} to occur not at maximum P_{evp} always. For such fluids, best therodynamic performance can be at various combinations of $[P_{evp}, T_5]$.

Table 4: Maximum efficiency and corresponding optimal operating conditions at various $T_{HTF, in}, [\Delta T_{pp} = 5^\circ C]$

$T_{HTF, in} (^\circ C)$	180	170	160	150
$P_{Evap} (MPa)$	3.6530	3.6530	3.6530	3.6530
$T_5 (K)$	448	438	428	417.17
η_{ORC}	8.841	8.796	8.725	8.617
$m_{ORC} (Kg/s)$	39.098	41.097	43.437	46.473
$m_{HTF} (Kg/s)$	36.012	39.083	42.810	47.458

Figure 3: Effects of turbine inlet temperature and pressure on the efficiency of ORC

4.2. Effect on Mass flow rate of HTF and working fluid

From the results, it is observed that the mass flow rate of working fluid in the cycle is low at high evaporator pressure and its corresponding upper limit of the turbine inlet temperature and vice versa. The variations in the mass flow rate of working fluid in the organic Rankine cycle at different evaporator pressures and turbine inlet temperatures ensures constant heat supply to the organic Rankine cycle and to ensure a constant net power output from the organic Rankine cycle. The values of the mass flow rate of R134a at various combinations of $[P_{Evap}, T_5]$ are given in Fig.5. A similar behaviour was observed for $\dot{m}_{HTF}$ Fig.6.

Figure 4: Variation of $\dot{m}_{ORC}$ with T_5 Figure 5: Variation of $\dot{m}_{HTF}$ with T_5

5. CONCLUSIONS

ORC is one of the preferred thermodynamic mode of utilising waste heat resources available in the nature as they are capable of utilising low-grade heat that is heat available at low temperatures as can be obtained from solar, biomass waste, exhaust of heat engines or any process waste heat. In the present study, one such ORC is considered, and its performance is optimized by way of maximising its thermodynamic efficiency under the constraint that it should always produce the same power output of 1MW. The Rankine cycle considered is assumed to be powered by a heat transfer fluid which receives its

heat from any of the low temperature heat source which in turn transfers that to the organic fluid in the organic cycle. From the Carnot theory of thermodynamic analysis, the efficiency depends on various thermodynamic cycle parameters. With fixed condenser pressure and fixed efficiency of the pump, which pumps the organic fluid from the exit of the condenser to the boiler, the power output of the cycle is also considered to be constant at 1MW. Under such constraints, the dependency of thermodynamic efficiency of the organic Rankine cycle evaporator pressure, inlet temperature of the organic fluid in to the turbine is analysed. In the present study, subjected to the constraints mentioned and to maximise the first law based thermodynamic efficiency, the optimum values of the turbine inlet temperature, evaporator pressure are determined with corresponding values of η_{max} . Results revealed that the maximum efficiency obtained at 3.6530 MPa and 453K values of evaporator pressure and turbine inlet temperature is 8.857%.

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