Tuijin Jishu/Journal of Propulsion Technology
ISSN: 1001-4055
Vol. 45 No. 3 (2024)

Parametric Optimization, Modelling and Performance Evaluation of Organic Rankine Cycle (ORC) as a Technological Option for Sustainable Waste Heat Recovery from Process Industries

Sandip B. Patil[1], Samir J. Deshmukh[2]

[1]Ph.D. Research Scholar, Anuradha Engineering College, Chikhali, SGBA University, Amravati, India.
[2]Prof., Department of Mechanical Engg., PRM Institute of Technology and Research, Badnera, India.

Abstract: - This paper presents parametric optimization, modelling and performance evaluation of Organic Rankine Cycle for sustainable waste heat recovery from process industry. Three different working fluids R134a, R152a and R245fa have been used for evaluation of ORC system performance. Organic Rankine Cycle model has been developed in MATLAB Simulink to carry out parametric optimization for selected working fluids. The inlet turbine pressure optimization has been carried out for maximum output work and system efficiencies considering saturated vapours at turbine inlet and constant pressure superheating at various pressures for chosen working fluids. The waste heat data of flue gas stream from selected process industry has been utilized for comparative analysis of the cycle performance using optimized parameters for three different working fluids under consideration. The results are examined for the waste heat flue gas at 177°C as heat source. The results demonstrate that among all selected working fluids, R245fa shows better performance and produces highest output work with maximum system efficiencies as compared to R134a and R152a. The power generated was 340.18 kW with system efficiency of 20.17% and second law efficiency of 58.15% having pinch point of 6.42°C with heat source flue gas. Thus, Organic Rankine Cycle using working fluid as R245fa seems to be a sensible approach for sustainable waste heat recovery and power generation from selected process industry.

Keywords: *Modelling, Parametric Optimization, Performance Evaluation, Waste heat recovery, Organic Rankine Cycle (ORC).*

1. Introduction

There is a great deal of potential for the world's industrial waste heat to meet some of its electricity needs. The inability of current power generation systems to convert low-quality heat into electrical power results in a significant loss of heat. The contribution of this industrial low-grade waste heat is approximately half or more of the total heat input. The direct release of low-grade waste heat energy into the atmosphere by industries leads to thermal pollution and serious environmental issues related to increase in greenhouse gas emissions. Reusing this low-grade waste heat (90 °C – 400 °C) is one of the biggest issues in power generation since plant efficiency drops when exhaust gas temperature falls below 350 ºC [1]. Several techniques, like steam Rankine cycles and Stirling engines, can be used to transform thermal energy into electrical energy. However, lower efficiency of these systems for low temperature (<400 ºC) heat supply makes them technically impractical [2]. Implementing waste heat recovery systems promotes energy conservation and lowers thermal pollution. So, it is of utmost importance to emphasize on developing an environment friendly energy conversion technology that generates electricity without pollution. As a result of the energy crisis and the progressive depletion of fossil fuels, many researchers, industrial and government sectors are now concentrating on low-grade energy recovery technologies, with an emphasis on Organic Rankine Cycle technology [3]. The process industries like glass, cement, iron-steel and many more has found commercial application of ORC technology for power generation from low-temperature waste heat [4]. The Organic Rankine cycle (ORC), in contrast to a typical power plant, uses hydrocarbons and refrigerants as its working fluid rather than water [5]. The system operation and efficiency greatly affected by thermodynamic properties of working fluids. Thus, the selection of working fluid and operating conditions plays a vital role in evaluating the system performance.

Depending on the slope of saturation vapour curve in temperature-entropy diagram (T-S curve), the ORC working fluids are classified in three different types, wet, dry and isentropic fluids. The wet fluids have negative slope, dry fluids have positive slope and fluids which have nearly infinite slope of T-S curve falls in the category of isentropic fluids. Based on the various shapes and slopes of the saturation vapour curves of the fluids, the ORC using working fluids R134a, R113, R11 and R12 shows the isentropic fluids are more appropriate for low temperature waste heat recovery using ORC [6]. An investigation of how working fluids affects the performance of organic Rankine cycles and their impact on thermal efficiency and total heat recovery efficiency indicate that the maximum value of total heat recovery efficiency increases as the waste heat source's inlet temperature rises and decreases when working fluids with lower critical temperatures are used [7]. Comparing the performance of an ORC with R123 and isobutene as working fluids, R123 shows greater thermal efficiency than isobutene [8]. An ORC system utilizing low grade energy sources to investigate the effects of several organic working fluids on system efficiency revels that as compared to dry fluids, wet fluids having extremely steep saturated vapour curves in the T-s plot perform better in terms of energy conversion efficiencies [9]. The thermodynamic and physical characteristics of certain non-conventional fluids that could be used in Organic Rankine Cycle (ORC) applications have been studied [10]. An investigation of ORC performance analysis and optimization for waste heat recovery utilizing HFC-245fa as the working fluid powered by exhaust heat concluded that increasing the use of exhaust heat can enhance the output power and efficiency of the system [11]. Using the BACKONE equation of state as a baseline, the thermodynamic performances of thirty-one pure working fluids for organic Rankine cycles have been investigated and made the observation that in subcritical processes using an internal heat exchanger, the maximum efficiency values are achieved for high boiling substances with an overhanging saturated vapour line. Also, the greatest amount of heat may be transported to a supercritical fluid and the least to a high-boiling subcritical fluid, according to a pinch analysis of the heat transfer from the heat carrier to the working fluid [12]. As compared to basic ORC, the regenerative ORC employing dry fluids, R113, R245Ca, R123, and isobutane offers the maximum thermal efficiency with the lowest level of irreversibility [13]. The impact of thermodynamic parameters on the Organic Rankine Cycle (ORC) performance implementing the genetic algorithm to optimize the thermodynamic parameters of the ORC for different working fluids, with the purpose of maximizing exergy efficiency indicate that the cycle utilizing R236EA exhibits the maximum exergy efficiency [14]. The experimental results reveal that the turbine, when tested with R-123 working fluid, performs well for saturated vapours at the turbine [15]. The comparative analysis of R125 and CO2 transcritical cycle for low grade heat source when optimized for power output shows that the cycle with R125 generates more power than the CO2 transcritical cycle [16]. Low-temperature solar organic Rankine cycle systems using fluids like R600a, R600, R290, R134a and R152a have been investigated to compare their theoretical performances along with thermodynamic and environmental characteristics [17].

A prototype low temperature Rankine cycle system with n-pentane as working fluid was developed and designed to produce electrical energy [18]. Research on fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants indicates that the alkyl benzenes family exhibits the highest efficiency [19]. Supercritical

Organic Rankine Cycles shows favourable results in terms of system efficiency when investigated for efficiency optimization [20]. As per the thermodynamic analysis, there are two ways to make a waste heat power generating system more efficient; one is to increase the heat/exergy input, and the other is to make it better at converting heat into work [21]. For decentralized electric energy generation, a small-scale Organic Rankine Cycle engine having good thermodynamic efficiency seems to be more appropriate technology [22].

Based on the brief review presented above, selection of proper working fluid for an Organic Rankine Cycle system is of utmost importance for maximum recovery of waste heat from thermal power plants and industrial waste heat streams. The prime objective of the present work is focused on parametric optimization, modelling and performance evaluation of Organic Rankine Cycle technology for sustainable waste heat recovery in order to identify the better working fluid which in turn gives best ORC performance. The ORC system is modelled in MATLAB Simulink environment and parametric optimization and performance analysis using working fluids R134a, R152a and R245fa have been investigated in the present work. The results are applied for the data of selected process industry and presented.

2. ystem Details and Theoretical Analysis of ORC System

2.1 System Details

The ideal Rankine cycle and basic Organic Rankine Cycle (ORC) works on same principle, the difference lies between the use of working fluids. ORC uses the organic fluids as working medium. These fluids have better thermodynamic properties to utilize low grade energy for conversion to high grade energy if used with ORC systems. The basic Organic Rankine Cycle system comprises of four main components; condenser, evaporator, pump and turbine or expander as shown in Fig. 1. The working fluid alternately changes its phase when circulated through the components of ORC system. The heat absorption and rejection by and from working fluid takes place in evaporator and condenser respectively.

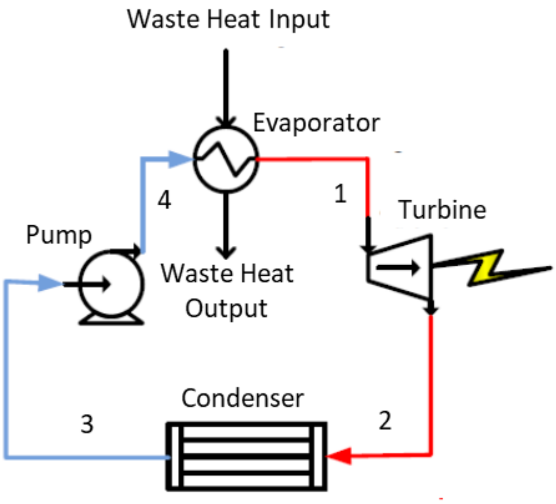

Fig. 1: Components of Basic Organic Rankine Cycle System

2.2 Thermodynamic Processes of ORC

The working fluid circulating through different components of ORC system has to undergo four important thermodynamic processes to complete the cycle. The cycle is plotted on Temperature-Entropy diagram and shown in Fig. 2 and Fig. 3 for two different conditions of turbine inlet.

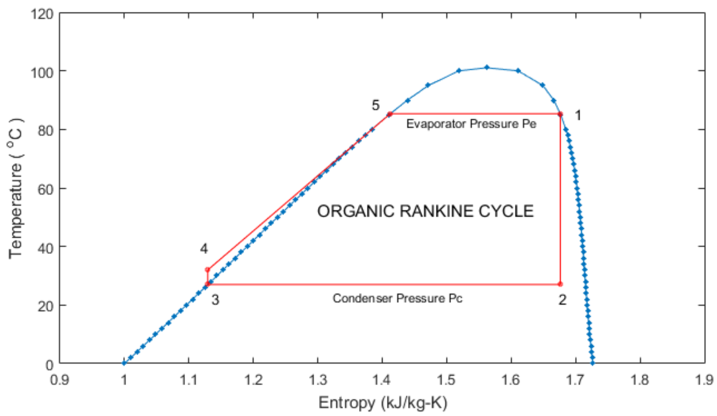

Fig. 2: T-s Plot of ORC (Saturated Vapours at Turbine Inlet)

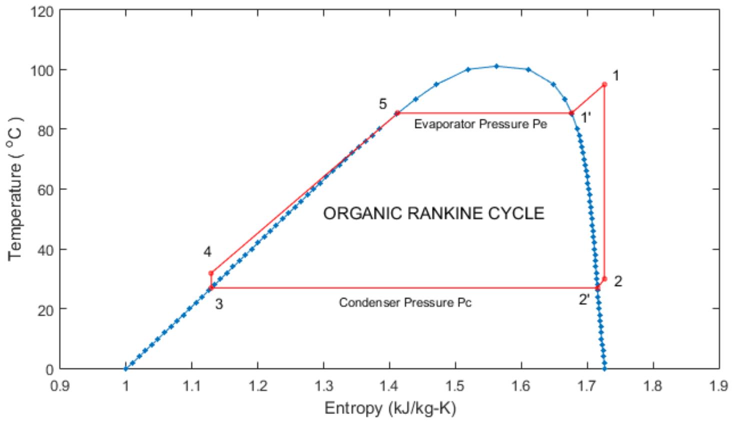

Fig. 3: T-s Plot of ORC (Superheated Vapours at Turbine Inlet)

Process (1-2): After absorbing the heat from the source in evaporator, the saturated or superheated vapours of working fluid are expanded in turbine or expander from high evaporator pressure to low condenser pressure. The expansion process is assumed to be isentropic. In this process required work is produced by the turbine.

Process (2-3): The condition of low-pressure vapours leaving the expander or turbine may be unsaturated, saturated or superheated depending on turbine inlet pressure and quality. These low-pressure vapours are entered into condenser where heat in the working fluid is removed and rejected to cooling medium leaving the condenser in the form of saturated liquid. If working fluid leaves the turbine as superheated vapours, the process will be subdivided into two parts; de-superheating (2-2') and condensation (2'-3).

Process (3-4): The saturated liquid from condenser is passed through pump in order to raise the pressure to evaporator pressure. The working fluid is compressed isentropically raising its temperature to small extent. In this process small amount of work is consumed by the pump. This is an isentropic compression process in which the working fluid leaving the condenser as saturated liquid is pumped to evaporator pressure.

Process (4-1): This is a heat addition process at constant evaporator pressure where working fluid absorbs the heat from high temperature heat source (Flue gas waste heat from process industry). Depending on heat transfer in evaporator, the state of working fluid may be saturated or superheated vapours at the exit. The process is subdivided into three parts; preheating (4-5), evaporation (5-1') and superheating (1'-1).

2.3 Theoretical Analysis of ORC System

Referring to Temperature-Entropy plot of ORC system shown in Fig. 3, theoretical equations for different thermodynamic processes involved for each component can be expressed as;

Turbine (Process 1-2):

Turbine Work generated;

$$W_{Turbine} = \dot{m}_{wf}(h_1 - h_2) \quad \ldots \text{(Eq. 1)}$$

Exergy destruction rate in Turbine;

$$I_{Turbine} = \dot{m}_{wf}T_0(s_2 - s_1) \quad \ldots \text{(Eq. 2)}$$

Since expansion process in turbine is assumed to be isentropic $s_1 = s_2$, Equation (Eq.2) reduces to;

$$\dot{I}_{Turbine} = 0 \quad \ldots \text{(Eq. 3)}$$

Condenser (Process 2-3):

This is constant pressure heat rejection process in which working fluid rejects its heat to cooling medium at cold source temperature T_L

Condenser Heat rejected is given by;

$$Q_{condenser} = \dot{m}_{wf}(h_2 - h_3) \quad \ldots \text{(Eq. 4)}$$

Condenser Exergy destruction rate is given by;

$$\dot{I}_{condenser} = \dot{m}_{wf}T_0\left[(s_3 - s_2) + \frac{(h_2 - h_3)}{T_L}\right] \quad \ldots \text{(Eq. 5)}$$

Pump (Process 3-4):

Work consumed by Pump;

$$W_{Pump} = \dot{m}_{wf}(h_3 - h_4) \quad \ldots \text{(Eq. 6)}$$

Exergy destruction rate in Pump;

$$\dot{I}_{Pump} = \dot{m}_{wf}T_0(s_4 - s_3) \quad \ldots \text{(Eq. 7)}$$

Since Pumping process in Pump is assumed to be isentropic $s_3 = s_4$, Equation (Eq.7) reduces to;

$$\dot{I}_{Pump} = 0 \quad \ldots \text{(Eq. 8)}$$

Evaporator (Process 4-1):

This is constant pressure process in which working fluid is heated by constant temperature heat source (Flue gases from Industry) at T_H

Heat transferred to working fluid by thermal source in evaporator is given by;

$$Q_{evaporator} = \dot{m}_{wf}(h_1 - h_4) \quad \ldots \text{(Eq. 9)}$$

Evaporator Exergy destruction rate is given by;

$$\dot{I}_{evaporator} = \dot{m}_{wf}T_0\left[(s_1 - s_4) + \frac{(h_4 - h_1)}{T_H}\right] \quad \ldots \text{(Eq. 10)}$$

System Total rate of Exergy destruction can be determined by combining Equations (Eq.3), (Eq.5), (Eq.8) and (Eq.10)

$$\dot{I}_{Total} = \dot{I}_{Pump} + \dot{I}_{Turbine} + \dot{I}_{evaporator} + \dot{I}_{condenser}$$

$$\dot{I}_{Total} = \dot{m}_{wf}T_0\left[(s_3 - s_2) + (s_1 - s_4) - \frac{(h_1 - h_4)}{T_H} + \frac{(h_2 - h_3)}{T_L}\right]$$

Since Pumping and Expansion Processes are assumed to be isentropic,

$s_1 = s_2$ and $s_3 = s_4$, Total Exergy destruction rate $\dot{I}_{Total}$ can be written as;

$$\dot{I}_{Total} = \dot{m}_{wf} T_0 \left[-\frac{(h_1 - h_4)}{T_H} + \frac{(h_2 - h_3)}{T_L} \right] \quad \text{… (Eq. 11)}$$

System Efficiency:

The pump work is neglected in calculation of System Efficiency because as compared to Turbine work it is very small.

From Eq. (Eq.1) and (Eq.9), The System Efficiency is given by;

$$\eta_I = \frac{W_{Turbine}}{Q_{evaporator}} \quad \text{… (Eq. 12)}$$

Second Law Efficiency:

Second law efficiency is calculated based on irreversibility or exergy destructions associated with different processes within the system, and expressed as;

$$\eta_{II} = \frac{W_{Turbine}}{(W_{Turbine} + \dot{I}_{Total})} \quad \text{… (Eq. 13)}$$

3. Modelling of ORC System

Modelling of Organic Rankine Cycle (ORC) System was carried out in MATLAB/Simulink using detailed thermodynamic analysis explained in sections 2.2 and 2.3 for three different working fluids R152a, R134a and R245fa. The different subsystem models were developed and combined together to form a complete model of ORC system for each working fluid as shown in Fig. 4(a). The subsystem models shown in Fig. 4(b) comprise of models for saturated and superheated properties of different state points at condenser and evaporator pressures and temperatures. It also includes the models for estimating turbine exit quality, performance output parameters, mass flow rates and pinch point temperature difference based on thermodynamic analysis. The ORC model developed in MATLAB/Simulink was used to perform the parametric optimization of the system.

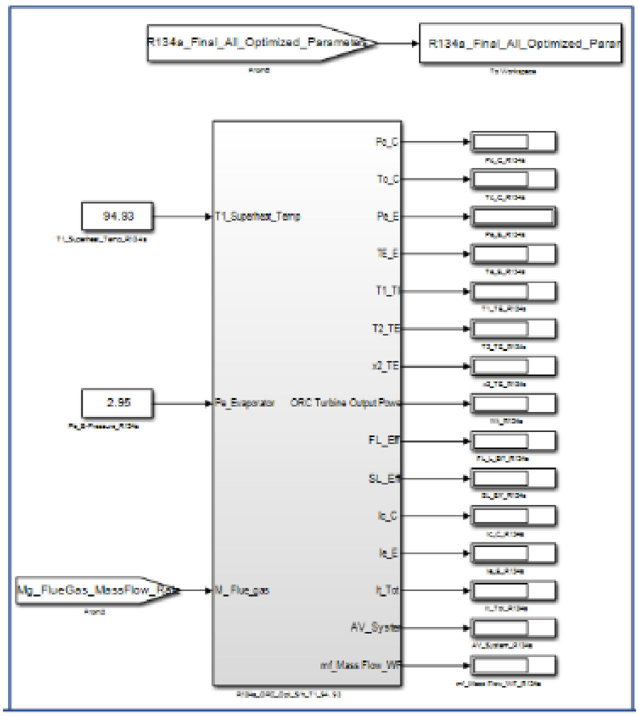

Fig. 4(a): Overall ORC System Model

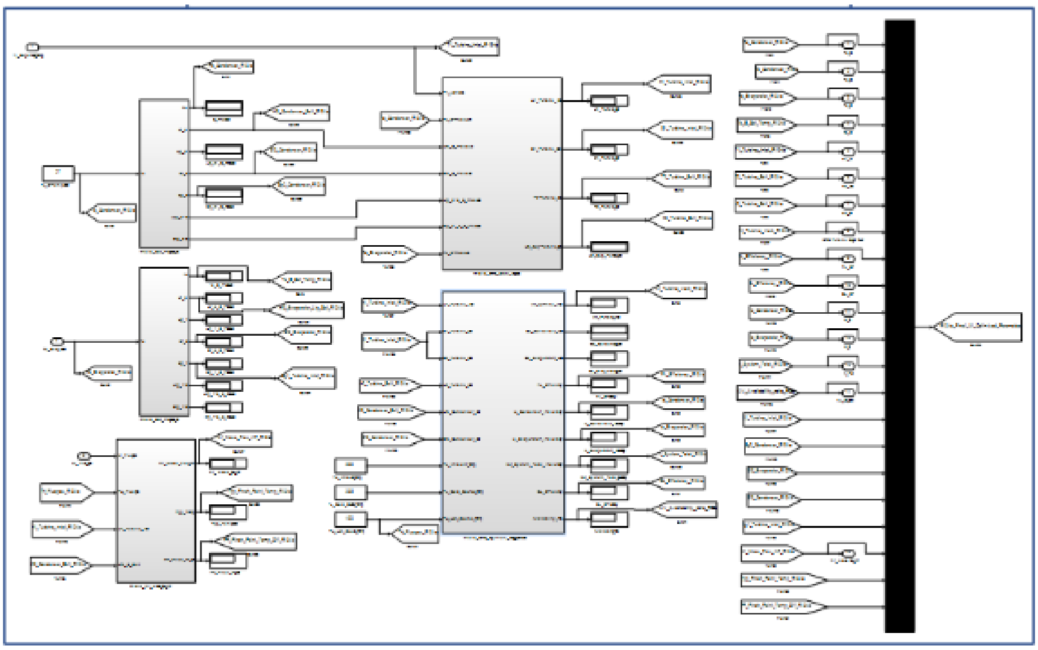

Fig. 4(b): ORC Subsystem Model

4. Parametric Optimization and Simulation of ORC System

The MATLAB Simulink models developed for selected working fluids were used to carry out the parametric optimization of ORC system. The optimization study was conducted in three different stages for each working fluid. The mass flow rate of 1 kg/s was considered for all stages of optimization. The models were simulated to record the optimized parameters to be used for performance evaluation of ORC system.

- Stage-I: ORC System – TIP Optimization (Turbine Inlet - Saturated)
- Stage-II: ORC System – TIP Optimization (Turbine Inlet - Superheated)
- Stage-III: ORC System – TIT Optimization (Optimum - TIP)

4.1 Stage-I: ORC System – TIP Optimization (Turbine Inlet – Saturated)

In this stage of optimization, the overall ORC system model was updated for input variation of inlet turbine pressure (P_e), in order to estimate optimal pressure at turbine inlet (TIP) for maximum output work from turbine (W_{T_Max}) and maximum system efficiency (η_{I_Max}). The turbine inlet condition was assumed as saturated vapours. The Fig. 5 shows the updated model.

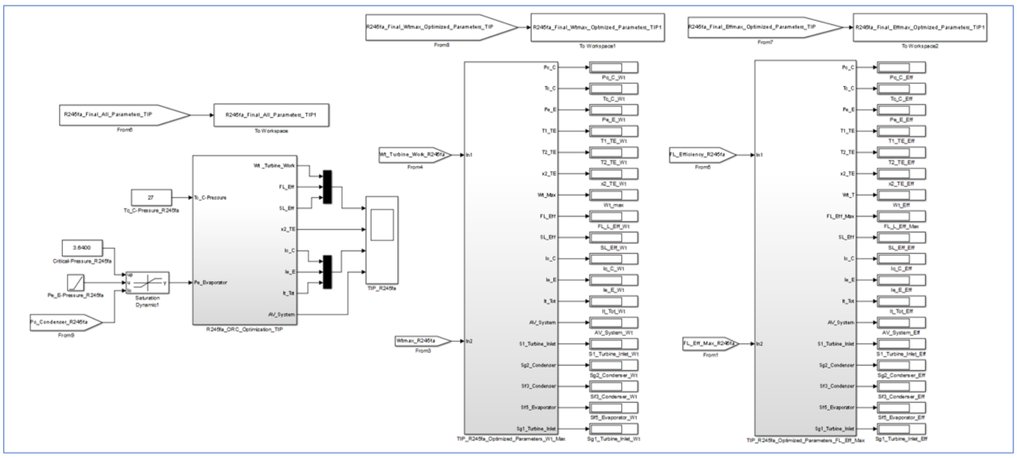

Fig. 5: ORC System Model – TIP Optimization (Turbine Inlet – Saturated)

The Fig. 6 shows the flow chart of detailed methodology adopted for stage-I TIP-Optimization.

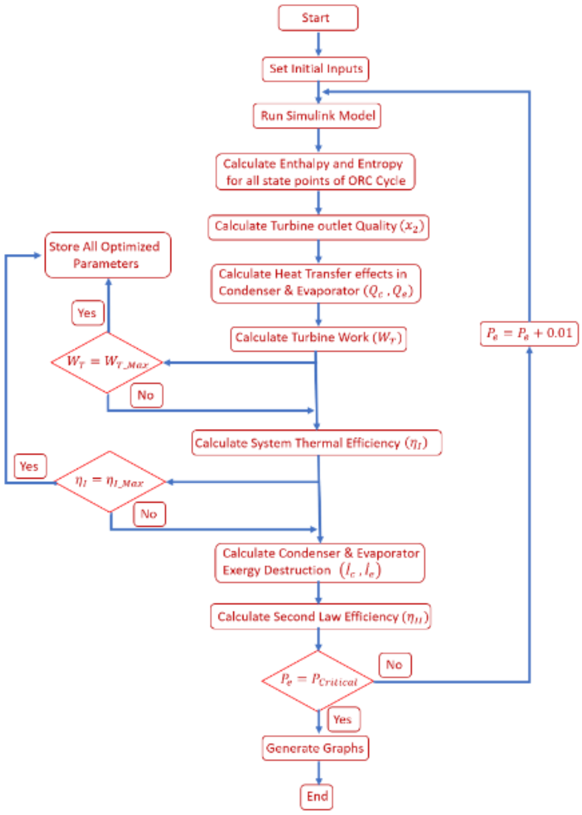

Fig. 6: Flow Chart - TIP Optimization (Turbine Inlet- Saturated)

Condenser temperature and simulation incremental step of pressure at turbine inlet were set as initial conditions. In order to keep the condenser pressure above atmospheric, the condenser temperature was assumed and set at 27°C for all selected working fluids as per the thermodynamic properties of working fluids selected. From the condenser pressure corresponding to selected condenser temperature of 27°C, the inlet turbine pressure was increased till critical pressure of working fluid with an incremental step of 0.01MPa. Assuming the isentropic expansion in turbine and turbine inlet condition as saturated vapours, the simulated parameters were recorded to obtained maximum output work from turbine (W_{T_Max}) and maximum system efficiency (η_{I_Max}). As per the optimization conditions, the model was simulated and results of simulation were tabulated for each selected ORC working fluid R152a, R134a and R245fa [Table. 1].

Table 1: Simulated Parameters-TIP Optimization (Turbine Inlet- Saturated) [R134a, R152a, R245fa]

Selected Working Fluid ➔	**R134a**		**R152a**		**R245fa**	
Properties (Optimization Condition) ➔	W_{T_Max}	η_{I_Max}	W_{T_Max}	η_{I_Max}	W_{T_Max}	η_{I_Max}
Condenser Pressure P_c (MPa)	0.7062	0.7062	0.6328	0.6328	0.1609	0.1609
Condenser Temperature T_c (°C)	27.00	27.00	27.00	27.00	27.00	27.00
Turbine Inlet Pressure [TIP] P_e (MPa)	3.586	3.966	3.513	4.243	3.091	3.381
Turbine Inlet Temp. [TIT] T_1 (°C)	94.93	99.92	100.11	109.99	145.06	150.01
Turbine Exit Temp. [TET] T_2 (°C)	27.00	27.00	27.00	27.00	41.66	37.87
Turbine Exit Quality x_2	0.8872	0.8225	0.8739	0.8084	1.0710	1.0526
Turbine Work Output W_T (kJ/kg)	27.24	25.77	46.75	46.12	53.32	53.17
System Efficiency η_I (%)	14.86	15.12	16.19	17.08	20.83	21.08
Second Law Efficiency η_{II} (%)	42.72	43.48	46.59	49.18	60.07	60.81
Mass Flow Rate $\dot{m}_{wf}$ (kg/s)	1.00	1.00	1.00	1.00	1.00	1.00

4.2 Stage-II: ORC System –TIP Optimization (Turbine Inlet–Superheated)

In this stage of optimization, the overall ORC system model was updated for input variation of inlet turbine pressure (P_e), in order to investigate the effects with superheating of turbine inlet vapours and to estimate revised optimal pressure at turbine inlet (TIP) for maximum output work from turbine (W_{T_Max}) and to ensure dry turbine exit quality (x2). This was achieved by superheating of turbine inlet vapours up to saturation temperature corresponding to optimal TIP obtained in previous stage-I optimization. The turbine inlet condition was superheated vapours. The Fig. 7 shows the updated model.

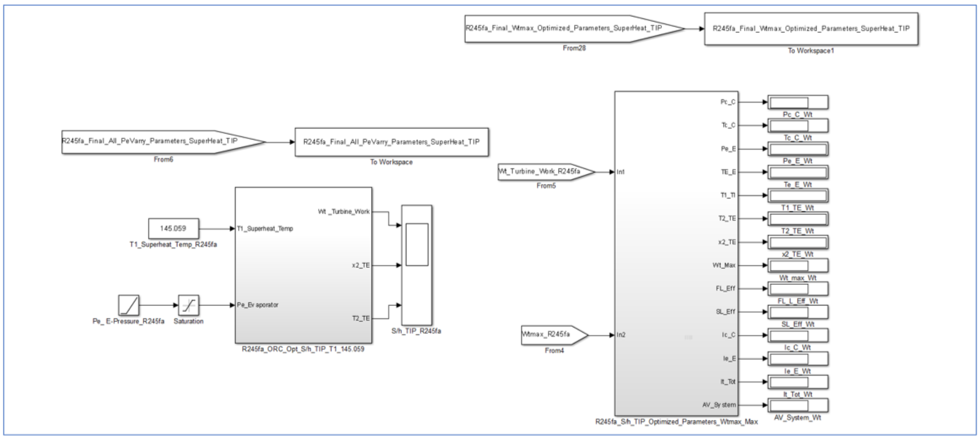

Fig. 7: ORC System Model – TIP Optimization (Turbine Inlet – Superheated)

The Fig. 8 shows the flow chart of detailed methodology adopted for stage-II TIP-Optimization.

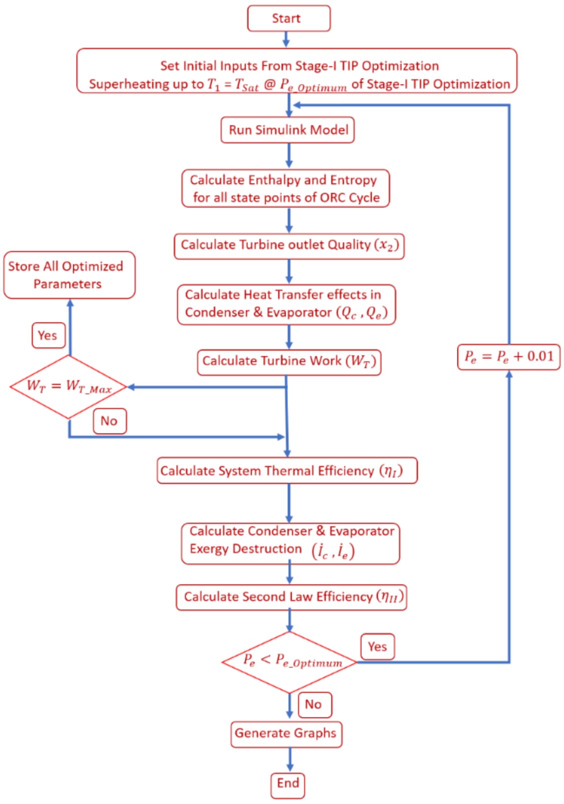

Fig. 8: Flow Chart - TIP Optimization (Turbine Inlet- Superheated Vapour)

The initial conditions related to Condenser temperature and simulation incremental step of pressure at turbine inlet were set same as that of stage-I optimization. For analyzing the effect with superheating of turbine inlet vapours, the turbine inlet temperature was set equal to saturation temperature corresponding to optimum TIP of previous stage-I optimization for maximum output work as per the thermodynamic properties of working fluids selected. From the set condenser pressure corresponding to selected condenser temperature of 27°C, the inlet turbine pressure was increased till optimum pressure of Stage-I TIP optimization of working fluid with an incremental step of 0.01MPa. Assuming the isentropic expansion in turbine and turbine inlet condition as superheated vapours, the simulation was carried out for maximum output work from turbine (W_{T_Max}) and dry turbine exit quality (x2). As per the optimization conditions, the model was simulated and results of simulation were tabulated for each selected ORC working fluid R152a, R134a and R245fa [Table 2].

Table 2: Simulated Parameters-TIP Optimization (Turbine Inlet- Superheated) [R134a, R152a, R245fa]

Selected Working Fluid ➔	**R134a**	**R152a**	**R245fa**
Properties (Optimization Condition) ➔	W_{T_Max}	W_{T_Max}	W_{T_Max}
Condenser Pressure P_c (MPa)	0.7062	0.6328	0.1609
Condenser Temperature T_c (°C)	27.00	27.00	27.00
Turbine Inlet Pressure [TIP] P_e (MPa)	2.95	2.66	2.50
Turbine Inlet Temp. [TIT] T_1 (°C)	94.93	100.11	145.06
Turbine Exit Temp. [TET] T_2 (°C)	29.87	28.46	58.03
Turbine Exit Quality x_2	1.0177	1.0006	1.1483
Turbine Work Output W_T (kJ/kg)	29.44	47.89	55.18
System Efficiency η_I (%)	14.13	14.77	20.17
Second Law Efficiency η_{II} (%)	40.63	42.46	58.15
Mass Flow Rate $\dot{m}_{wf}$ (kg/s)	1.00	1.00	1.00

4.3 Stage-III: ORC System –TIT Optimization (Optimum - TIP)

In this stage of optimization, the overall ORC system model was updated for input variation of inlet turbine temperature (T_1), in order to investigate the effects with superheating of turbine inlet vapours and to estimate optimal superheating temperature at turbine inlet (TIT) for maximum system efficiency (η_{I_Max}). This was achieved by superheating of turbine inlet vapours from saturation temperature corresponding to optimal TIP obtained in previous stage-II optimization up to available waste heat source temperature (T_H). The turbine inlet pressure was kept constant. The Fig. 9 shows the updated model.

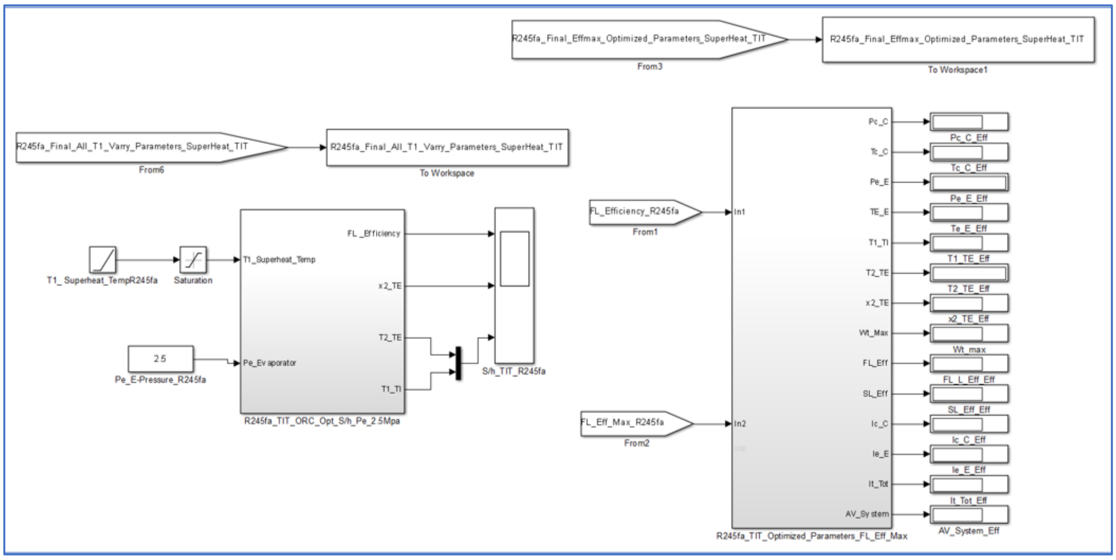

Fig. 9: ORC System Model – TIT Optimization (Optimum-TIP)

The Fig. 10 shows the flow chart of detailed methodology adopted for stage-III TIT-Optimization.

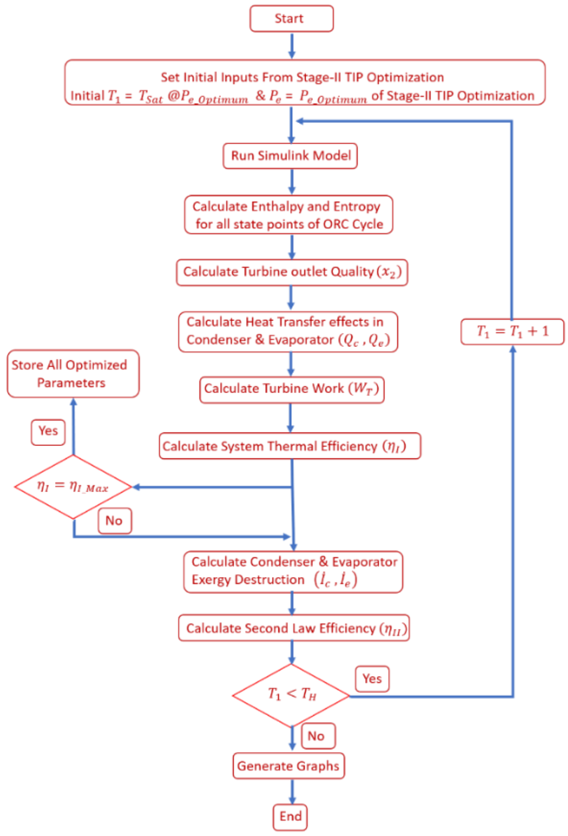

Fig. 10: Flow Chart - TIT Optimization (Optimum TIP)

The initial inputs for TIT optimization include setting of constant evaporator or inlet turbine pressure (P_e) equal to optimal TIP ($P_{e\_optimum}$) found in stage-II optimization and initial superheating temperature (T_1) equal to saturation temperature corresponding to optimal TIP selected. For analyzing the effect with superheating of turbine inlet vapours and to estimate optimum superheating temperature, from the set initial conditions the inlet turbine temperature (TIT) was increased till available waste heat source temperature (T_H) with an incremental step of 1ºC. Assuming the isentropic expansion in turbine, the simulation was carried out for maximum system efficiency (η_I). As per the optimization conditions, the model was simulated and results of simulation were tabulated for each selected ORC working fluid R152a, R134a and R245fa [Table 3].

Table 3: Simulated Parameters-TIT Optimization (Optimum-TIP) [R134a, R152a, R245fa]

Selected Working Fluid ➔	**R134a**	**R152a**	**R245fa**
Properties (Optimization Condition) ➔	η_{I_Max}	η_{I_Max}	η_{I_Max}
Condenser Pressure P_c (MPa)	0.7062	0.6328	0.1609
Condenser Temperature T_c (ºC)	27.00	27.00	27.00
Turbine Inlet Pressure [TIP] P_e (MPa)	2.95	2.66	2.50
Turbine Inlet Temp. [TIT] T_1 (ºC)	138.00	178.00	151.00
Turbine Exit Temp. [TET] T_2 (ºC)	79.85	110.17	65.51
Turbine Exit Quality x_2	1.2783	1.3317	1.1826
Turbine Work Output W_T (kJ/kg)	38.66	68.66	57.08
System Efficiency η_I (%)	14.45	15.27	20.19
Second Law Efficiency η_{II} (%)	41.55	43.93	58.20
Mass Flow Rate $\dot{m}_{wf}$ (kg/s)	1.00	1.00	1.00

4.4 Overall ORC Model-Performance Evaluation (Optimized Parameters)

In the final stage, in order to estimate the overall ORC system performance using optimized parameters (Turbine Inlet Pressure -TIP, Superheat Temperature -TIT) for selected working fluids, all the individual ORC system models of each working fluid were combined along with boiler system model. The Fig. 11 shows the overall ORC model.

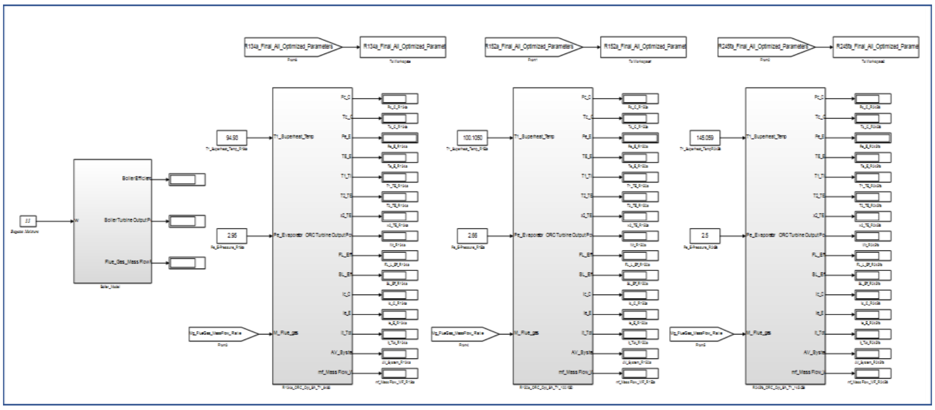

Fig. 11: ORC System Model – Performance Evaluation (Optimized Parameters)

Table 4: Performance Evaluation of ORC using available Waste Heat (Flue Gas)

Boiler System						
Moisture Content in Bagasse ➔	**50%**			**35%**		
Flue Gas Mass Flow Rate M_{fg} (kg/s)	11.89			14.78		
Boiler-Turbine Power (kW)	1238			1785		
Boiler Efficiency (%)	61.31			67.81		
ORC System						
Selected Working Fluid ➔	**R134a**	**R152a**	**R245fa**	**R134a**	**R152a**	**R245fa**
Condenser Pressure P_c (MPa)	0.7062	0.6328	0.1609	0.7062	0.6328	0.1609
Condenser Temperature T_c (ºC)	27.00	27.00	27.00	27.00	27.00	27.00
Turbine Inlet Pressure P_e (MPa)	2.95	2.66	2.50	2.95	2.66	2.50
Evaporator Temperature T_e (ºC)	85.38	85.93	133.55	85.38	85.93	133.55
Turbine Inlet Temp. T_1 (ºC)	94.93	100.11	145.06	94.93	100.11	145.06
Turbine Exit Temp. T_2 (ºC)	29.87	28.46	58.03	29.87	28.46	58.03
Turbine Exit Quality x_2	1.0177	1.0006	1.1483	1.0177	1.0006	1.1483
Turbine Power Output W_T (kW)	191.70	200.28	273.54	238.40	249.07	340.18
System Efficiency η_I (%)	14.13	14.77	20.17	14.13	14.77	20.17
Second Law Efficiency η_{II} (%)	40.63	42.46	58.15	40.63	42.46	58.15
Condenser Irreversibility Rate $\dot{I}_{condenser}$ (kW)	18.78	17.46	21.90	23.35	21.72	27.24
Evaporator Irreversibility Rate $\dot{I}_{evaporator}$ (kW)	261.35	253.94	175.00	325.02	315.80	217.63
System Total Irreversibility Rate $\dot{I}_{Total}$ (kW)	280.13	271.40	196.90	348.37	337.52	244.87
Availability Ratio (Φ)	79.35	79.99	85.48	79.35	79.99	85.48
Mass Flow Rate of working Fluid $\dot{m}_{wf}$ (kg/s)	6.51	4.18	4.96	8.10	5.20	6.16
Pinch Point Temperature T_{PP} (ºC)	127.24	119.00	139.97	127.24	119.00	139.97
Pinch Point Temp. Difference (ºC)	41.86	33.07	6.42	41.86	33.07	6.42

After simulating Stage-III TIT optimization, the optimum Superheat temperature for maximum system efficiency was compared with the results of Stage-II Optimization superheat temperature and it was observed that the optimum Superheat temperature (T_1) and turbine exit temperature (T_2) of Stage-III Optimization was high which leads to higher heat rejection rates in condenser. Finally, the simulated parameters obtained in Stage-II were taken as optimum parameters for better performance of ORC. The optimized parameters were used in Simulink Model to evaluate the ORC performance taking into account the waste heat of flue gases (M_{fg}) from Boiler Plant. The Boiler Model was simulated for two different conditions of moisture content in bagasse viz; 50% and 35% and simulated parameters were used for performance evaluation of ORC System. The evaluated results of ORC with optimized parameters for selected working fluids R152a, R134a and R245fa are tabulated in Table 4.

5. Result and Discussion

The simulation data recorded from the workspace of MATLAB Simulink using developed models of boiler system and different stages of ORC optimization was used to generate the graphs in order to discuss and investigate the effects of operating conditions on performance parameters. The optimization data obtained during stage-I turbine inlet pressure (TIP) optimization considering saturated vapours at inlet of turbine for selected working fluids R134a, R152a and R245fa of ORC system are presented in Table 1. The data obtained during Stage-II TIP optimization considering superheating of turbine inlet up to TIT of Stage-I optimization for maximum work are presented in Table 2 for ORCs with selected working fluids. Table 3 presents the stage-III TIT optimization data for maximum efficiency at inlet turbine pressure of stage-II optimization with superheating up to available waste heat flue gas temperature for ORCs with selected working fluids.

5.1 Temperature-Entropy (T-s) Plots

Taking the optimized parameter values from Table 1, the T-s diagrams of ORC for working fluid R134a, R152a and R245fa are plotted on corresponding T-s plots and shown in Fig. 12, Fig. 13 and Fig. 14 respectively. All the ORCs on T-s diagram for respective working fluids were plotted for three different conditions of optimization:

- Optimum Work (W_T) by varying turbine inlet pressure (TIP) up to critical pressure of corresponding working fluid.
- Optimum Efficiency (η_I) by varying turbine inlet pressure (TIP) up to critical pressure of corresponding working fluid.
- Superheating at constant turbine inlet pressure (TIP) [2.95 MPa for R134a, 2.66 MPa for R152a and 2.5 MPa for R245fa] till inlet turbine temperature (TIT) of stage-I optimization condition [94.93°C for R134a, 100.11°C for R152a and 145.06°C for R245fa].

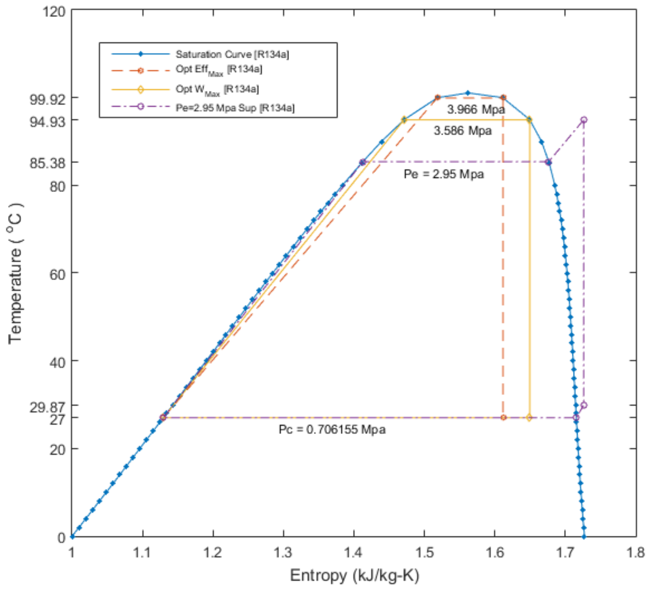

Fig. 12: T-s Plot-ORC R134a at Opt. Work, Opt. Efficiency and Superheating at P_e = 2.95MPa

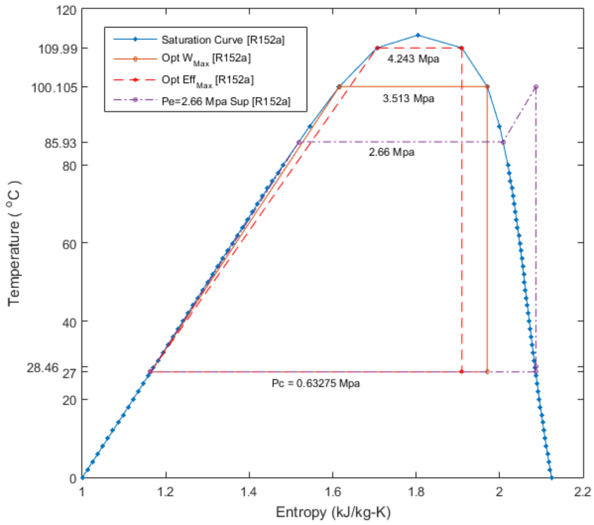

Fig. 13: T-s Plot-ORC R152a at Opt. Work, Opt. Efficiency and Superheating at P_e = 2.66 MPa

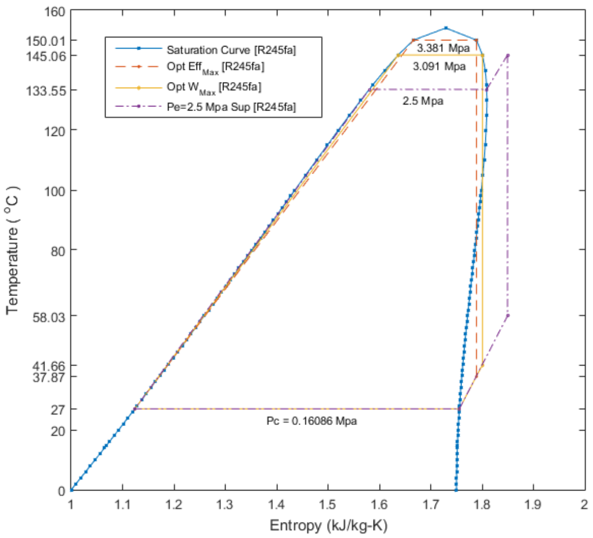

Fig. 14: T-s Plot-ORC R245fa at Opt. Work, Opt. Efficiency and Superheating at P_e = 2.5 MPa

5.2 Work (W_T) and Efficiency (η_I) Effects for variation of Inlet Turbine Pressure (TIP) with saturated vapours at Inlet of Turbine.

Based on the optimization data recorded during stage-I TIP optimization, the output turbine work (W_T) and system efficiency (η_I) effects for variation of inlet turbine pressure (TIP) with saturated vapours at inlet of turbine for selected fluids R134a, R152a and R245fa are shown in the Fig. 15, Fig. 16 and Fig. 17 respectively.

It is observed that the turbine work (W_T) and efficiency (η_I) both increases nonlinearly with inlet turbine pressure for all selected working fluids R134a, R152a and R245fa. For optimization condition of maximum work (W_{T_Max}), the optimal parameters; maximum turbine work (W_{T_Max}), inlet turbine pressure (P_e), inlet turbine temperature (T_1) and turbine exit quality (x_2) respectively for selected working fluids were found to be; for R134a: 27.24 kJ/kg, 3.586 MPa, 94.93ºC and 0.8872, for R152a: 46.75 kJ/kg, 3.513 MPa, 100.11ºC and 0.8739, for R245fa: 53.32 kJ/kg, 3.091 MPa, 145.06ºC and 1.0710. Whereas for optimization condition of maximum efficiency (η_I) the increase in efficiency was observed almost till critical pressure and the optimal parameters; maximum efficiency (η_{I_Max}), inlet turbine pressure (P_e), inlet turbine temperature (T_1) and turbine exit quality (x_2)

Tuijin Jishu/Journal of Propulsion Technology
ISSN: 1001-4055
Vol. 45 No. 3 (2024)

respectively for selected working fluids were found to be; for R134a; 15.12%, 3.966 MPa, 99.92ºC and 0.8225, for R152a: 17.08%, 4.243 MPa, 109.99ºC and 0.8084, for R245fa: 21.08%, 3.381 MPa, 150.01ºC and 1.0526.

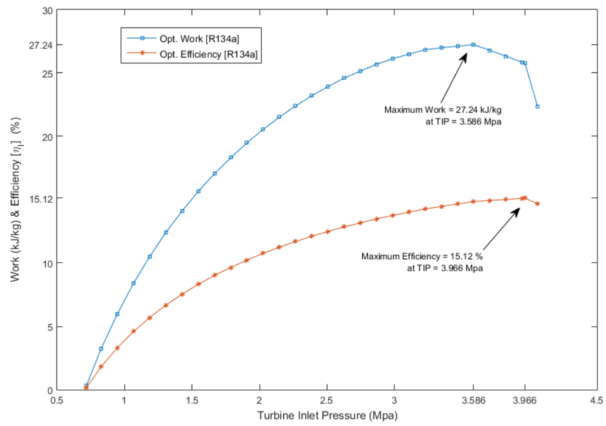

Fig. 15: Work (W_T) and Efficiency (η_I) vs Turbine Inlet Pressure [TIT-Saturated]-R134a

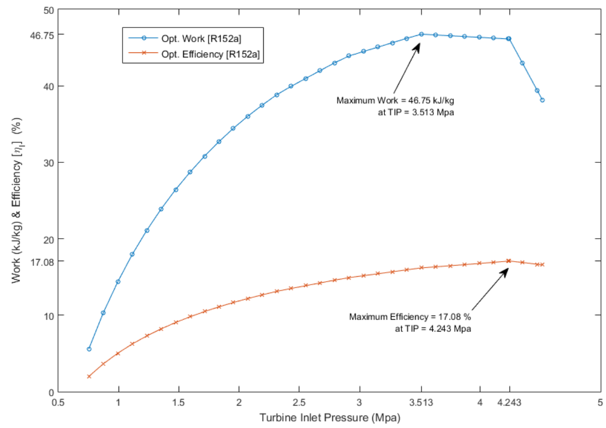

Fig. 16: Work (W_T) and Efficiency (η_I) vs Turbine Inlet Pressure [TIT-Saturated]-R152a

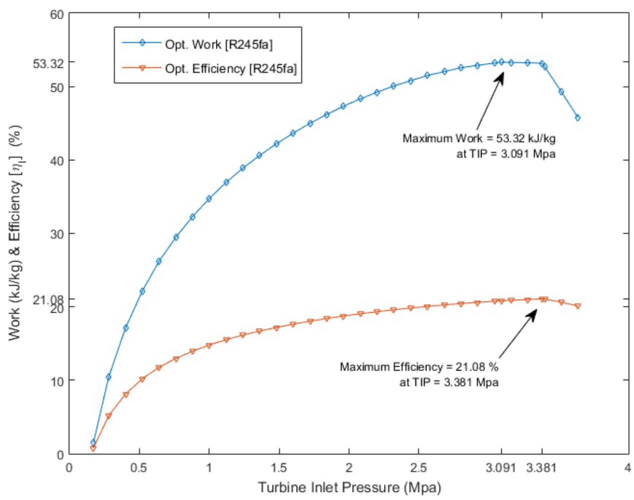

Fig. 17: Work (W_T) and Efficiency (η_I) vs Turbine Inlet Pressure [TIT-Saturated]-R245fa

In overall comparison of the results of stage-I TIP Optimization depicted from Fig. 15, Fig. 16 and Fig. 17 for the effects on turbine output work and system efficiency with variation in inlet turbine pressure, it is observed that the ORC with working fluid R245fa showing better performance with maximum work of 53.32 kJ/kg and maximum efficiency of 21.08% as compared to the ORCs with working fluids R134a and R152a. Also, the turbine exit quality for R134a and R152a are found to be wet vapours whereas dry vapours for R245fa, which is important deciding parameter in accordance with corrosion effects of turbine blades during expansion process.

5.3 Effects on Turbine Exit Quality (x_2) for variation of inlet Turbine Pressure (TIP) with saturated vapours at Inlet of Turbine.

From the results of stage-I TIP optimization for optimum work and efficiency discussed in previous section 5.2 in reference with Fig. 15, Fig. 16 and Fig. 17, it is of utmost importance to focus on turbine exit quality at the inlet turbine pressure corresponding to the optimum values of Work and efficiency.

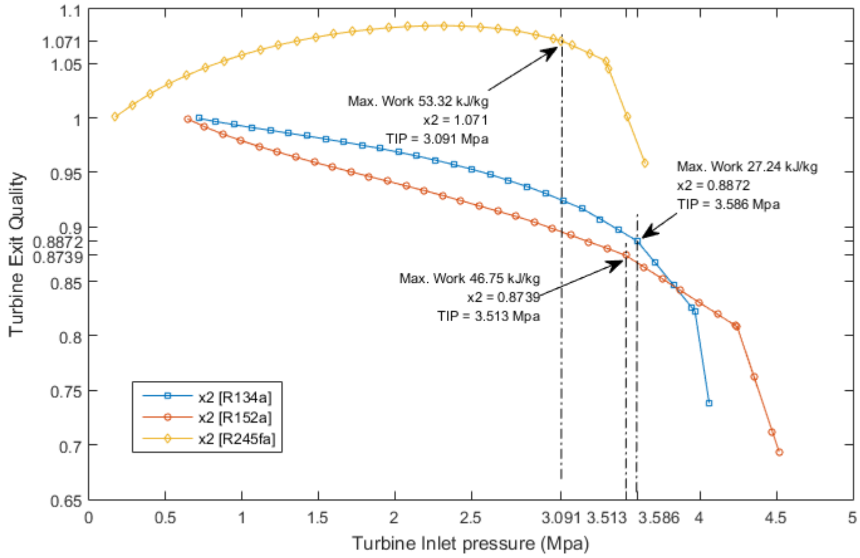

Fig. 18: Turbine Exit Quality (x_2) vs Turbine Inlet Pressure (TIP) for R134a, R152a and R245fa [TIT-Saturated]

Fig. 18 shows the effects on turbine exit quality (x_2) with variation of inlet turbine pressure for all selected working fluids. It is observed that for R134a and R152a the turbine exit quality is wet vapours which is responsible for formation of dewdrops at the latter stage of expansion in turbine. Presence of wet vapours in working segment of turbine damages the turbine blades. Whereas for R245fa though the quality at turbine exit is dry, it is observed from T-s plot of R245fa shown in Fig. 14 of section 5.1 that during expansion the turbine working segment goes in to multi-phase region. The reason behind this is due to change in the slope of saturation vapour curve. As per the T-s plot, positive slope of saturation curve changes to negative. So, in order to save turbine blades from corrosion effects it is mandatory to ensure always dry vapours throughout the turbine working segment during expansion. This can be achieved by superheating of vapours before turbine inlet. To investigate the optimum inlet turbine pressure and superheat temperature, TIP and TIT optimization were conducted and results tabulated in Tables 2 and 3 are explained in subsequent sections 5.4, 5.5 and 5.6.

5.4 Effects on Turbine Exit Quality (x_2) and Work (W_T) for variation of inlet Turbine Pressure (TIP) with superheated vapours at Inlet of Turbine.

As discussed in previous section 5.3, stage-II TIP optimization was conducted with superheating of turbine inlet vapours up to the inlet turbine temperature (TIT) corresponding to optimum inlet turbine pressure of stage-I TIP optimization. The data is plotted to analyze the effects on work (W_T) and turbine exit quality (x_2) for variation of inlet turbine pressure for selected fluids R134a, R152a and R245fa as shown in Fig. 19, Fig. 20 and Fig. 21 respectively.

It is observed that the turbine work (W_T) increases to maximum and starts decreasing with increase of inlet turbine pressure TIP for all selected working fluids. For optimization condition of maximum work (W_{T_Max}) using superheat temperature (T_1) of 94.93 ºC, 100.11 ºC and 145.06 ºC for selected working fluids R134a, R152a and R245fa respectively, the optimal parameters; maximum work (W_{T_Max}), inlet turbine pressure (P_e), and turbine exit quality (x_2) were found to be; For R134a: 29.44 kJ/kg, 2.95 MPa, and 1.018. For R152a: 48.17 kJ/kg, 3.1 MPa, and 0.9439. For R245fa: 55.18 kJ/kg, 2.5 MPa, and 1.1483.

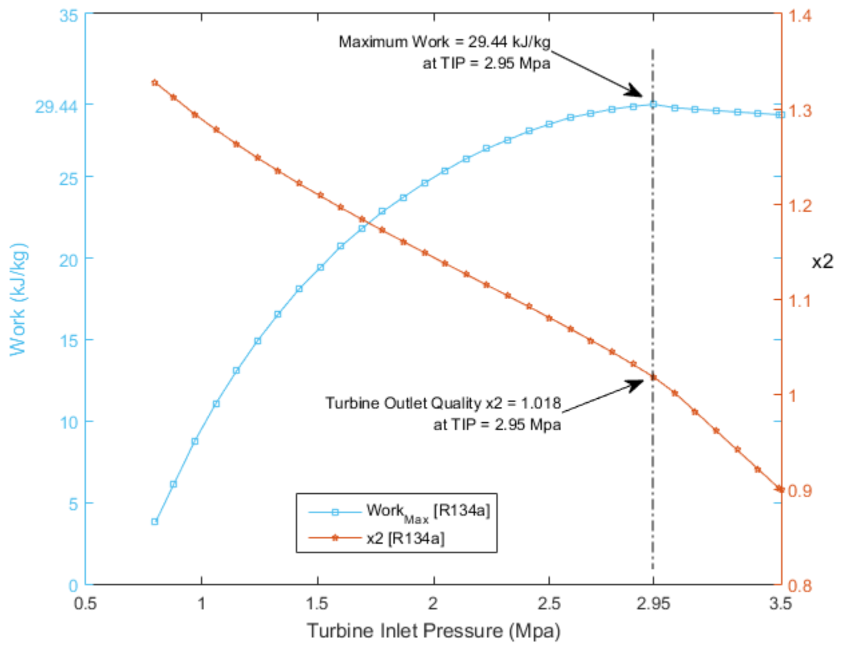

Fig. 19: Work (W_T) and Turbine Exit Quality (x_2) vs Turbine Inlet Pressure (TIP) for R134a [Superheat Temperature $T_1 = 94.93^oC$]

Considering the results for working fluid R152a, the turbine exit quality (x_2) of 0.9439 obtained was still wet which may lead to formation of dewdrops at latter stage of expansion and may damage the turbine blades. So, on investigation from Fig. 20 the feasible data was selected for corrected turbine inlet pressure of 2.66 MPa. The optimum work (W_T) and turbine exit quality (x_2) corresponding to corrected inlet turbine pressure was found to be 47.89 kJ/kg and 1.0006 respectively with turbine exit temperature (T_2) of 28.46 ºC.

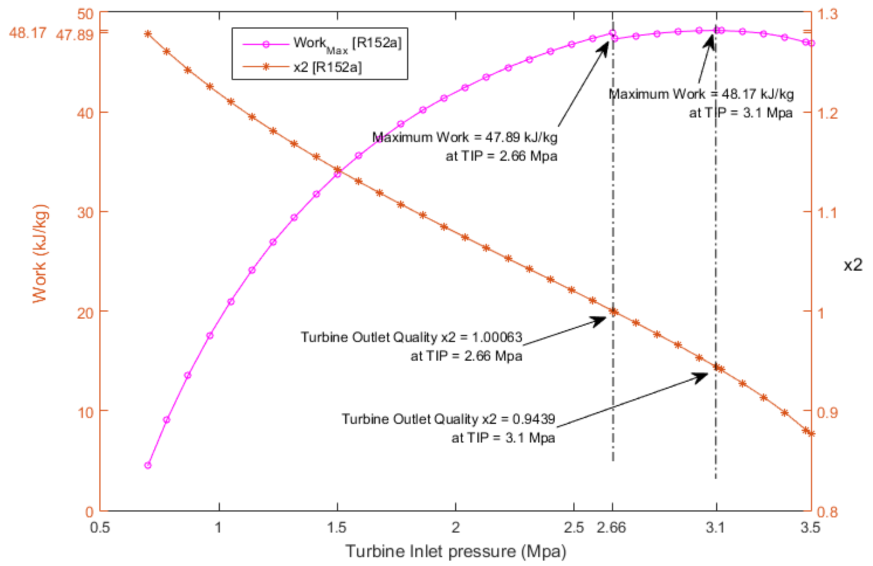

Fig. 20: Work (W_T) and Turbine Exit Quality (x_2) vs Turbine Inlet Pressure (TIP) for R152a [Superheat Temperature $T_1 = 100.11^oC$]

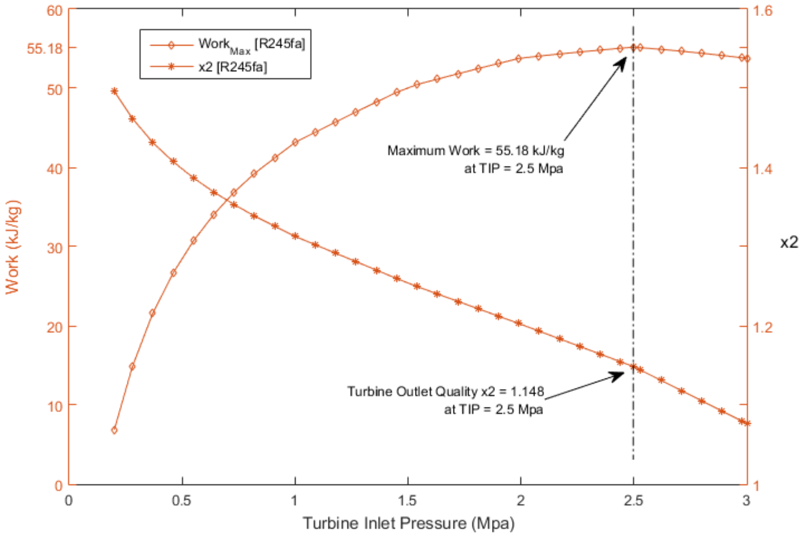

Fig. 21: Work (W_T) and Turbine Exit Quality (x_2) vs Turbine Inlet Pressure (TIP) for R245fa [Superheat Temperature $T_1 = 145.06^oC$]

In overall comparison of the results of stage-I TIP Optimization with saturated vapours at turbine inlet depicted from Fig. 15, Fig. 16 and Fig. 17 and stage-II TIP Optimization with superheated vapours at turbine inlet depicted from Fig. 19, Fig. 20 and Fig. 21 for the effects on turbine output work with variation in inlet turbine pressure, it is observed that all selected working fluids showing increase in the turbine work output when operated with superheating of turbine inlet vapours. The ORC with working fluid R245fa showing better performance compared to the ORCs with working fluids R134a and R152a.

Comparing Stage-I and Stage-II optimization results for R245fa, it is observed that the turbine output work has been increased by around 3.49% with dry vapours throughout the turbine working segment avoiding multi-phase problem and restricting corrosion effects on turbine blades during expansion process.

5.5 Effect on Turbine Exit Quality (x_2) for Variation of Inlet Turbine Temperature (TIT) with Superheated Vapours at Inlet of Turbine.

The Fig. 22 shows the effect of variation of inlet turbine temperature on turbine exit quality mentioning the details of stage-II optimization and stage-III optimization results as per Table 2 and Table 3.

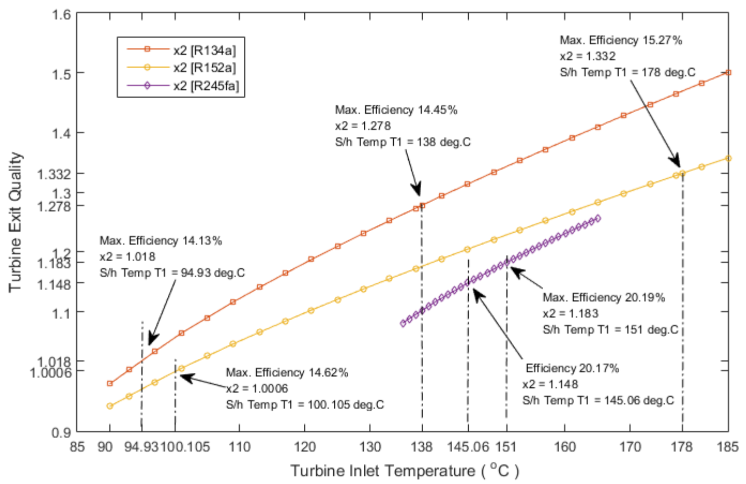

Fig. 22: Turbine Exit Quality (x_2) vs Turbine Inlet Temperature (TIT) for R134a, R152a and R245fa [TIT-Superheated]

The comparative analysis has been carried out for turbine inlet temperature of superheating and turbine exit temperature for effective heat rejection in condenser. On investigation, it was observed that the results obtained during stage-III TIT optimization [Table 3] giving high values of inlet turbine temperature (T_1) and turbine exit temperature (T_2) with very small variation in system efficiency for all selected fluids as compared to the stage-II optimization results [Table 2]. The higher values of inlet turbine temperature (T_1) lead to increase in load of evaporator for available waste heat source and higher turbine exit temperature (T_2) leads to high heat rejection rates in condenser increasing the quantity of circulating water as cooling medium. So, finally stage-II optimization data from Table 2 was selected and applied for ORC to generate power with available waste heat of flue gases from selected process industry. The final outcomes of the study are tabulated in Table 4 for comparison and recommendation of better working fluid among all selected to generate maximum power from ORC System.

5.6 Overall Performance Evaluation of ORC System using Optimized Parameters.

Final optimized parameters were used to estimate the performance of ORC system for all selected fluids and the results are tabulated in Table 4.

Pinch Point Analysis: Fig. 23 shows pinch point analysis for R134a, R152a and R245fa respectively.

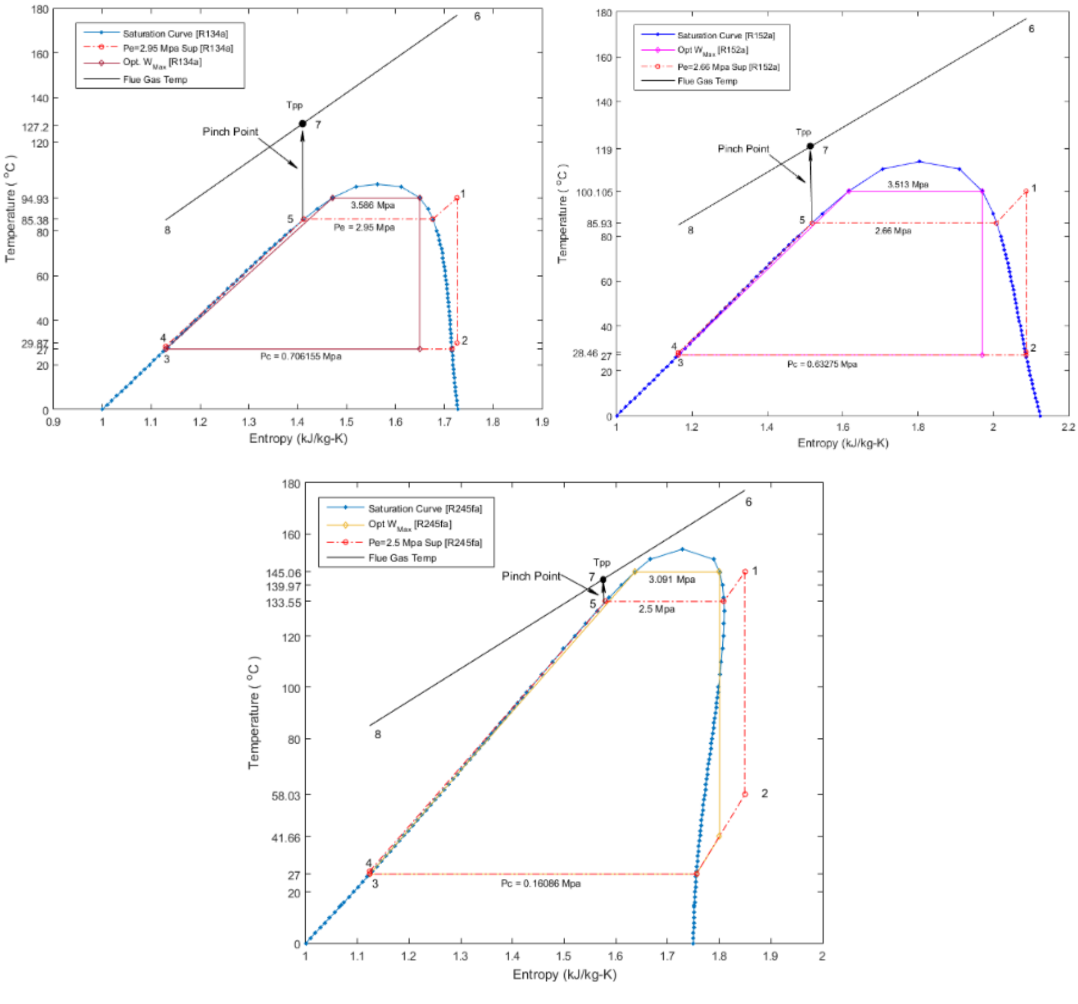

Fig. 23: Pinch Point Temperature Diagrams of ORC for R134a, R152a and R245fa

It is observed that the temperature difference at pinch point between working fluid and flue gas are higher for R134a and R152a and optimum for R245fa. For R134a temperature at pinch point (T_{PP}) was found to be 127.24ºC with PPTD of 41.86ºC. For R152a temperature at pinch point (T_{PP}) was found to be 119.00ºC with PPTD of

33.07ºC. For R245fa temperature at pinch point (T_{PP}) was found to be 139.97ºC with PPTD of 6.42ºC. R245fa showing better performance as compare to R134a and R152a.

ORC System Overall Performance: For ORC to utilize the available waste heat in flue gases from selected sugar factory as process industry, the mass flow rate of flue gases (M_{fg}), Boiler Efficiency and Boiler Turbine Power output were estimated by running developed Boiler system model. The model was run at two different conditions of moisture content (50% and 35%) in the fuel (Bagasse) used in Boiler. Utilizing the available waste heat in flue gases at 177ºC and estimated mass flow rates of flue gases at 50% & 35% moisture content in boiler fuel (Bagasse) [Table 4], ORC system performance is evaluated for optimized parameters and tabulated in Table 4.

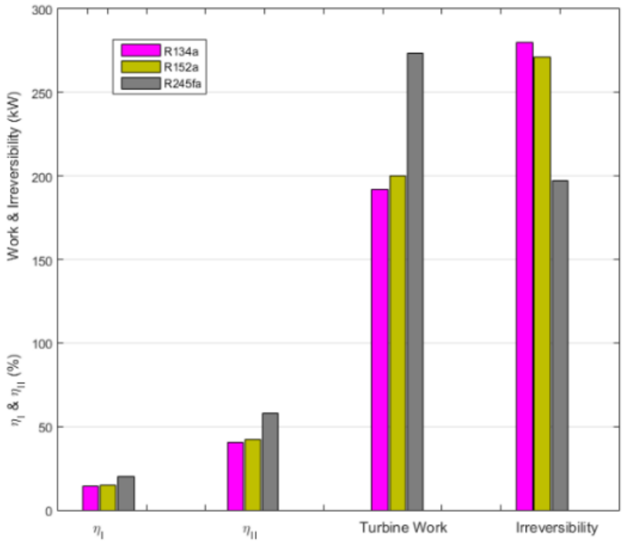

Fig. 24: Overall Performance of ORC System for R134a, R152a and R245fa [Bagasse Used: 50% Moisture]

Fig. 24 and Fig. 25 shows the overall performance for all working fluids of ORC system. The outputs/parameters plotted for comparison are system efficiency (η_I), second law efficiency (η_{II}), turbine power output (W_T) and system total irreversibility rates ($\dot{I}_{Total}$).

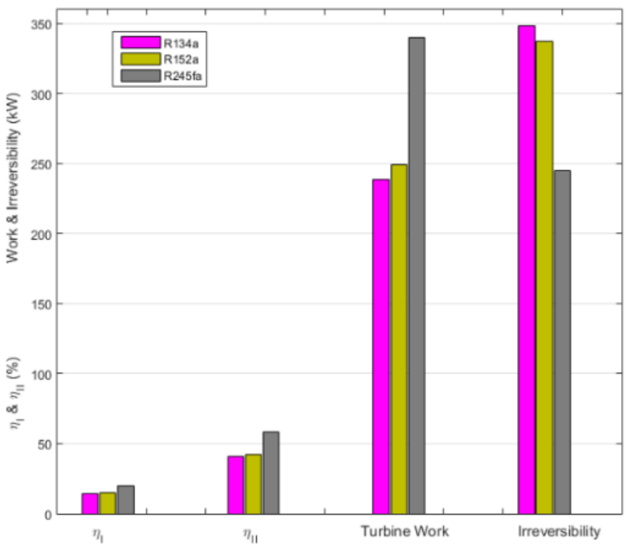

Fig. 25: Overall Performance of ORC System for R134a, R152a and R245fa [Bagasse Used: 35% Moisture]

Fig. 26 shows the irreversibility rates associated with components condenser and evaporator for R134a, R152a and R245fa. It is observed that the high irreversibility rates are associated with evaporator of ORC system. Also, R245fa shows lowest irreversibility rates as compared to R134a and R152a.

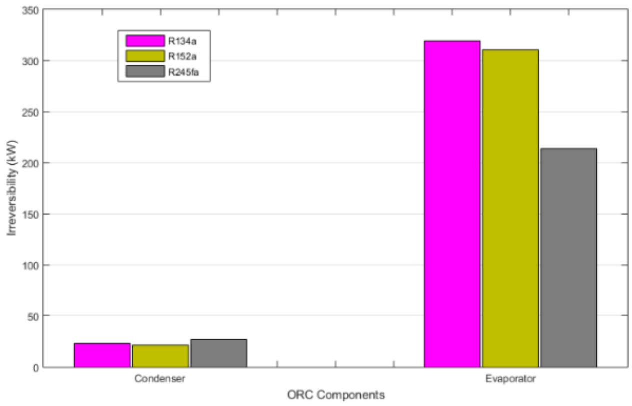

Fig. 26: Component-wise Irreversibility Rates for R134a, R152a and R245fa

From the Fig. 24, Fig. 25 and Fig. 26, it is observed that ORC with working fluid R245fa shows better performance among all selected working fluids. The ORC with R245fa will generate more power (W_T) with higher efficiencies (η_I) and (η_{II}) and lower irreversibility rates ($\dot{I}_{Total}$) compared to R134a and R152a. The ORC with R245fa will generate 340.18 kW power with efficiency (η_I) of 20.17% at working fluid mass flow rate of ($\dot{m}_{wf}$) as 6.16 kg/s. Also, the second law efficiency (η_{II}) of 58.15% with Pinch Point of 6.42^{o}C is obtained.

6. Conclusion

Organic Rankine Cycle (ORC) using R134a, R152a and R245fa as working fluids has been analyzed in the present research work. With respect to the abilities of selected working fluids to convert low-grade waste heat energy, the results are compared for final recommendation of better working fluid. The flue gases available at the exhaust of selected process industry with 177^{o}C temperature and 14.78 kg/s mass flow rate is utilized for power generation from ORC system. All selected working fluids are investigated using parametric optimization of turbine work output, system efficiencies with respect to TIP along saturated vapour curve and TIT with constant pressure superheating through simulation using developed MATLAB Simulink Models of ORC System. The following results are concluded based on the present analysis.

- Instead of using higher evaporating temperatures near the temperature of waste heat source, the working fluids with moderate evaporating temperature can be used for optimal performance of ORC system.
- Based on the optimization results, the T-s plots shows that the system efficiency is affected by the slope of saturated vapour curve.
- For the working fluid R134a, the TIP-Optimization gives optimum work of 27.24 kJ/kg with efficiency of 14.86% and turbine exit vapour quality of 0.8872 at TIP of 3.586 MPa. The turbine exit vapour quality is reduced to 0.8225 for an optimum efficiency of 15.12% at TIP of 3.966 MPa. The TIT optimization for constant pressure superheating at 2.95 MPa shows 138oC as optimum value of TIT for superheating and higher turbine exit temperature of 79.85^{o}C which leads to higher heat rejection rates in condenser. To avoid this, superheating temperature corresponding to temperature of TIP optimization is fixed to 94.33^{o}C. The results show work output of 29.44 kJ/kg with efficiency of 14.13% and improved turbine exit vapour quality of 1.0177. For TIT- optimization, the efficiency is slightly reduced as compared to TIP-Optimization but the work output is increased with dry vapours at turbine exit which avoid droplet formation in turbine working section and saves turbine blades from damage.
- For the working fluid R152a, the TIP-Optimization gives optimum work of 46.75 kJ/kg with efficiency of 16.19% and turbine exit vapour quality of 0.8739 at TIP of 3.513 MPa. The turbine exit vapour quality is reduced to 0.8084 for an optimum efficiency of 17.08% at TIP of 4.243 MPa. The TIT optimization

for constant pressure superheating at 2.66 MPa shows 178ºC as optimum value of TIT for superheating and higher turbine exit temperature of 110.17ºC which leads to higher heat rejection rates in condenser. To avoid this, superheating temperature corresponding to temperature of TIP optimization is fixed to 100.11ºC. The results show work output of 47.89 kJ/kg with efficiency of 14.77% and improved turbine exit vapour quality of 1.0006. For TIT- optimization, the efficiency is slightly reduced as compared to TIP-Optimization but the work output is increased with dry vapours at turbine exit which avoid droplet formation in turbine working section and saves turbine blades from damage.

- For the working fluid R245fa, the TIP-Optimization gives optimum work of 53.32 kJ/kg with efficiency of 20.83% and turbine exit vapour quality of 1.0710 at TIP of 3.091 MPa. The turbine exit vapour quality is reduced to 1.0526 for an optimum efficiency of 21.08% at TIP of 3.381 MPa. Though the turbine exit quality is dry but the working fluid goes in two-phase region during expansion in turbine working section. The TIT optimization for constant pressure superheating at 2.5 MPa shows 151oC as optimum value of TIT for superheating and higher turbine exit temperature of 65.51ºC which leads to higher heat rejection rates in condenser. To avoid this, superheating temperature corresponding to temperature of TIP optimization is fixed to 145.06ºC. The results show work output of 55.18 kJ/kg with efficiency of 20.17% and turbine exit vapour quality of 1.1483. For TIT- optimization, the efficiency is slightly reduced as compared to TIP-Optimization but the work output is increased with dry vapours always in turbine working section which avoid droplet formation in turbine working section and saves turbine blades from damage.
- The total system exergy destruction or irreversibility rate increases and second law efficiency decreases as the Pinch Point temperature difference increases.
- From among all selected working fluids, R245fa gives the best results as compared to R134a and R152a. So, R245fa is recommended for the Organic Rankine Cycle under the considered utilization of waste heat from selected process industry for generation of power. It can generate 340.18 kW power with total mass flow rate of working fluid ($\dot{m}_{wf}$) as 6.16 kg/s and efficiency (η_I) of 20.17% having Pinch Point of 6.42ºC with heat source flue gas and second law efficiency (η_{II}) of 58.15%.

Refrences

[1] Minea V. (2014). Power generation with ORC machines using low grade waste heat or renewable energy. Applied Thermal Engineering, 69(1-2), 143-154.

[2] Desai NB, Bandyopadhyay S. (2015). Integration of parabolic trough and linear Fresnel collectors for optimum design of concentrating solar thermal power plant. Clean Technologies and Environmental Policy, 17(7), 1945-1961.

[3] Desai NB, Bandyopadhyay S. (2016). Thermo-economic analysis and selection of working fluid for solar organic Rankine cycle. Applied Thermal Engineering, 95, 471-481.

[4] Quoilin S, Van Den Broek M, Declaye S, Dewallef P, Lemort V. (2013). Techno-economic survey of organic rankine cycle (ORC) systems. Renew Sustain Energy Rev, 22, 168-186.

[5] Padilla RV, Archibold AR, Demirkaya G, Besarati S, Goswami DY, Rahman MM, Stefanakos EL. (2012). Performance analysis of a rankine cycle integrated with the goswami combined power and cooling cycle. ASME J Energy Resour Technol, 134(3), 032001.

[6] Hung TC, Shai TY, Wang SK. (1997). A review of Organic Rankine Cycle (ORCs) for the recovery of low-grade waste heat. Energy, 22(7), 661-667.

[7] Liu BT, Chein KH, Wang CC. (2004). Effect of working fluids on Organic Rankine Cycle for waste heat recovery. Energy, 29(8), 1207-1217.

[8] Cong CE, Velautham S, Darus AN. (2005). Solar thermal Organic Rankine Cycle as a renewable energy option. Jurnal Mekanikal, 20, 68-77.

[9] Hung TC, Wang SK, Kuo CH, Pei BS, Tsai KF. (2010). A study of organic working fluids on system efficiency of an ORC using low grade energy sources. Energy, 35(3), 1403-1411.

[10] Maizza V, Maizza A. (2001). Unconventional working fluids in Organic Rankine Cycle for waste energy recovery systems. Applied Thermal Engineerin, 21 (3), 381-390.

[11] Wei D, Lu X, Lu Z, Ju Z. (2007). Performance analysis and optimization of Organic Rankine Cycle (ORC) for waste heat recovery. Energy Conversion and Management, 48(4), 1113-1119.
[12] Saleh B, Koglbauer G, Wendland M, Fischer J. (2007). Working fluids for low temperature Organic Rankine Cycles. Energy, 32(7), 1210-1221.
[13] Pedro JM, Chamra LM, Srinivasan K, Chandramohan S. (2008). An examination of regenerative Organic Rankine Cycle using dry fluids. Applied Thermal Engineering, 28 (8-9), 998-1007.
[14] Dai Y, Wang J, Gao L. (2009). Parametric optimization and comparative study of Organic Rankine Cycle (ORC) for low grade waste heat recovery. Energy Conversion and Management, 50(3), 576-582.
[15] Yamamoto T, Furuhata T, Arai N, Mori K. (2001). Design and testing of the ORC. Energy, 26, 239-251.
[16] Young JB, Minsung K, Chang KC, Sung JK. (2011). Power-based performance comparison between carbon dioxide and R125 transcritical cycles for a low-grade heat source. Applied Energy, 88, 892-898.
[17] Techanche BF, Papadakisa G, Lambrinosa G, Frangoudakisa A. (2009). Fluid selection for a low-temperature solar organic Rankine cycle. Applied Thermal Engineering, 29 (11-12), 2468-2476.
[18] Nguyen VM, Doherty PS, Riffat SB. (2001). Development of a prototype low temperature Rankine Cycle electricity generation system. Applied Thermal Engineering, 21, 169-218.
[19] Drischer U, Bruggemann D. (2007). Fluid selection for the Organic Rankine Cycle (ORC) in biomass power and heat plants. Applied Thermal Engineering, 27, 223-228.
[20] Schuster A, Karellas S, Aumann R. (2010). Efficiency optimization potential in supercritical organic Rankine cycles. Energy, 35, 1033-1039.
[21] Guo J, Xu M, Cheng L. (2010). Thermodynamic analysis of waste heat power gener ation system. Energy, 35(7), 2824-2835.
[22] Invernizzi C, Iora P, Silva P. (2007). Bottoming micro-Rankine cycles for micro-gas turbines. Applied Thermal Engineering, 27, 100-110.