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# Performance Analysis of Organic Rankine Cycle Technology to Exploit Low-Grade Waste Heat to Power Generation in Indian Industry

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***Abstract*—**The demand for energy is cumulatively increasing with time. Since the availability of conventional energy resources is dying out gradually, significant interest is being laid on searching for alternate energy resources and minimizing the wastage of energy in various fields. In such perspective, low-grade waste heat from several industrial sources can be reused to generate electricity. The present work is to further the adoption of the Organic Rankine Cycle (ORC) technology in Indian industrial sector. The present paper focuses on extending the previously reported idea to the next level through a comparative review with three different working fluids using practical data from an Indian industrial plant. For comprehensive study in the simulation platform of Aspen Hysys®, v8.6, the waste heat data has been collected from a current coke oven gas plant in India. A parametric analysis of non-regenerative ORC and regenerative ORC is executed using the working fluids R-123, R-11 and R-21 for subcritical ORC system. The primary goal is to determine the optimal working fluid considering various system parameters like turbine work output, obtained system efficiency, irreversibility rate and second law efficiency under applied multiple heat source temperature (160 °C- 180 °C). Selection of the turbo-expanders is one of the most crucial tasks for low-temperature applications in ORC system. The present work is an attempt to make suitable recommendation for the appropriate configuration of the turbine. In a nutshell, this study justifies the proficiency of integrating the ORC technology in Indian perspective and also finds the appropriate parameter of all components integrated in ORC system for building up an ORC prototype.

***Keywords*—**Organic rankine cycle, regenerative organic rankine cycle, waste heat recovery, Indian industry.

## I. INTRODUCTION

IN current years, increasing energy needs, rise in thermal load demand in commercial, domestic sectors and industrial, and shortage of fossil fuels coupled with the faulty power supply and high fuel price have made people to consider on the greater use of renewable energy application. Apart from this, use of refrigerants with high global warming potential (GWP), $CO_2$ emissions from the combustion of fossil fuels in the power generation leads to effects detrimental to the environment [1]-[4]. Among the thermodynamic cycles, ORC poses to be a promising technology that can minimize global environmental pollution, reduce energy consumption [5] and enhance thermal energy efficiency by utilizing low and medium grade heat energy from sources such as solar, agricultural waste, geothermal, waste heat from industrial process, power plants and automobiles exhausts [6]-[8].

Waste heat recovery technologies are in dire need of innovative approaches to take a leading role in the present era of reduction in carbon footprint [9], [10]. In Paris climate change summit, it was suggested to reduce the greenhouse gas emission by 55% [11]-[14], and probable solution for the reduction of greenhouse gas emissions can be moving into cleaner heat sources, increasing energy efficiency and reusing the unutilized waste heat.

There are different types of cycles for low-grade waste heat recovery like Kalina cycle, Goswami cycle, Uehara Cycle, Maloney and Robertson cycle. These cycles work on low-grade waste heat temperature using ammonia – water mixture as working fluid. However, ORC works on low and medium grade temperature ranges, and it can use all different types (dry, wet and isentropic) of working fluids for power generation [1], [15]-[18]. Quoilin et al [8] present a current review of the ORC technologies for low-grade waste heat conversion. Industrial adoption of ORC technology is important because it will lead to improving energy efficiency, mitigating energy price hike, protecting the environment by reducing Greenhouse Gases (GHG) lessening primary energy consumption [19], [20]. Selection of the right industrial process is one of the key issues for waste heat recovery. Different researchers have shown that glass, iron and steel and cement industries are the most suitable candidates for converting the waste-heat to power [5].

Till date, the small size ranges ORC prototypes are on experimentation or under development in different research institutes. Few manufactures are making the small range capacity ORC power plant (1-100 kW), such as Electratherm, GMK, Verdicorp, Zuccato Energia, etc [9].

In this study, the possibilities of conversion of low-grade waste heat to power for the Indian industries using ORC have been explored. The most suitable working fluid for the low-grade heat conversion has been identified through the comparison of the performances based on three working fluids. This study provides an insight into the directions of future research on low-grade waste heat (LGWH) for power conversion and postulated a new technology for the development of LGWH power plant technology for Indian

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industries.

Currently, the heat recovery is a method rarely implemented in Indian industries [21], [22]. The upstream waste heat treatment facilities employ chimneys discharging low-temperature gases, resulting in loss of the associated thermal power. Waste heat can be recovered for the generation of thermal, mechanical or electrical energy. The present paper presents the opportunities to recover waste energy and use it for producing electricity. In this regard, ORC has been implemented using the data from a coke oven plant in India.

## II. Industrial Data Collection: Waste Heat Recovery Utilization Opportunities

The installed capacity of CHP in Bengal Energy Ltd is 40 MW from 0.6 MTPA LAM Coke Production capacities. Case study is shown in Table I.

TABLE I
THE DATA OF CHP IN BENGAL ENERGY LTD. [1]

| Plant details | System parameter |
|---|---|
| Power production capacity | Gas turbine generator, 40 MW, 11 kVA |
| Excess waste heat temperature from coke oven battery | 950-1050 ºC |
| flow rate coke oven | 95000- 110000 N $m^3$/ h / Each Battery |
| Delivery grid | West Bengal State Electricity Distribution Company |
| ID fan capacity | ID Fan 125 kW x 4-24 hour Running |
| Chiller type | Cooling Towers, Water-cooled Condenser |
| CHP system efficiency | 90 % |
| Waste heat temperature from CHP (Unutilized) | 165 ºC-180 ºC |

## III. Methodology

An existing coke oven gas plant ‘Bengal Energy Ltd’ is located in Kharagpur, West Bengal, India. This plant has LGWH temperature ranging between 165º-180ºC. In our study we have used this waste heat temperature as a waste heat input. Two different ORC models have been studied for LGWH recovery: ORC and Regenerative Organic Rankine cycle (RORC). These two different systems have been selected for comparison to identify the advantages of LGWH recovery. The CHP system exhaust temperature is about 200 ºC under normal conditions, but the exhaust temperature in the chimney exhibits 165º-180ºC. Finally, the exhaust flow is typically 95000- 110000 N $m^3$/ h / Battery (coke burning cell), depending on ambient temperature.

In this paper, three different organic fluids have been examined using Aspen Hysys®, v8.6 and their thermodynamic performances are compared. The Aspen models have also been evaluated to decide the best operating condition to get maximum output power. Based on the parametric study of ORC cycles in Aspen Hysys®, v8.6, the turbo expander design has been incited. The design principles of Kun and Sentz [10] and Balje [11] has taken for turboexpander design. Balje’s $n_s d_s$ chart was used to obtain the preliminary design parameters of turbo expander. The detailed design of the turbine wheel would provide recommendations for improved cycle performance.

### A. Simulation Model

The ORC system is used for conservation of low-grade energy sources. This system uses high molecular mass organic fluid for low grade waste heat recovery system. A schematic of the ORC cycle is depicted in Fig. 2 (a). The ORC consists of (a) Shell and Tube Heat Exchanger (E-100), (b) Turbo Expander (K-100), (c) Condenser (E-101) and (d) Working Fluid Pump (P-100). The regenerative cycle consists of two thermodynamic cycles. The diagrams of the cycles are depicted in Fig. 2 (b). The Regenerative Cycle consists of an (a) Shell and Tube Heat Exchanger (E-100), (b) Regenerator (E-101), (c) Turbo Expander (K-100), (d) Condenser (E-102) and (e) Working Fluid Pump (P-100).

Ref 0 actual ambient condition, Ref 1, 2 turbine inlets, turbine outlet/ Regenerator (E-101) inlet, Ref 2*, 4* Regenerator (E-101) outlet/condenser inlet, Regenerator (E-101) outlet/ Shell and Tube Heat Exchanger (E-100) inlet

Ref 3, 4 pump inlets, pump outlet/ Regenerator (E-101) inlet

### B. Selection of Working Fluid

Selection of working fluid is based on the temperature of the waste heat source. Properties like higher temperature for decomposition, higher stability to evade chemical deterioration, non-flammability, non- toxic and non-explosive nature are vital while selecting the working fluids. Table II lists the selected working fluids to be studied and Fig. 2 shows the T-s plot of saturation curves drawn using Aspen Hysys®, v8.6. The criteria for selecting the working fluids are:

1. Isentropic and dry working fluids are applicable for ORC and RORC technology for LGWHR. Since these fluids have superheated zone after the isentropic expansion no liquid droplets enter the turbo expander.
2. In this study, water cooler has been used for condensing the working fluids and it is assumed that the maximum ambient air temperature is 30 °C with minimum temperature approach of 4 °C. Working fluids with normal boiling temperatures within a range of 0 °C to 30 °C, were selected.
3. R-11, R-21 and R-123 being members of chlorofluorocarbon (CFC) and hydro-chloro-fluorocarbon (HCFC) family of refrigerants, as chlorine is dissipated at lower altitudes, these molecules have relatively short atmospheric life times and lower ODP [12].

### C. Analysis and Calculation Model

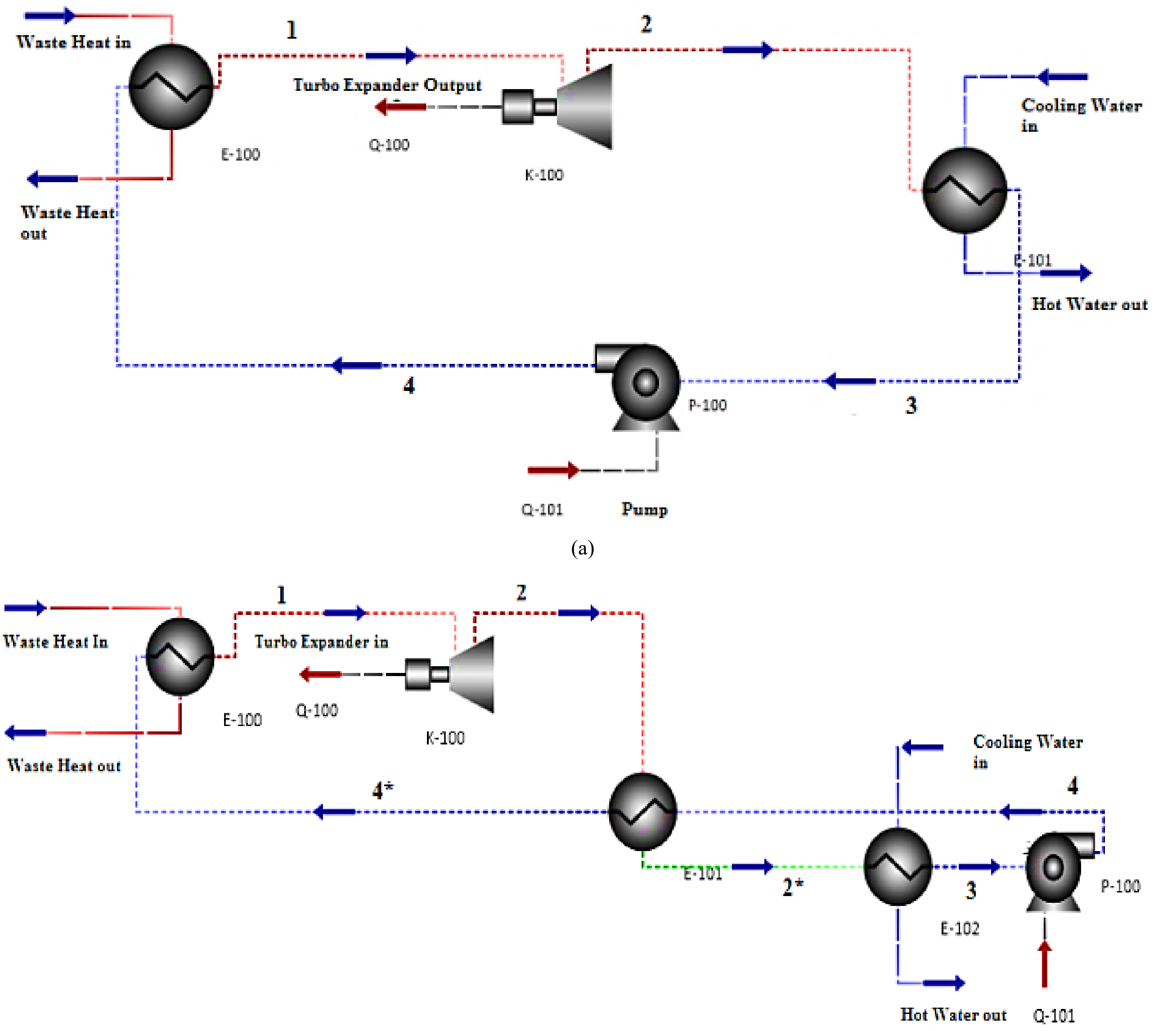
The software Aspen Hysys®, v8.6, made the thermodynamic analysis of these cycles. The parameters used in making the simulation model are shown in Table III. The energy balance equation of ORC and RORC are shown in Tables IV and V.

### D. ORC Cycle Energy Balance Equation

Equation (1) shows the external irreversibilities occurring inside the ORC and RORC system:

$$\dot{I} = \dot{m}T_0\left[\Sigma s_{outlet} - \Sigma s_{inlet} + \frac{dS_{sys}}{dt} + \sum_k \frac{q_k}{T_k}\right] \tag{1}$$

where $\dot{I}$ = irreversibility rate [kW], $T_0$ = is the ambient temperature, $q_k$ = is the heat transferred from all heat source to the working fluid and $T_k$ = refers to the temperature of all heat sources, $\dot{m}$ = mass flow rate [kg/sec]. When the system reaches the steady-state, $dS_{sys}/dt = 0$,



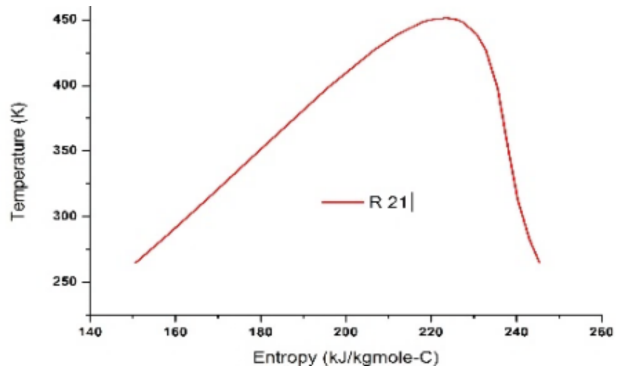

(a)

(b)

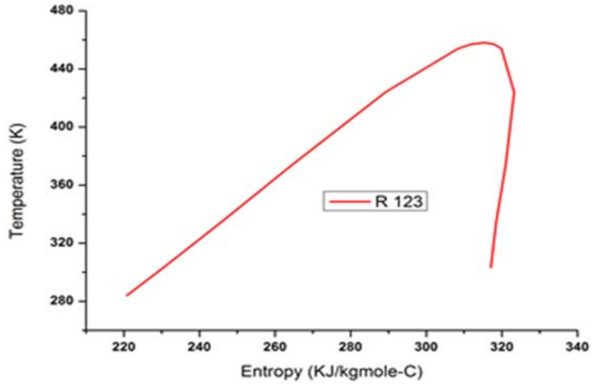
Fig. 1 (a) Layout of the ORC simulated in Aspen Hysys®, v8.6 (b) Layout of the regenerative cycle simulated in Aspen Hysys®, v8.6. The numbers in this diagram indicated the fluid flow condition

TABLE II
SELECTED WORKING FLUIDS DATA [1]

| Working fluids | | Molecular mass | NBP | $T_C$ | $P_C$ | Safety data | | Environmental data | | |
|---|---|---|---|---|---|---|---|---|---|---|
| ASHRAE Number | Chemical formula and Name | | °C | °C | MPa | LFL (%) | ASHRAE 34 Safety group | Atmospheric Life (years) | ODP | GWP 100 year |
| R-11 | $CCl_3F$ (Trichlorofluoromethane) | 137.37 | 23.7 | 198 | 4.41 | None | A1 | 45 | 1 | 4750 |
| R-21 | $CHCl_2F$ ( Dichlorofluoromethane) | 102.92 | 8.9 | 178.3 | 5.18 | None | B1 | 1.7 | 0.01 | 151 |
| R-123 | $CHCl_2CF_3$(2,2-Dichloro-1,1,1-trifluoroethane) | 152.93 | 27.8 | 183.7 | 3.66 | None | B1 | 1.3 | 0.02 | 77 |




(a)

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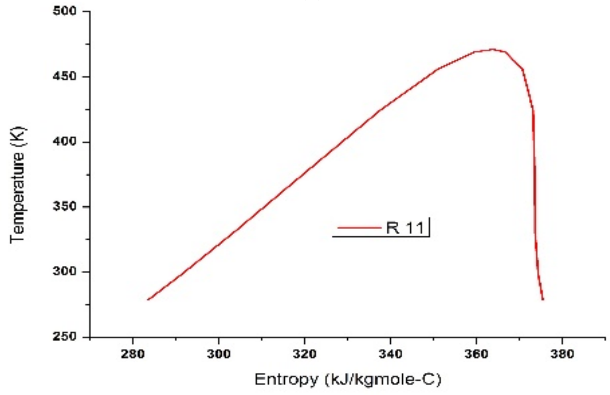



(c)

Fig. 2 Types of working fluid based on t-s plot: dry (a) R 21 and (c) R 11 and (b) isentropic R 123) [1]

TABLE III
PARAMETER USED IN MAKING THE SIMULATION MODEL

| Parameter | Value | Parameter | Value |
|---|---|---|---|
| Avg. ambient air temperature | $30^0$C | Turbo expander adiabatic efficiency | 75% |
| Exhaust gas flow rate | 0.5 kg/s | Pump efficiency | 75% |
| Exhaust gas outlet temperature | $100^0$C | Heat exchanger pressure drop | 25 kPa |
| Exhaust gas inlet temperature | $300^0$C | Property | Peng – Robinson |
| Temperature difference in heat exchangers | $4^0$C | Package | |

The Irreversibility rate for the ORC cycle under steady state flow condition can be expressed as:

$$\dot{I}_{Total} = \dot{m}T_0[\frac{h_4 - h_1}{T_H} + \frac{h_2 - h_3}{T_L}] \tag{2}$$

The availability ratio for the ORC cycle in steady state flow condition can be expressed as:

$$\Phi = (\dot{Q}_b - \dot{I}_{Total}) / \dot{Q}_b \tag{3}$$

The system efficiency of ORC is expressed as:

$$\eta_{Thermal} = (\dot{W}_t - \dot{W}_p) / \dot{Q}_b \tag{4}$$

The second law efficiency of the ORC is expressed as:

$$\eta_{Second} = \dot{W}_t / (\dot{W}_t + \dot{I}_{total}) \tag{5}$$

TABLE IV
ENERGY BALANCE FOR ORC SYSTEM

| Thermodynamic Process | ORC cycle component | Energy balance equations | Equation Number |
|---|---|---|---|
| Process, 4-1 | Shell and tube heat exchanger (E-100) | $Q_{E-100} = m(h_1 - h_4)$ | (6) |
| Process, 1-2 | Turbo expander (K-100) | $W_T = m(h_1 - h_2)$ | (7) |
| Process, 2-3 | Condenser(E-101) | $Q_{E-101} = m(h_2 - h_3)$ | (8) |
| Process, 3-4 | Working Fluid Pump (P-100) | $W_p = m(h_3 - h_4)$ | (9) |

*E. Regenerative Cycle Energy Balance Equation*

The system efficiency of RORC is expressed as:

$$\eta_{Thermal} = (\dot{W}_t - \dot{W}_p) / \dot{Q}_b \tag{10}$$

The Irreversibility rate for the regenerative cycle in steady state flow condition can be expressed as:

$$\dot{I}_{Total} = \dot{m}T_0[-\frac{h_1 - h_{4*}}{T_H} + \frac{h_{2*} - h_3}{T_L}] \tag{11}$$

The Irreversibility ratio for the RORC cycle in steady state flow condition can be expressed as:

$$\Phi = \dot{I}_{Total} / \dot{Q}_b \tag{12}$$

The second law efficiency of the RORC is expressed as:

$$\eta_{Second} = \dot{W}_t / (\dot{W}_t + \dot{I}_{total}) \tag{13}$$

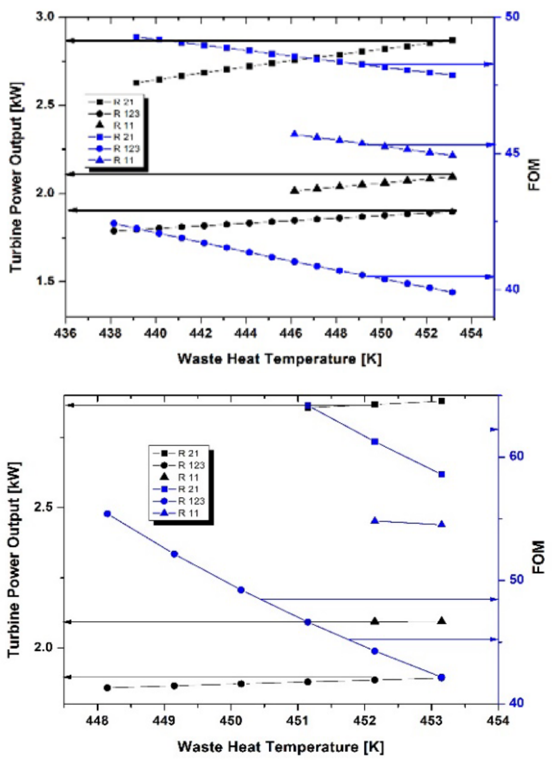
TABLE V
ENERGY BALANCE EQUATION FOR RORC SYSTEM

| Thermodynamic Process | ORC cycle component | Energy balance equations | Equation Number |
|---|---|---|---|
| Process, 4-1 | Shell and tube heat exchanger (E-100) | $Q_{E-100} = \dot{m}(h_1 - h_4)$ | (14) |
| Process, 1-2 | Turbo expander (K-100). | $W_T = \dot{m}(h_1 - h_2)$ | (15) |
| Process, 2-3 | Condenser (E-102) | $Q_{E-101} = \dot{m}(h_4 - h_3)$ | (16) |
| Process, (4-4* & 2-2*) | Regenerator (E-101) | $(h_2 - h_{2^*}) = -(h_4 - h_{4^*})$ | (17) |
| Process, 3-4 | Working Fluid Pump (P-100) | $W_p = m(h_3 - h_4)$ | (18) |

## IV. RESULT AND DISCUSSION

### A. The Net Power Output in Turbine and FOM

This section focuses on determining the suitable power cycle and working fluid through a comparative analysis of two-performance parameters viz. turbine power output and FOM within the range of WHT reported in Table I, using (7) and (15). Fig. 3 (a) shows that the fluids except R 11 are capable to generate power within the total considered WHT range using ORC. However, the fluids are suitable only in the higher range of WHT in case of RORC in Fig. 3 (b). At prescribed WHT range, lower FOM physically indicates higher thermal efficiency. The FOM of the fluid R 21 decreases with WHT and its power output gradually increases with WHT as shown in Fig. 3 (a). Hence, the fluid R 21 in ORC system seems to be the most suitable combination for the data reported in Table I. The FOM is define as (FOM= $\eta_{Thermal} / (1 - T_H / T_L)$).




a

b

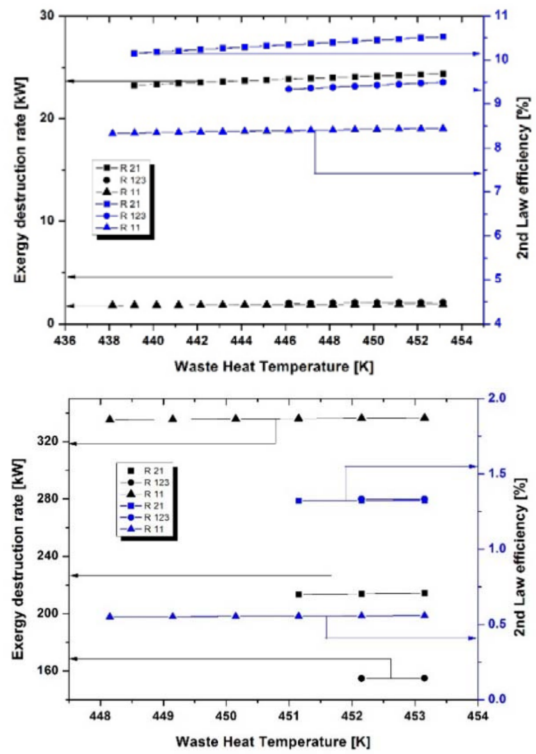
Fig. 3 Variations of turbine work output and FOM with WHT using selected working fluids in (a) ORC, (b) RORC

### B. Exergy Destruction Rate and Second Law Efficiency

Figs. 4 (a) and (b) show the difference of exergy destruction rate and second law efficiency obtained from (2) and (11) with increase in WHT for the considered ORC and RORC using the selected working fluids. The system exergy destruction rate also depends upon the temperatures of waste heat source. As compare to ORC the RORC has extra regenerator, the improvement in efficiency and irreversibility are lower than the ORC cycle. It has been concluded that the regenerative cycle seems to be the most appropriate cycle for the waste heat recovery power generation in higher temperature ranges only. It can be seen from Fig. 4 (a) that the second law efficiency increases linearly with increase in WHT. It is primarily due to the variations in enthalpies at turbine inlet/outlet and is also based on the turbine work output as well as irreversibility rate as shown in Fig. 4 (b). The maximum second law efficiency has been achieved in ORC cycle in this study.

### C. Exergy Efficiency of Cycle

Figs. 5 (a) and (b) show the effect of WHT on the exergy efficiency. A mathematical model has been developed using exergy energy analysis with four different components of each of ORC and RORC cycles. The fluid R21 is found to be suitable in combination of ORC approach, in terms of several beneficial factors like lower FOM, higher power output and higher exergy efficiency. In the due course, it has been observed that the most crucial component in small scale ORC system is turbine which has been studied in detail in the succeeding section.




a

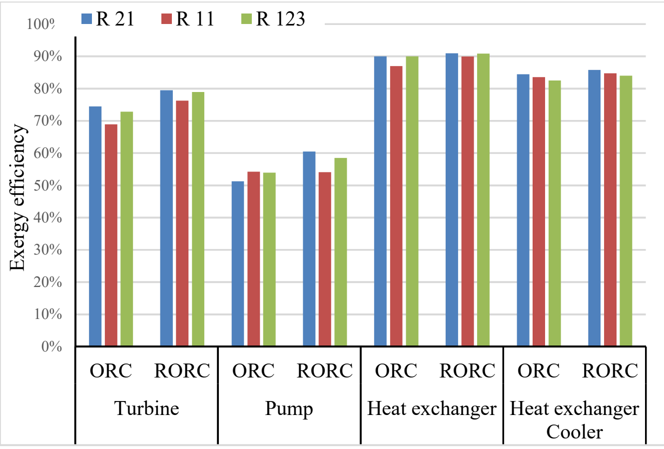
b

Fig. 4 Variations of exergy destruction rate and of 2nd law of efficiency with WHT using selected working fluids in (a) ORC, (b) RORC

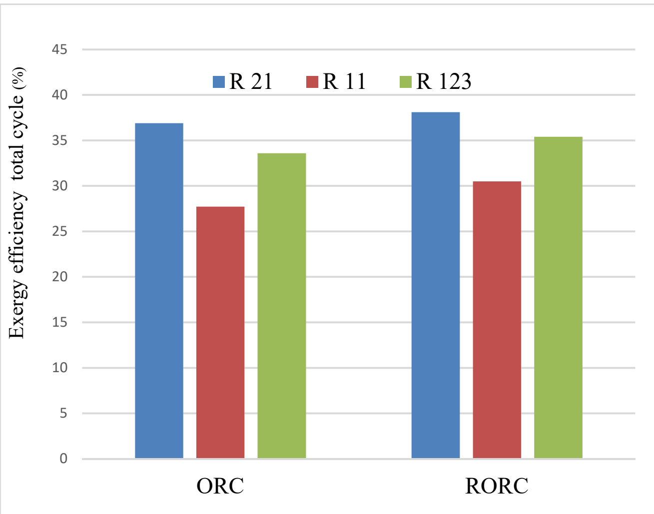
### *D. Design of Turbine Wheel*

In this section, a detailed design of the turbine wheel for each ORC cycle has been obtained. The turbine wheel blade was designed based on the methodologies proposed by Kun and Sentz [10] and Balje [11]. The operating conditions and the design parameters used in the turbine design methodology is shown in Table VI. $K_1$ and $K_2$ are the free parameters that controls the blade geometry. To compare the performance, the mass flow rate and the static pressure at the exit of the turbine wheel was kept constant for all the cycles.

Table VII shows the major geometrical parameters of the turbine wheel for the different ORC cycles. The major parameters of a turbine system include the rotational speed (N), the diameters of the wheel at inlet (D), the hub ($D_{hub}$) and tip ($D_{tip}$) at the exit. Table VII shows that with change in fluid there is not much difference in the wheel diameter and the rotational speed but significant difference in the power produced. It may be observed that R21 produces the maximum power.




(a)




(b)

Fig. 5 (a) Exergy efficiency of each component of the ORC and RORC cycle @ 180 °C (b) Cycle exergy efficiency of ORC and RORC cycle @ 180 °C

TABLE VI
PARAMETER USING FOR DESIGNING OF ORC TURBO EXPANDER.

| Working Fluid | R 123 | R 11 | R 21 | Design Parameter | R11 | R21 | R123 |
|---|---|---|---|---|---|---|---|
| Total pressure at inlet (kPa) | 790 | 2825 | 3975 | Specific speed | 0.55 | 0.55 | 0.55 |
| Total Temperature at inlet (K) | 373.8 | 449.0 | 449.0 | Specific diameter | 3.7 | 3.7 | 3.7 |
| Static pressure at exit (kPa) | 160 | 160 | 160 | $K_1$ | 1.02 | 1.02 | 1.02 |
| Mass flow rate (Kg/s) | 0.051 | 0.051 | 0.051 | $K_2$ | 1.03 | 1.03 | 1.03 |
| Expected efficiency (%) | 75 | 75 | 75 | ε | 1.5 | 1.5 | 1.5 |

TABLE VII
MAJOR GEOMETRIC PARAMETER OF THE PROTOTYPE TURBINE WHEEL

| Geometric parameter | R11 | R21 | R123 |
|---|---|---|---|
| Number of rotor blades | 7 | 7 | 7 |
| Wheel diameter | 19.72 mm | 19.8 mm | 20.49 mm |
| Wheel tip diameter | 13.6 mm | 13.65 mm | 14.13 mm |
| Wheel hub diameter | 4.76 | 4.78 mm | 4.946 mm |
| Rotational speed | 24497 rad/s | 28443 rad/s | 22322 rad/s |
| Tip speed at inlet | 241.5 m/s | 281.6 | 228.7 |
| Velocity ratio | 0.7302 | 0.7303 | 0.7302 |
| Power produced | 2.109 kW | 2.865 kW | 1.89 kW |
| Blade thickness | 1 mm | 1 mm | 1 mm |
| Volumetric flow rate at turbine wheel exit | 0.006743 m3/s | 0.007552 | 0.006892 |
| Mean relative velocity angle at turbine wheel exit | 37.88 | 36.89 | 36.46 |

## V. CONCLUSION

A thermodynamic investigation into regenerative Rankine cycle employing a novel integrated system for enhanced waste heat recovery is undertaken. Results show significant prospects for heat recovery toward enhanced work output. The low waste heat recovery technology has significant potential to be integrated in the Indian energy sector. In this report, the different technological aspects and corresponding challenges are analysed. Both the ORC and the RORC have been simulated using the Aspen Hysys®, v8.6 simulators. Analysis with various working fluids has also been carried out to compare multiple characteristic parameters like efficiency, volumetric flow rate and irreversibility. The thermodynamic analysis and the turbine design indicate that the ORC cycle is the most suitable technique for the recovery of the low waste heat in this case study and the R 21 seems to be the most suitable fluid. The challenges regarding the ORC system have been identified. Further investigation is yet to be carried out in this field in due course of time.

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