Design of a Solar Organic Rankine Cycle Prototype for 1 kW Power Output

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Abstract

Rising energy demand due to industrial development, population growth, is pushing the mankind for utilizing more and more conventional energy sources such as coal, oil and gas. There is a need to minimize the use of such types of resources because, it contributes to the global warming, pollution and climate change. Use of alternative sources of energy such as solar, hydro, wind, tidal, geothermal, biofuel, and nuclear are preferable and are promising for the modern world. Solar energy, which is abundantly available in Jorhat area, can be used for power generation using Organic Rankine Cycle (ORC) Technology, is the source of energy selected for this work. Use of solar energy can reduce the load on the conventional energy sources. Solar parabolic trough collector (PTC) system is employed as the evaporator of the solar organic Rankine cycle (SORC) system. Working fluid for the subcritical ORC is R245fa. Reciprocating piston type expander is used for the expansion of the working fluid. The 1 kW capacity alternator coupled to the expander shaft can convert the mechanical power into electricity. Two heat exchangers have been designed for the ORC prototype, one is an air cooled cross-flow heat exchanger for cooling the hot organic vapours and one shell and tube condenser (water cooled) for condensing the vapour into liquid state. Theoretical modelling of the prototype assembly is done using DWSIM and thermo-economic analysis has been carried out. Results indicate that the system can generate electricity in the range 439-763 W. The 1st law and 2nd law efficiencies of the cycle varies from 25.13 to 37.07% and 29.69 to 43.57% respectively. The payback period for the system is estimated to be around 17.27 years.

Keywords — *Organic Rankine Cycle (ORC), R245fa, solar, thermo-economic analysis.*

I. INTRODUCTION

Population growth, leading to the rise in energy demand, resisting the economic growth and access to modern technological advancements as well as uneven distribution of resources is a serious concern to the present and future of this planet. Due to uncontrolled population growth in the past, there are a tremendous number of industries worldwide to meet several demands and huge number of transportation vehicles on the roadways, airways and in the water. The heat sources and engines operating these systems are releasing wasted heat and toxic pollutants to the environment which is causing global warming and climatic change and many other problems. These issues can be addressed and the pollution can be reduced to a certain limit by switching to renewables and implementing organic Rankine cycle (ORC) technology.

During the rainy seasons, there are frequent power cuts in Assam. Which causes interruption in several works. Also, there is a lack of sufficient power plants in Assam to meet the electrical power demand in the state. There are lots of remote places where grid electricity does not covers its range. These problems can be tackled by the development of small-scale off-grid power plants which can provide electricity to the household and other needs as backup to the main grid electricity and provide full time electricity to the areas where the main grid electricity has not reached yet. However, micro-scale biomass-fired/solar powered ORC-based CHP (Combined Heat Power) units (<10kWe), having a great potential to meet the energy needs of buildings, have yet to be demonstrated or commercialized [1]. In a work done by Emily Spayde et al. [2], they have been able to determine the economic, energetic and environmental benefits that could be obtained from the implementation of a combined solar-powered ORC with electric energy storage (EES) to supply electricity to several commercial buildings including a large office, a small office, and a full service restaurant. The operational strategy for the ORC-EES system consists in the ORC charging the EES when the irradiation level is sufficient to generate power, and the EES providing electricity to the building when there is not irradiation (i.e., during night time). Electricity is purchased from the utility grid unless it is provided by the EES. The potential of the proposed system to reduce primary energy consumption (PEC), carbon dioxide emission (CDE), and cost was evaluated. Furthermore, the available capital cost for a variable payback period for the ORC-EES system was determined for each of the evaluated buildings. The effect of the number of solar collectors on the performance of the ORC-EES was also studied. Results indicate that the proposed ORC-EES system is able to satisfy 11%, 13%, and 18% of the electrical demand for the large office, the small office and the restaurant, respectively.

Another work done by Bianchi et al. [3], reports an experimental activity carried out for performance characterization of a prototypical micro-ORC energy system. In particular, the paper presents the test bench developed in the laboratories of the University of Bologna and the first obtained results in terms of thermodynamic performance and main components characterization. The ORC system comprises a small reciprocating three-piston expander, run on R134a as operating fluid. Heat is provided to the ORC from an external source, via hot water at temperature below 100°C, in order to simulate a low-enthalpy heat recovery process. The system rejects unused heat via a water cooled condenser. Thus, the investigated ORC is a plug and play system, requiring only to be connected to the hot and cold heat sources. The ORC system has been tested for prolonged operation at various thermal input conditions. In particular, the behavior of the key cycle parameters and performance indexes (e.g. max. and min. pressures, superheating temperature, expander isentropic efficiency, electric power output, etc.) are investigated as function of pump rotational speed (i.e. organic fluid mass flow rate), for three different set point values of the hot source (65°C, 75°C, 85°C). The operating thermodynamic cycle has been completely characterized by means of a real time measurement and acquisition tool, developed in LabVIEW environment. Performance variations of the system have been monitored: the electric power output ranges between 0.30 to 1.2 kW, with gross efficiency in the range 2.9-4.4%, while the expander "electro-isentropic" efficiency results in the range of 35-42%.

The ORC process is similar to the Steam process, which uses water as working fluid [9].The main difference being the working fluid which is an organic fluid. The organic fluid having higher molecular weight as compared to water and having low evaporation temperature is suitable for low grade heat (<300°C heat source) recovery to generate power. The conversion efficiency of the ORC systems are very low due to various factors such as:

1. Heat addition is less (of low quality).
2. Expander efficiency is low.
3. Heat loss in the components and other losses such as mechanical friction, pressure drops, etc.
4. Exergy destruction in the components.

The main advantage of using ORC to generate power is that it can utilize the energy which would otherwise be wasted. Various low grade heat sources such as solar, biomass, engine waste heat, industrial waste heat, and geothermal energy can be harnessed by using ORC. This can significantly reduce the thermal pollution as well as the consumption of fossil fuels.

This paper presents the design, simulation and thermo-economic analysis of a small scale solar ORC prototype. The calculations were done on the basis of the monthly average data by using prediction equations from literature [4] for a period of nine months starting from March to November. Ten days from each month has been excluded taking into consideration the unfavorable weather conditions. From these data, it is possible to assume a rough figure of the attainable power in kWh for a year. Which in turn will give an idea about how much units of energy can be saved by using the prototype. The average lifetime for the prototype is estimated to be around 20 years. The payback period can be calculated and linked to the economic analysis of the model.

Fig. 1 shows the block diagram of the SORC (Solar Organic Rankine Cycle) prototype. The liquid refrigerant, R245fa is drawn by the pump from the reservoir and is passed through the evaporator. The evaporator for this system is the solar parabolic trough collector system which directly evaporates the working fluid from liquid to vapor state. The hot vapors are stored in the TES (Thermal Energy Storage). The high pressure and temperature vapors are then expanded in the expander to deliver work. The low pressure vapors at the exit of the expander are then cooled in the heat exchanger to 50°C before entering the condenser. The heat exchanger designed for this system is a cross-flow, air-cooled heat exchanger which uses forced draft fan for cooling the hot stream of vapors. The condenser condenses the vapors to liquid form. The condenser employed in the system is a shell and tube heat exchanger which uses water as cooling medium and provides sub-cooling after condensation to 45°C.

II. EXPANDER

Expanders, in general, can be categorized into two types: one is the velocity type, such as axial turbine expanders; the other is the volume type, such as screw expanders, scroll expanders, and reciprocal piston expanders [1]. The low capacity ORCs are mostly dominated by scroll machines, which are either modified from HVAC (Heating, Ventilation, and Air Conditioning) compressors, or manufactured by some companies. There are other types of volumetric expanders such as screw expander, rotary vane expander, rotary piston expander (Wankel type) and reciprocating piston expander, on which research and development works have been carried out and have been installed successfully in ORCs.

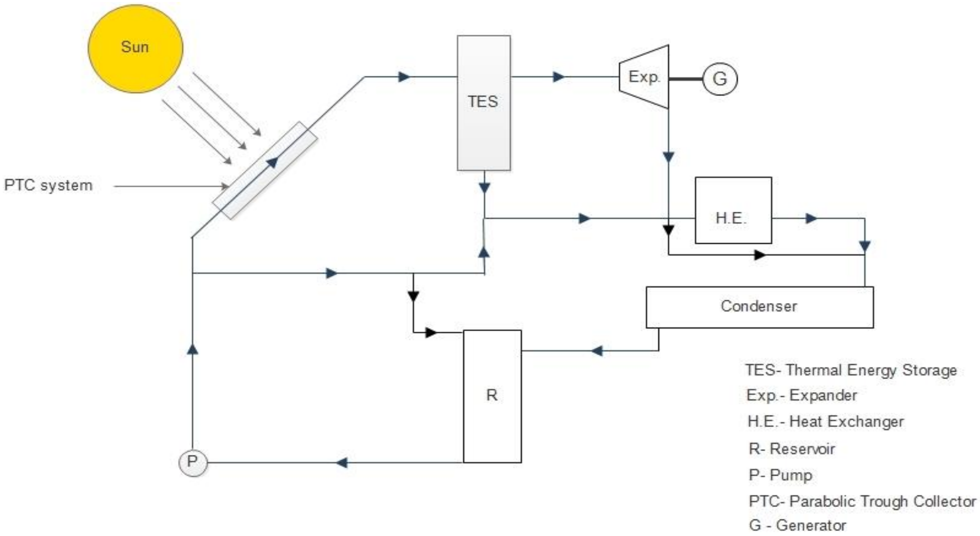

Fig 1: Block diagram of the ORC system

But, in India, we face a lack of maturity of the ORC market, and the expensive and complex expanders are not preferable for micro/small scale applications. Hence, the author has resorted to reciprocating piston type expander due to their wide availability and since it can be modified from waste ICEs (Internal Combustion Engine) which are either not in workable conditions or are banned by new pollution regulatory norms. Piston expanders actually show some advantages over other expansion machines, such as larger built-in volume ratio, high achievable operating pressures and temperatures, their ability to ingest liquid and low rotational speeds [17]. The maximum expander efficiency is expected to be 70%. Very few works have been done on piston expanders so far. To know more about advantages and disadvantages of reciprocating piston expanders, it is recommended to pay attention to below specified characteristics [13]:

1) This type is complex to design and manufacture.

2) It is expensive especially to manufacturing of various parts.

3) Reciprocating piston expanders have large friction losses because of the large number of interacting surfaces such as between piston rings, piston and cylinder wall.

4) In ORC system, by suing dissolving oil into working medium the impact of losses will be reduced.

5) Achievable efficiency is not as much as turbine expanders.

6) It is possible to act as an expander under two-phase medium condition.

7) Reciprocating expanders are not so sensitive to non-stable operating situation unlike turbine expanders.

8) Due to several static and dynamic parts not only the noise, vibration and durability of system should be modified but also the weight of expansion machine is heavy.

9) There are valve and torque impulses.

10) Because of a lot of movable parts, the reliability and balancing of this type are problems.

11) Reciprocating expander can be coupled to crank shaft directly.

In general, the turbo-machines are used in the ORC systems that has power output greater than 50 kW. In the case of 20-50 kW output power range, screw expanders are suitable. For power output range of 1-10 kWs, scroll expanders followed by piston and vane expanders are preferable [12]. A Landelle et al. [15], worked on a transcritical ORC with scroll expander and the maximum expander efficiency was found to be 66.5 % with 6 kW of gross production and supercritical entry conditions (T: 118.6 °C; P: 43.3 bar). Various types of expanders are shown in Fig. 2. Construction types of the four volumetric expanders, namely: piston, screw, scroll and vane expanders are explained below with brief description from literature [14].

Piston: The classical volume expander is the reciprocating piston expander. It can show high expansion efficiencies (e.g. 70% in Eilts et. al. (2012) [16]). The achievable volume ratios of volumetric expanders are in the range of 10 (Lemort et. al., 2013 [10]) or slightly higher. However, it needs a lot of bearings and in addition inlet and outlet valves which makes the design complex and costly. Liquid in the cylinder can cause damage. Thus, the piston expander should not be applied for wet expansion.

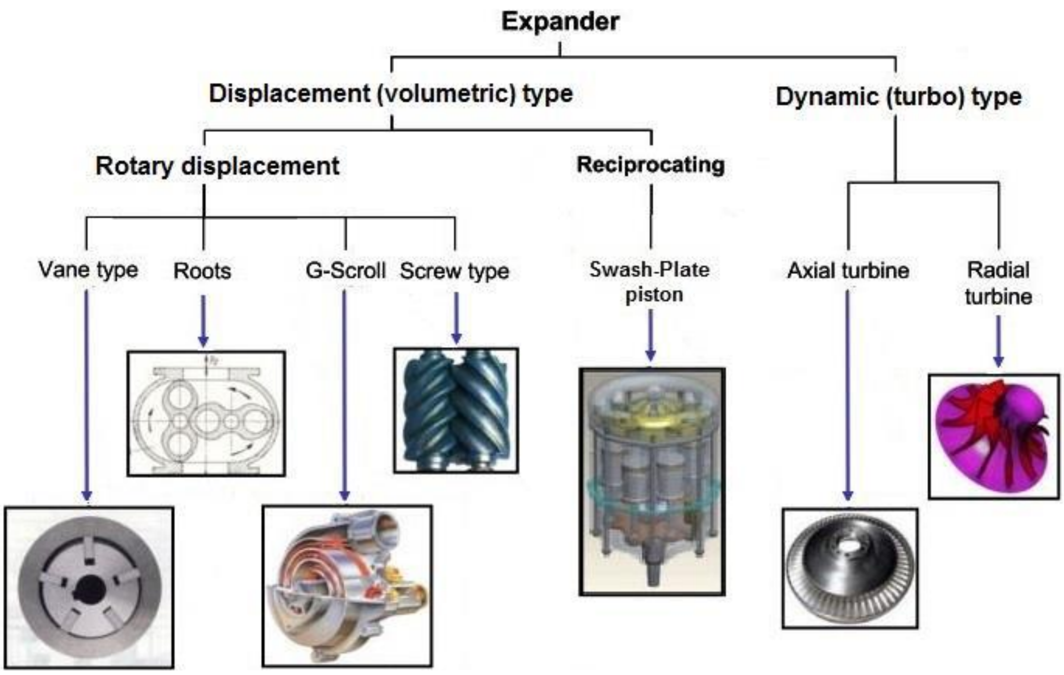

Fig 2: Diagram of the various expander types [5]

Screw: The screw expander expands the fluid continuously. It does not need any valves but at least four bearings for the two rotors. The rotors are not in contact with each other. Lubrication is required for sealing purposes. Even lubricated the necessary rotational speed is the highest for volumetric expanders. Without lubrication the rotational speed must be high (> 10,000 rpm). Therefore, standard generators are not suitable. Possible volume ratios (VRAT) are in the range of 5, efficiencies of around 50% (Eilts et. al., 2012 [16]) might be acceptable. A certain amount of wetness can be handled by a screw expander.

Scroll: A scroll expander is a comparatively simple device: it consists of two spirals, one of which is rotating. It can be mounted directly on the shaft of the generator avoiding any additional bearing. Volume ratio is below 5 (Lemort et. al., 2013 [10]). Wang et. al. (2009) [18] reported measured efficiencies in the range of 70% even for a quite small machine (< 1 kW). Droplets are no problem for a scroll expander.

Vane: The rotating vane expander is working continuously with a rather small rotational speed. Built in volume ratios are rather small (VRAT < 5). The vanes are in contact with the casing. Lubrication is required, which can spoil the working fluid. Furthermore, high friction losses and wear have to be expected. Rotating vane air motors are well known and widely used in industry. Their efficiencies are usually in the range of 30-40%. However, Badr et.al. (1984) [19] report measured efficiencies of 80%.

III. SELECTION OF THE WORKING FLUID

Working fluids were always classified into three types: dry, isentropic and wet. Organic fluids were always isentropic or dry, these fluids have better thermal performance in waste heat recovery systems than wet fluids (e.g. water) [6]. Optimal working fluid selection should be done by excluding potentially harmful fluids and by simulation and thermo-economic analysis.

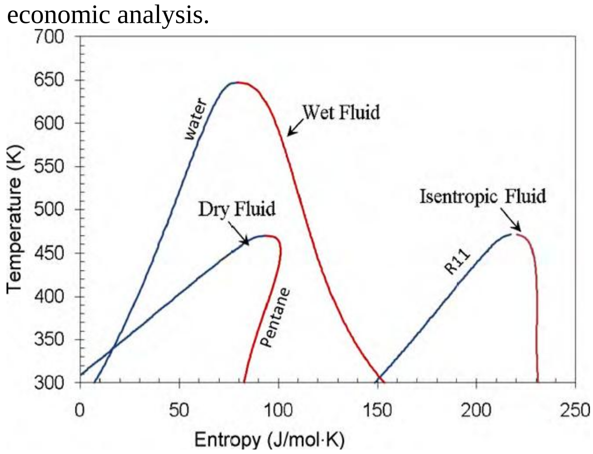

Fig 3: Three types of working fluids- dry, isentropic and wet [6]

Cycle configuration plays an important role in the selection of working fluid for an ORC system. It also depends on the operating conditions. Table I displays a list of pure working fluids used in ORCs [6]. P_c corresponds to the critical pressure and T_c corresponds to the critical temperature of the working fluid.

IV. TABLE I

PURE WORKING FLUID CANDIDATE for ORC [6]

Category and name	ASHRAE number	P_c bar	T_c °C
Hydrocarbons (HCs)			
Ethane	R-170	48.7	32
Propene	R-1270	45.3	91
Propane	R-290	41.8	96
Cyclopropane	HC-270	54.8	124
Propyne	-	56.3	129
Isobutane	R-600a	36.4	135
Isobutene	-	39.7	144
N-butane	R-600	37.9	152
Neopentane	-	31.6	160
Isopentane	R-601a	33.7	187
N-pentane	R-601	33.6	196
Isohexane	-	30.4	225
N-hexane	-	30.6	235
N-heptane	-	27.3	267
Cyclohexane	-	40.7	280
N-octane	-	25.0	296
N-nonane	-	22.7	321
N-decane	-	21.0	341
N-dodecane	-	17.9	382
Benzene	-	48.8	298
Toluene	-	41.3	319
p-Xylene	-	34.8	342
Ethylbenzene	-	36.1	344
N-propylbenzene	-	32.0	365
N-butylbenzene	-	28.9	388
Perfluorocarbons (PFCs)			
Carbon-tetrafluoride	R-14	36.8	- 46
Hexafluoroethane	R-116	30.5	20
Octafluoropropane	R-218	26.8	73
Perfluoro-N-pentane	PF-5050	20.2	149
Decafluorobutane	R-3-1-10	23.2	113
Dodecafluoropentane	R-4-1-12	20.5	147
Chlorofluorocarbons (CFCs)			
Trichlorofluoromethane	R-11	43.7	197
Dichlorodifluoromethane	R-12	39.5	111
Trichlorotrifluoroethane	R-113	33.8	213
Dichlorotetrafluoroethane	R-114	32.4	145
Chloropentafluoroethane	R-115	30.8	79
Hydrofluorocarbons (HFCs)			
Trifluoromethane	R-23	48.3	26
Difluoromethane	R-32	57.4	78
Fluoromethane	R-41	59.0	44
Pentafluoroethane	R-125	36.3	66
1,1,1,2-Tetrafluoroethane	R-134a	40.6	101
1,1,1-Trifluoroethane	R-143a	37.6	73
1,1-Difluoroethane	R-152a	44.5	112
1,1,1,2,3,3,3-Heptafluoropropane	R-227ea	28.7	101
1,1,1,3,3,3-Hexafluoropropane	R-236fa	31.9	124
1,1,1,2,3,3-Hexafluoropropane	R-236ea	34.1	139
1,1,1,3,3-Pentafluoropropane	R-245fa	36.1	153
1,1,2,2,3-Pentafluoropropane	R-245c	38.9	174
Octafluorocyclobutane	RC-318	27.8	114
1,1,1,2,2,3,3,4-Octafluorobutane	R-338mccq	27.2	159
1,1,1,3,3-Pentafluorobutane	R-365mfc	32.7	187
Hydrofluoroolefins (HFOs)			

2,3,3,3-Tetrafluoropropene	HFO-1234yf	33.8	94.7
Hydrochlorofluorocarbons (HCFCs)			
Dichlorofluoromethane	R-21	51.8	178
Chlorodifluoromethane	R-22	49.9	96
1,1-Dichloro-2,2,2-trifluoroethane	R-123	36.6	183
2-Chloro-1,1,1,2-tetrafluoroethane	R-124	36.2	122
1,1-Dichloro-1-fluoroethane	R-141b	42.1	204
1-Chloro-1,1-difluoroethane	R-142b	40.6	137
Siloxanes			
Hexamethyldisiloxane	MM	19.1	245
Octamethyltrisiloxane	MDM	14.4	291
Decamethyltetrasiloxane	MD2M	12.2	326
Dodecamethylpentasiloxane	MD3M	9.3	354
Octamethylcyclotetrasiloxane	D4	13.1	312
Decamethylcyclopentasiloxane	D5	11.6	346
Dodecamethylcyclohexasiloxane	D6	9.5	371
Alcohols			
Methanol	-	81.0	240
Ethanol	-	40.6	241
Fluorinated ethers			
Pentafluorodimethylether	RE125	33.6	81
Bis-difluoromethyl-ether	RE134	42.3	147
2-Difluoromethoxy-1,1,1-trifluoroethane	RE245	34.2	170
Pentafluoromethoxyethane	RE245mc	28.9	134
Heptafluoropropyl-methyl-ether	RE347mcc	24.8	165
Ethers			
Dimethyl-ether	RE170	53.7	127
Diethyl-ether	R-610	36.4	193
Inorganics			
Ammonia	R-717	113.3	132
Water	R-718	220.6	374
Carbon dioxide	R-744	73.8	31

Another parameter used for the choice of working fluid is on the basis of heat source temperature. According to Wang et. [21], the optimal selection of working fluids corresponding to the heat source temperature level are shown in Fig. 4 [6].

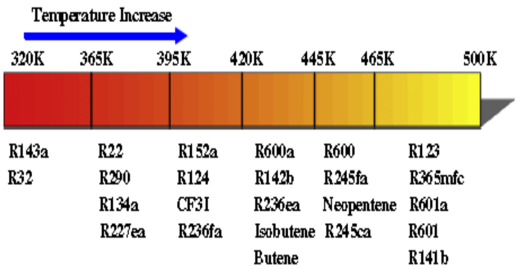

Fig 4: The optimal selections of working fluids corresponding to the heat source temperature level [6]

The selection of working fluid is done by a methodology proposed by [7], which includes parametric evaluation of the selected working fluids. In total, 7 pure organic working fluid candidates have been shortlisted for the ORC system. Selection criteria is based on the parametric analysis which includes GWP (Global Warming Potential), ODP (Ozone Depletion Potential), Safety factor, Critical temperature and pressure, Molecular mass, Atmospheric lifetime. These parameters are tabulated and compared to select the optimum working fluid having good impact on the environment as well as on the system performance. Table 2 displays the shortlisted working fluid candidates for the SORC prototype along with the parameters.

The decision criteria table is based on the favourable characteristics of the working fluid. Fluids having molecular mass below 100 kg/kmol are not selected because implementation of such types of fluids will result in high fluid velocity which in turn, lead to the high expander rotational speed. This is avoided because it will impose restrictions on the system to incorporate costly bearing and lubrication requirements as the bearing load and friction will be higher in such types of system. Only those fluids are accepted which have critical temperature above 100°C. Because, if the temperature of the working fluid rises above the critical temperature, then it is not possible to liquefy the fluid unless the pressure is reduced. The collector system used in this prototype is parabolic

V. TABLE II

LIST of SHORTLISTED PURE ORGANIC WORKING FLUIDS [7, 8]

Sl. No.	Working fluid	Molecular mass (kg/kmol)	Citical Temperature (°C)	Critical Pressure (MPa)	ASHRAE Safety group	Atmospheric Lifetime (yr)	ODP	GWP
1.	R123	152.9	183.68	3.662	B1	1.3	0.02	77
2.	R124	136.5	122.28	3.624	A1	5.8	0.022	609
3.	R125	120	66.18	3.629	A1	29	0	3500
4.	R134a	102.02	101	4.059	A1	34.2	0	3220
5.	R143a	84	72.89	3.776	A2L	52	0	4470
6.	R152a	66	113.26	4.517	A2	1.4	0	124
7.	R245fa	134	154.05	3.640	B1	7.6	0	1030

VI. TABLE III

DECISION CRITERIA TABLE [7]

Working Fluid	Molecular mass	Critical temperature	Critical Pressure	Safety Factor	Atmospheric Lifetime	ODP	GWP
R123	√	√	√	√	×	×	√
R124	√	√	√	√	√	×	√
R125	√	×	√	√	√	√	×
R134a	√	√	√	√	√	√	×
R143a	×	×	√	√	√	√	×
R152a	×	√	√	√	×	√	√
R245fa	√	√	√	√	√	√	√

trough type collector, which can increase the temperature of those working fluids which have low critical temperature, above their critical point and is not preferred because it will increase the cooling load of the heat exchanger. The accepted fluid parameters are marked as '√' and rejected fluid parameters are marked as '×'. Finally, the working fluid having the highest accepted marks is selected as the fluid for the SORC system, which is R245fa.

VII. OPERATING MAPS

In a method, proposed by [10], the selection of the working fluid and the expander can be done simultaneously by using operating maps. Operating maps for reciprocating piston expander is shown in Fig. 5.

For a given expander technology and working fluid, the operating map shows a triangular shape. The upper limit corresponds to the working fluid critical temperature. The top left-hand corner corresponds to a too high expansion ratio, while the down right-hand corner corresponds to a too high volume flow rate [10].

It can be observed from the operating map, for the evaporating temperature range of 80-150°C, the piston expander with R245fa matches well for the proposed model having evaporating temperature range of 74.47-97.79°C.

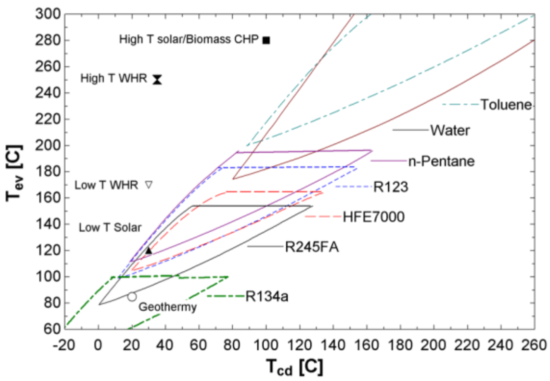

Fig 5: Operating maps for piston expanders [10]

VIII. DESIGN

A. Evaporator

The evaporator employed in the model is a line focusing cylindrical parabolic trough collector system which directly evaporates the pressurized liquid working fluid into vapour state. The diagram of the PTC system is shown in Fig. 6. This type of collector is chosen for the system because of the attainment of higher efficiency as compared to flat

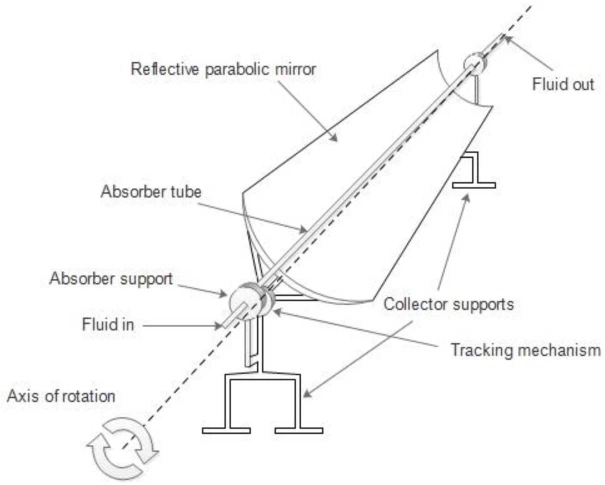

Fig 6: PTC system

plate collectors (FPC) and evacuated tube collectors (ETC), and due to its simpler tracking mechanism (single axis) as compared to parabolic dish type collectors (PDC) which involves two axis tracking mechanism.

The design has been done from the literature [4], considering 1 kW of output power. The collector efficiency is found to be in the range of 48.33-53.09%.

B. Heat Exchanger

A cross-flow, air cooled heat exchanger has been designed following literature [11], to cool the organic vapours to 50°C before entering into the condenser. It uses finned tubes and forced draft fan for cooling. Fig. 7 shows the diagram of the heat exchanger. The designed component can cool R245fa vapours from 100°C to 50°C. This can assure smooth and uninterrupted operation of the system even when the expander is not in line.

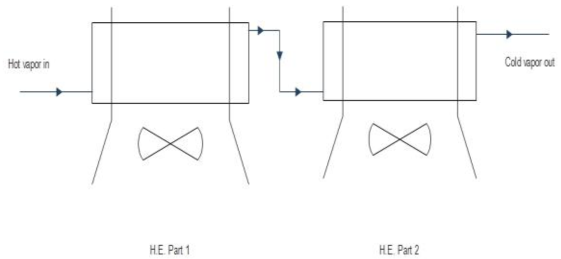

Fig 7: Diagram of the heat exchanger

The heat exchanger has two parts, part 1 and part 2 and are assembled in such a way that the outlet of part 1 becomes inlet to part 2. Hot vapours coming out from the expander enters the bottom of the header and leaves from the top of the heat exchanger. The fans attached to the heat exchangers are forced draft cooling fans which circulates the atmospheric air from the bottom of the tube bundles. Hence, overall there is a concurrent contacts. But, actually there is cross flow contact with multi-pass flow of organic vapours.

C. Condenser

The condenser for the ORC system is a shell and tube, water cooled heat exchanger which has been designed from literature [11]. It is designed to condense the organic vapours at 50°C and to provide sub-cooling to 45°C.

Although most of the existing micro-ORC setups uses Brazed Plate Heat Exchangers (BPHEs), but here in this work, a shell and tube condenser is designed. The advantages of using shell and tube condenser over BPHE are listed below [11]:

1) Low cost as compared to BPHEs
2) Internal leakage or mixing of two fluids is more common in BPHE compared to shell and tube condenser.
3) Liquid containing suspended particles tend to plug the flow area in BPHE easily and so frequent cleaning becomes necessary. In shell and tube condenser, choking can be delayed/avoided by keeping higher velocity in tubes or by selecting bigger size tubes.

The diagram of the condenser is shown in Fig 8 below. Inverted U-seal is used to provide sub-cooling. The height of the inverted U-seal determines the degree of sub-cooling.

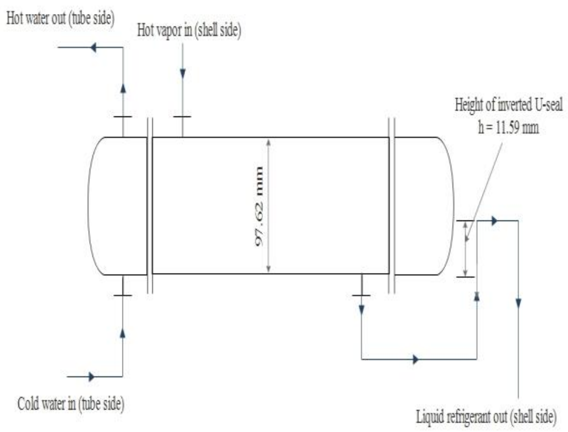

Fig 8: Diagram of the shell and tube condenser

IX. SIMULATION RESULTS and THERMO-ECONOMIC ANALYSIS

Calculations were done for the system by assuming the average values for a day of the month. The duration of the collection time is estimated using simulation results obtained from DWSIM. For 21st date of every months, from March to November, a set of data has been obtained. From this set of data, the average output power is calculated for the available collection period in kWh. This is then multiplied by 20 days to obtain the monthly average output power by eliminating 10 days of the month. Similarly this has been done for each considered months and summed to get the yearly average output power for the system. Thermodynamic analysis of the system is also done. The net work done (W_{net}), 1st law efficiency (η_I) and 2nd law efficiency (η_{II}) were computed. The results were plotted as shown in Fig. 8 & 9.

Thermodynamic analysis is done following literature [20]. The analysis is based on the following assumptions: (1) the system is operating under steady-state condition, (2) no undesired pressure drop and heat loss occur in the system, (3) working fluid at the evaporator and condenser exits is saturated, and (4) isentropic efficiencies for the turbine and pump.

With the assumptions above, the energy balance of each component based on the first law of thermodynamics is

$$\sum E_{in} + Q = \sum E_{ex} + W \qquad (1)$$

where E_{in} and E_{ex} are the energy rate in and out; Q is the heat transfer rate; and W is the power output.

The net power output of the ORC system (W_{net}) is calculated by Eq. (2)

$$W_{net} = W_{tur} - W_{pump} \qquad (2)$$

Based on the simulation results obtained from DWSim, the values of the W_{net} at different input conditions were computed. The 1st law and 2nd law efficiencies of the cycle (η_I and η_{II}) are defined by Eqs. (3) and (4)

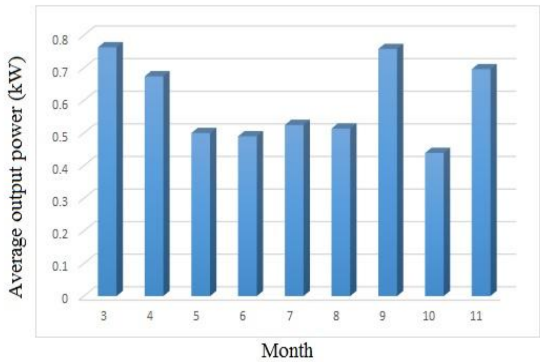

Fig 8: Average electrical output power vs. corresponding months from March to November (3-11)

$$\eta_I = W_{net}/Q_{in} \qquad (3)$$

$$\eta_{II} = W_{net}/\left[Q_{in}\left(1 - \frac{T_o}{T_m}\right)\right] \qquad (4)$$

where T_o is the ambient temperature; T_m is the mean heat source temperature; Q_{in} is the heat transfer rate in the evaporator.

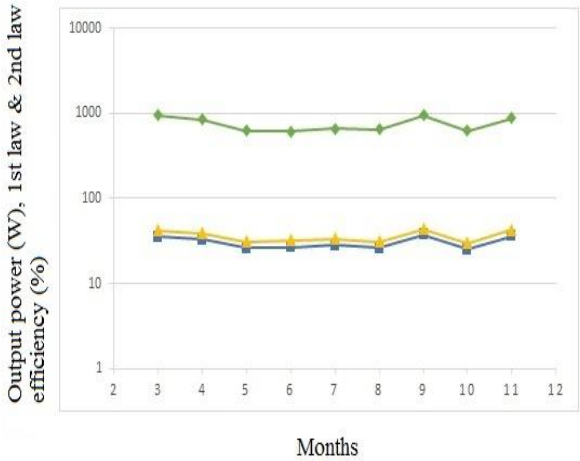

Fig 9 Expander Output Power in kW (green), 1st law efficiency in % (blue), 2nd law efficiency in % (yellow) vs. corresponding months

The simulation results in the form of power consumption of components or generation in case of expander is shown in Fig 10 at 12:30 hrs of 21st March at Kaziranga University campus.

All the values are averaged to estimate the overall system performance. This has been done prior to actual modelling so that a rough idea of the performance evaluation can be done on the system. The performance indices are the output power range, sizing, cost, efficiency, reliability, etc. The costing has been done and the details are tabulated as shown in Table 4.

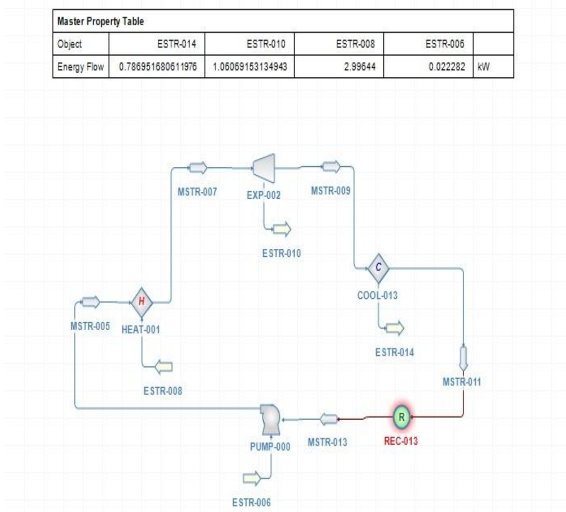

Fig 10: DWSIM simulation result obtained as of 12:30 hrs on 21st March

If the system is operated for a period of one year to generate electricity, it can save up to Rs 5,080.3 at Rs 7.9/kWh. It has been assumed that the lifetime of the SORC system is 20 years. Then, it can save up to Rs 1,01,606 on electricity bill. Payback period for the system is 17.27 years.

X. TABLE IV
THE COSTING of the SYSTEM COMPONENTS

Sl. No.	Component	Estimated cost in Rs
1.	Collector system	15000
2.	Working fluid feed pump	2500
3.	Refrigerant	3000
4.	Cross-flow heat exchanger	15000
5.	Shell and tube condenser	10000
6.	Expander	2500
7.	Instrumentation	2500
8.	Fabrication	2500
9.	Alternator	34720
	Total cost	87720

XI. CONCLUSION AND SCOPE FOR FUTURE WORK

This paper presents the design, simulation and thermo-economic analysis of a small-scale SORC. In general, the solar thermal power systems are not meant for operation without an auxiliary source of power [4]. They are either operated in hybrid mode with boilers, or other heat sources such as geothermal, engine waste heat, industrial waste heat, etc. However, this work is mainly concentrated on solar thermal power and simulation has been carried out using the predicted values of solar radiation attained for a certain duration of the year for Kaziranga University Campus. There is a scope for future to attach the system with biogas fired boiler to ensure continuous supply of power for domestic or commercial applications. This can reduce the consumption of fossil fuels and subsequently increase the production and utilization of renewable sources of energy.

Other benefits from the system can be ensured such as CHP and cogeneration of hot water, in case the system is operated at higher temperature range. For this, further works has to be done such as modification of the system components, change in the cycle configuration, implementation of auxiliary heat source, selection of suitable working fluid, automation, etc. By doing so, the hot air coming out of the cooling fan of the heat exchanger can be used for room heating purpose during winter season. And the hot water coming out of the condenser can be utilized for several useful purposes. Which will improve the overall system performance.

The T-s diagram of a typical ORC having organic working fluid as R245fa is shown in Fig 11. The four processes of the system completing the cycle are:

- Process (1-2): *Isentropic compression*
- Process (2-3): *Constant pressure heat addition*
- Process (3-4): *Isentropic expansion*
- Process (4-1): *Constant pressure heat rejection*

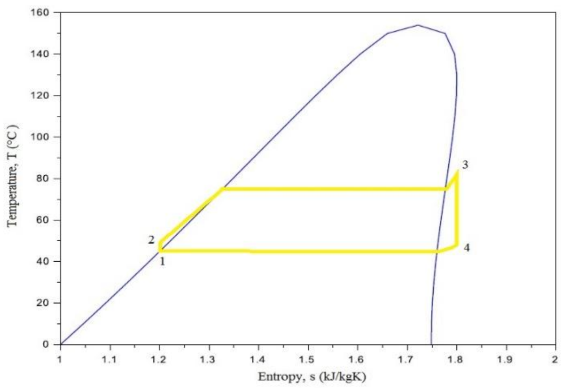

Fig 11: A subcritical ORC having R245fa as the working fluid

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