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Review and Preliminary Analysis of Organic Rankine Cycle based on Turbine Inlet Temperature

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A comprehensive review of various applications is done with an emphasis on power generation using Organic Rankine Cycle (ORC). The effect of working fluids and different configuration is analysed with varying Turbine Inlet Temperature (TIT). A parametric study for first law efficiency analysis is also done to get a better insight of the ORC. The results obtained suggest the reheat and recuperated cycle as most reasonable option for low temperature power generation. The maximum first law efficiency attained, at turbine inlet temperature of 150°C, are 8.99%, 9.68%, 9.79%, 10.39%, 10.83% and 11.48% for R-254fa, R-236fa, R-236fa, R-227ea, R-134a and R-152a respectively. The results also show that the reheat cycle should not be applied to low temperature applications.

Keywords: Rankine cycle, organic working fluid, power generation, turbine inlet temperature.

1. Introduction:

Due to dramatic rise in the crude oil prices in 1970s, there has been a search to harness various alternate source of energy[1]. Solar energy is one of the most readily available, non-polluting, alternate source of energy. It is used in various applications such as direct conversion i.e. photovoltaic or indirect conversion as solar thermal applications as desalination, day lighting and water and space heating[2]–[4]. The need for conversion of solar energy (low grade) into electricity (high grade) is of prime importance. With the use of high efficiency photovoltaic panels, available at subsidized rate due to Indian government policy, the cost of power generation in rural parts of the country are reduced. The problem with the photovoltaic panel is that it works only in availability of day light and stores energy in batteries for period of non-availability of solar radiation. This increases the cost as the batteries needs to be replaced in every three years. Solar thermal energy conversion system is used to convert the irradiative energy into heat for various application such as air drying[5], solar water heating [6] and solar desalination[7],[8], etc. These systems used various collection devices to collect the irradiative sun energy on to a receiver or absorber. The temperature achieved in these collection devices are in the range of 80 -90°C for flat plate and 130-200°C for parabolic concentrating collector [9][10][11]. In recent decade, various cycles have been proposed for the usage at low temperature power generation namely, Trilateral Flash Cycle [12][13], Kalina Cycle [14][15],

Thermo-Acoustic Engine [16] and Organic Rankine Cycle [17]–[19].

Organic Rankine Cycle (ORC) is very similar to Rankine Cycle (RC) in terms of its working principle and processes involved. The ORC uses organic fluids such as Hydro-Carbons (HC), Hydro-Chloro-Fluoro-Carbons (HCFC), Fluorocarbons (PFC) and Zeo-tropic mixtures (i.e. mixture of HFC, HC and PFC) as the working fluid instead of water.

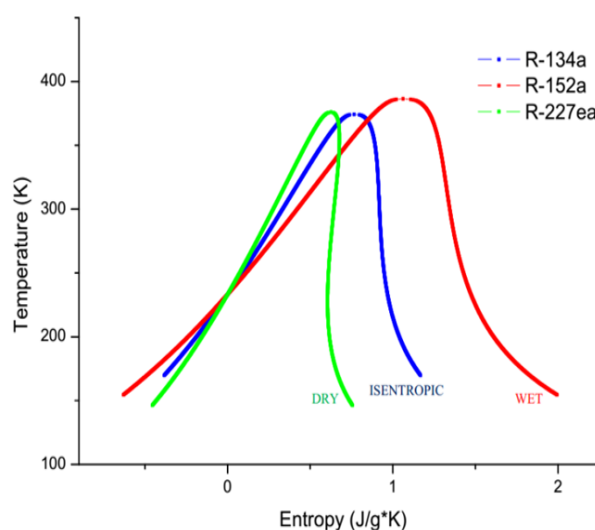


Figure 1: Classification of working fluids

The reason for using organic fluids is positive and isentropic slopes of their saturation curve which enables

them to harness the energy at lower temperature heat source such as waste heat recovery [20], ocean thermal energy conversion [21], geothermal [22], solar thermal power [23], etc.

The working fluid is most important factor in exploiting the low temperature sources in the best possible way. The working fluids are characterized as wet, isentropic and dry fluids depending upon the slope of the saturation curve in TS chart[24][25] as shown in the Figure 1. Working fluid like R-152a, when expanded in an isentropic process from a saturated vapor line, the outlet of the expansion device is in wet region due to the negative slope of the saturation curve thus these type of working fluid is termed as wet working fluid. Fluids like R-134a tends to have the outlet of an isentropic expansion process on the saturation curve thus is termed as isentropic working fluids. These fluids have infinite slope in considered temperature range (i.e. TIT= 150°C and TOT= 40°C). Similarly fluid like R-227ea have the outlet of the isentropic expansion in the dry or superheated region due to the positive slope of the saturation curve in the considered range of temperature. These fluids are termed as dry fluids and these fluids are mostly used in the ORC's.

2. Applications of ORC

2.1. Waste heat recovery

There are various applications suggested in the literature where ORC proved to be a good option for harnessing the energy into useful work. One of such option is in harnessing the waste heat recovery in process industries and has been investigated in 1995 [26]. The use of a turbo generator for lower transmission losses was suggested due to better maintenance in the capacity range of 100 kW. The working fluid selected for the analysis was R-114 initially but due to the international compulsions, the alternative for R-114 was found to be toluene due to its stability in the test range of temperature up to 400°C.

Later ORC was used in Molten Carbonate Fuel Cell (MCFC) plant for waste heat recovery[27]. They suggested that for a low capacity fuel cell plant of about 5 MW have a bottoming cycle in the range of 650 KW, having waste heat temperature of 450 to 600°C. They found that up to 5MW, the efficiency gains are in the range of 10-15%.

Theoretical and experimental simulation were performed to analyse the effect of heat source temperature, evaporating pressure of working fluid and evaporator size on the first law efficiency of the cycle [28]. It was found that the high evaporating pressure leads to high efficiency both experimentally and theoretically. The maximum overall efficiency was found to be around 5% for heat source temperature of 85°C and condenser pressure of 0.45MPa. It was proved that for non-isothermal source of heat extraction, evaporator leads to maximum exergy destruction and pump was least influential.

Thermodynamic simulations were performed for the

ORC optimization using waste heat from Gas Turbine Modular Helium Reactor (GT-MHR)[20]. The energy loss in preheater was around 50% for GT-MHR which was 4 to 19% in case of GT-MHR ORC combined cycle, since energy is utilized in heating ORC fluid. There was an increase of 5 to 10% in efficiency of both first law and second law. The simple ORC was found to be the best suited from thermodynamic and economic point of view, whereas the recuperated ORC was found suitable for CHP applications.

An ABB ORC power plant was constructed in order to increase the efficiency of cement plant by utilizing the waste heat from the preheater [29]. The plant produced 9 MW or 10.5 GWh/ year of electrical power. The results suggest that the ORC power plants can boost the electrical efficiency of the plant by 20%. ORC plant not only reduced the CO₂ emissions but also reduces the water usage.

A different kind of waste heat recovery from the flushing cinder water using double organic Rankine cycle (DORC) was simulated in MATLAB and REFPROP for pinch point temperature difference analysis [30]. Since the flushing cinder water was not available for the half of the time of operation, the use of storage tank was essential for continuous power generation and cycle operation. Thus the use of double cycle was preferred. The optimal evaporating temperature was found to be 330.15 K. The use of dry and isentropic fluids were suggested over the wet fluids for the minimum temperature difference between fluids of heat exchanger.

The waste heat generated during the enhanced crude oil development was exploited for power generation of around 600-900 kW [31]. The trans-critical ORC used R-134a as working fluid optimized for heat to power conversion. The change in ambient temperature from 10°C on cold night to 28°C on hot day showed the deterioration of power in order of 200 kW for same design. A large heat exchanger was suggested for warmer ambient condition due to the fact that power increase for increased heat transfer performance. The degradation of heat source in the range of 6 K equate the net power output for smaller heat exchanger.

Corn cob biomass waste was used as heat source for ORC evaporator and theoretical farm area was obtained for varying electrical power, thermal efficiency and mass flow rate [32]. The temperature of HTF was varied from 110°C to 130°C with mass flow rate from 100 lit/min to 200 lit/min. The working fluid used was R-141b. The maximum efficiency was achieved at maximum temperature and flow rate. The corn cob appears to be a feasible option with 8.5 kW electrical power at an efficiency of 10.3% utilizing corn from a farm area of 101.1 hector/year for rural society under consideration.

The ORC could also be employed as bottoming cycle for small commercial gas turbine of 100 kW for improvement of efficiency to make it at par with internal combustion engine [33]. The optimized bottoming cycle was tested for six working fluids with different turbo expanders to find the best suited for selected working

fluid and operating conditions. The turbines (axial or radial) had an advantage over the scroll machines as low density of fluids need large displacement and thus needs parallel machines.

A micro-CHP system for residential complex using ORC powered by natural gas fired boiler was investigated experimentally [34]. The heat source temperature was varied from 65°C to 85°C for maximizing electrical power. The maximum electricity generated at a heat source temperature of 84.1°C was 77.4 W at an efficiency of 1.66%. The result suggested that optimization for net efficiency by changing expander generator set could be possible.

2.2. Ocean Thermal Energy Conversion

In another application ORC was applied for power generation in ocean thermal energy conversion[21]. The net power output was optimized for the fixed evaporator and condenser temperature range. The results showed that the ammonia was best suited for the optimized OTEC power generation.

Two case studies on simple OTEC and OTEC combined with solar collectors were studied to increase the efficiency[35]. The results suggested the selection of proper working fluid in the available range of TH and TL. The use of wet fluids in the OTEC was suggested because no regeneration was considered. The dry fluids tends to have a superheated vapor at turbine outlet and with no regeneration load on the condenser increases decreasing the overall efficiency.

An objective parameter of net power output to total heat transfer area was used to suggest the optimal operating parameters for the OTEC plant[36]. The results were evaluated with warm sea water inlet, cold sea water inlet temperatures at 25 to 28°C and 5 to 8°C respectively. Ammonia was suggested as the most optimal working fluid followed by R-152a, R-600a, R-134a and R-245fa. The results further suggest that the performance of the OTEC is more strongly affected by the cold sea water rather than warm sea water.

The OTEC application was always considered in the region of warm sea water but the simulation suggested that the application of steam condenser outlet in the evaporator allowed the use in colder regions as in case of Punta Alcalde station in Chile[37]. The results of simulation suggested an increase in the efficiency of 1.3% with incorporation of OTEC and use of flash evaporator allows the desalination of 5.8 million tons per year.

Feasibility of a C-OTEC was studied and analysed for increasing the power output by extracting the heat at the condenser for heating of ORC[38]. The efficiency of primary rankine cycle also increases due to reduced condenser pressure with C-OTEC. The suggested system had a positive effect on the cycle but also delivers an additional power.

A model of EP-OTEC to analyse the effect of using liquid vapour ejector and motive pump to improve the

efficiency of OTEC was developed [39]. The system used R-152a as the working fluid to develop a gross power of 20 kW by varying the heat source temperature from 24°C-29°C keeping the condenser inlet from deep sea level at 5°C. The results suggested an improvement of 38% in EP-OTEC overall efficiency from the OTEC.

2.3. Geo-Thermal heat Application

An initial feasibility of the geothermal ORC was analysed in 1989 [40]. They suggested the use of ORC on downwell pump to exploit the geothermal resource available in Mexico. They only disadvantage that could be defined was the use of expensive technology over already available flash cycle.

The 5.4 MW ORC using geothermal as the source was dynamically modelled [41]. The working fluid for the simulation was pentane. The model employed PID Simulink block (P: 0.8, I: 0.01, D: 0.01) as controller. The results indicated that the model with feedback control gives better stability and matches better with observations made in the plant. The average percentage error at the outlet pressure is around 3.5% which is also very less compared to a maximum error of 9.3%.

The Zeotropic mixtures were used as the working fluid in the extraction of heat from low enthalpy geothermal source [42]. Results suggested the non-isothermal evaporation due to glide match, leads to the increase in efficiency and decrease in the irreversibility in condenser particularly. The increase in exergy efficiency was estimated in the range of 5 to 15% as compared to the most efficient pure component of the mixture.

For validation of an existing ORC using low temperature geothermal source, a model was developed based on 7200 measured data in a year [43]. The results of the study indicated that a hybrid solar geothermal source was up to 3.4% efficient in second law analyses as compared to geothermal source alone. The hybrid system had an efficiency increase to 17.9%. The results suggest that hybridization should be adopted over solo geothermal or solo solar source

2.4. Combined Heat and Power Application

Similarly the waste heat recovery from biomass for combined heat and power application using the turbo generator was being presented in 3rd Munich discussions and meeting on energy conversion from biogas at Munich Germany [44]. The ORC was compared with the Stirling concept engine. The ability of the working fluid to retain its thermal stability was the limiting factor for maximum cycle temperature in ORC application. Biomass combustion require larger boiler for achieving high temperature due to its lower calorific value[45]. No sliding parts in ORC also provide low frictional resistance to the particulate flue gases of biomass combustion. The author compared the various methods in which the ORC can be used with biomass i.e. direct electricity, combined heat and power (CHP) and with biogas burner, etc. and suggested that it is better to be

utilized in combined heat and power application.

In the town of Lienz the demand for power and district heat demand was met by the biomass CHP plant by supplying a heat of about 60 TWhr and an electricity of 7.2 TWhr [46]. The fuel used in thermal oil boiler and hot water boiler of 6.5 MW and 7.0 MW capacity was biomass. In order to increase the efficiency of the CHP plant the flue gases were allowed to pass over economizer to recover waste heat at nominal capacity of 1.5 MW.

The ORC was coupled to a refrigeration cycle with a common condenser [47]. In order to eliminate the use of electrical power for compression, the compressor is directly coupled to turbine. The working fluid was also selected as R-134a which is same for both refrigeration and power cycle. COP of different cycle configurations were studied and results suggested that COP of the cycle can be increased to 80% if recuperator, re-heater and economizer are installed. The COP of the modified system is varied in the range of 0.84 to 2.18.

2.5. Solar Thermal and Applications

The use of ORC for power generation using solar energy with working fluid R-114 was analyzed for the climatic condition of Kuwait [48]. Two types of heat rejection sink namely air-cooled and water-cooled were used for net heat collection of ORC at a stratified temperature of 100°C which had been taken on past experience basis. The cumulative energy output at 100°C and 20°C wet bulb temperatures was 100 kWh/m² higher for water-cooled than air-cooled due to comparatively lower temperature achieved in the condenser.

Various configuration of ORC i.e. simple ORC, recuperated ORC and Reheat and recuperated ORC have been simulated in Aspen Technology Simulation package ASPEN thermal modelling software [49]. The corresponding temperature of the solar source was taken as 304°C which is on the higher side of the technology available. The analysis was performed for 1 MW of net power available and obtained the efficiency of 23% and with the existing technology of parabolic trough collector proves to be a viable option.

In the experimental study for generation of fresh water along with power generation using ORC with R-245fa as the working, suggested the optimization of collectors slope to be at 35° increasing the annual fresh water production by 1.38% [50]. The increase in the collector area leads to the increase in the fresh water production, whereas the efficiency of the system is reduced.

The heat transfer mode, evaporation temperature of working fluid along with the tilt factor adjustment were asserted as the key factor for the optimization of low temperature solar thermal electric generation using CPC collector [23]. The electric efficiency and work output of the system increases with rising evaporating temperature and then eventually drops. The optimal evaporation temperature of the working fluid (R123) was found to be around 120°C, for cities like Bombay, Lhasa Singapore,

etc.

A hybrid solar thermal-PV RO plant was constructed and feasibility study was made for Chalki island case in Greece [51]. The results suggested that the cost effectiveness was not achieved in the hybrid plant but a competitive price of water production could be achieved with the subsidy for renewable, decrease in the parabolic trough cost and increase in capacity factor. The Levelized Cost of water (LCW) was used as the comparative parameter which includes both the cost of investment and the operation and maintenance cost. The LCW was reduced by 27% with 80% subsidy and 10% with a reduction of 20% cost of parabolic trough collector. With both the cost reduction taken into consideration the LCW reached a competitive price of 8.35 €/m³.

Supercritical ORC was coupled with a RO unit to cogenerate power and fresh water achieving a cycle efficiency of 21% at no load conditions [52]. The optimized efficiency was found to be around 14% even for the low solar radiation. The power generated from the system was used for the fresh water production at a constant rate of 40 m³/hr.

In order to increase the total power production from a concentrating photovoltaic modules, ORC was employed to recover waste heat generated. [53]. The results indicated that the overall efficiency of CPV-ORC was 2.5% higher than CPV. The working fluid used was R-245fa. The annual cost of energy production was 0.113 €/kWh for CPV-ORC which was much lower than 0.147 €/kWh for CPV.

The use of High Concentration Photovoltaic (HCPV) cells in power generation was wasting 60% of its energy as heat [54]. This heat can be extracted with a liquid cooling or can be used for power generation with Micro-ORC. The analysis suggests that MORC gets 9% more net power generation efficiency than the liquid cooled HCPV at the normal room temperature. The working fluid suggested based on the micro channel boiling analysis is R-365mfc.

A successful investigation for the use of a hermetic sealed compressor as hermetic scroll expander with an isentropic efficiency of 63- 65% in the pressure ratio of 2.4 to 4 lead to the use of ORC in Low temperature applications with small and micro power generation demands [55]. R-134a was found suitable for power generation in the low temperature range due to its dry nature at the exit of the turbine, increasing the life cycle of the turbine.

All the above stated examples indicate that ORC is a feasible solution for power generation in low temperature for some extra power in existing high temperature power generation applications. The suggested working fluid for various applications along with their critical pressure and temperature and Environmental impacts as GWP and ODP are listed in the table 2.1

Table 2.1: Working fluid for various applications

Application	Working Fluid	ODP	GWP (over 100 yr)	Critical Temperature.(°C)	Critical Pressure (MPa)	Operating Range(°C)
Solar Thermal	R-134a	0	1430	101.08	4.06	50 to 150
Geothermal	Pentane	0	11	196.55	3.37	100 to 140
CPV/Thermal	R-245fa	0	950	155.01	3.65	70 to 150
Bio Fuel	Toluene	0	0	318.6	4.13	200 to 350
Waste heat Recovery	R134a	0	1430	101.08	4.06	50 to 150
	R245fa	0	950	155.01	3.65	125 to 270
	Toluene	0	0	318.6	4.13	250 to 400
Ocean Thermal	Ammonia	0	0	132	11.3	15 to 25

3. Effect of working Fluid

A number of working fluid (i.e. 20) used in the low temperature solar ORC have been compared on the basis of volume flow rate, efficiency, mass flow rate, pressure ratio, toxicity, flammability, GWP, ODP [56]. R-134a outperforms as the most compatible alternative for solar application. Some of the other working fluid after R-134a that can be used with a precaution for their flammability are R-152a, R-600, R-600a, and R-290.

The use of dry hydrocarbons was suggested in Organic Rankine cycle [25]. The effect of critical temperature of different fluids was examined for thermal and exergetic efficiency. The use of recuperator is a good option for dry hydrocarbons as the efficiency is increased. The results show that hydrocarbons outperformed some refrigerants.

A study to suggest the working fluids for solar organic Rankine cycle used Refprop 8.0 database with 117 organic fluids [57]. The working fluids were categorized into refrigerants, high performance non- refrigerants and medium performance non refrigerants. The selection was based on the order of highest thermal/exergy efficiency, then highest net power output and with lowest vapor expansion ratio. The results showed that the enhancement of exergy efficiency is limited to 5% when the collector efficiency was increased from 70% to 100%.

Nine working fluids were compared for different condition in the utilization of the heat available at the engine exhaust [58]. The power output was fixed at 10 kW. The impact of the working fluid on the environment and safety levels were also examined. R-245fa and R-254ca proved to be the most environment friendly for waste heat recovery trough engine exhaust. R-11 was also suggested having higher performance in thermal analysis.

A preferable ranking was given to the 8 mostly used working fluids based on the spinal point method [59]. The working fluids in preferred ranking were HFE-7000, HFE-7100, PF-5050, R-123, n-pentane, R-245fa, R-134a. The selection criteria for ranking was low environmental impacts, higher thermodynamic efficiency, low boiling point, and safety standards. This does not include the specific requirement for the heat source and thus the ranking may change according to the specific heat

source.

CO₂ was mixed with the hydrocarbons for suppressing flaming properties, to be used in organic Rankine cycle as working fluid [60]. The mixture had 70% mole fraction of hydrocarbons and 30% of CO₂. Propane mixture showed higher efficiency than that of iso-pentane. The irreversibility analysis indicated that the main source of irreversibility was regenerator.

A simulation of two stage Organic Rankine Cycle with internal heat recovery used simulation package Aspen plus (V7.3) to present the effect of intermediate condensate pressure [61]. The author suggested that saturated two stage cycle out performs standard and recuperated cycles in both the studies i.e. fixed source temperature and fixed pinch difference of 10 K. The efficiency enhancement was found to be 2.5% over other cycle configuration.

The performance of ORC is largely dependent on the working fluid properties [62]. The use of silver Nano-pentane as working fluid for CHP was suggested and tested for both the power only and CHP applications. The results indicated that pentane alone was good but became better with the addition of silver Nano particles. The increase in efficiency with silver Nano pentane was around 10%. The use was restricted with the rise in the cost of the plant with additives.

Zeotropic mixtures having different phase change process with a variation in temperature of evaporation were used as working fluid [63]. The thermal model proposed and programmed in MATLAB 2010a was verified by data available in theoretical studies. The improvement in the performance of the cycle using zeotropic mixtures had a positive correlation with the temperature glide. Jacob number used to predict the performance of an ideal cycle not having a rigorously negative correlation with thermal efficiency.

An analysis was performed to find the operating conditions for solar operated ORC using mixtures of working fluids [64]. A total of 15 mixtures were selected out the initially suggested 91 mixtures on the basis of maximum operating power. The NeoPentane-2-FluoroMethoxy-2-MethylPropane mixture 70% Neo-Pentane was found to the best with maximum power generation.

4. Different Cycle Configurations

There are various methods suggested to increase the efficiency of a thermodynamic cycle. One of them can be by changing the configuration of the base cycle (i.e. by exploiting available energy in the system or by adding some more amount of energy to it). The various configurations achieved by making a modification in the base cycle are examined to evaluate the best suitable cycle configuration to harness energy at low temperature source.

4.1. Simple ORC cycle

Figure 2 shows the schematics of a Simple Organic Rankine Cycle (SORC), configured with an expansion device, condenser, pump and a boiler. The working fluid leaving the turbine is cooled in the condenser at constant pressure to change the phase of the working fluid from saturated or superheated vapor to saturated liquid. This saturated liquid is then pumped to the boiler pressure in an isentropic compression process. In boiler the working fluid is heated to the desired inlet temperature of the turbine. In turbine an isentropic expansion process extracts the work from the working fluid. The processes are same as in any Rankine cycle i.e. an isobaric heat rejection in condenser (process 1-2), isentropic compression in pump (process 2-3), isobaric heat addition in boiler (process 3-4) and isentropic expansion in turbine (process 4-1) as shown in Figure 3

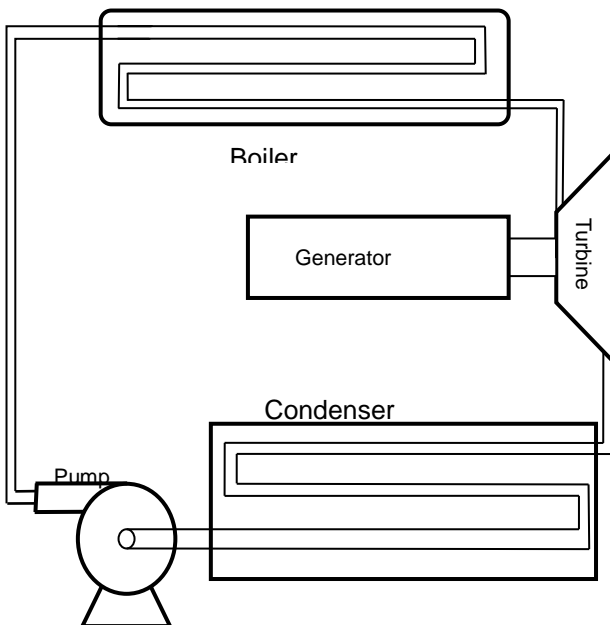


Figure 2: Schematics of SORC

This simple cycle is base cycle used in various applications of ORC. Energetic and exergetic analysis using two simple ORC cycles was performed for the solar-geothermal source. The energy efficiency improved from 16.5 to 74% but the exergy efficiency gave an insight of the system with an increase from 26.5 to

36.0% in multi generation system [65].

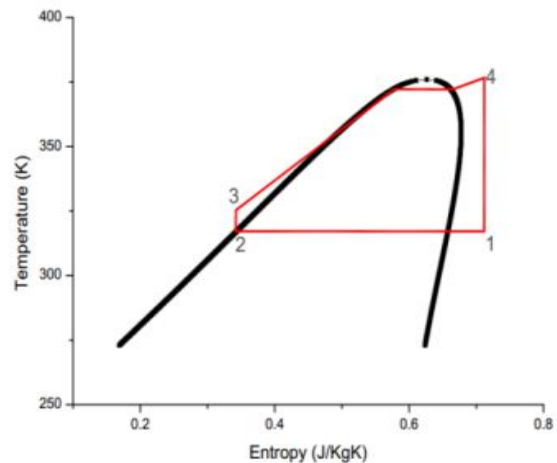


Figure 3: T-S chart of SORC

Micro-scale ORC was design to make use of concentrated Photovoltaic modules for higher power generation efficiency. The system recorded an increase of 8.8% efficiency than normal liquid cooled high concentration photovoltaic system [54]. The simple cycle is not only used as an additional cycle to improve the efficiency of a working cycle or power generation system but is also used as a standalone system. A parabolic trough concentrated solar thermal power generation unit with R-245fa as the working fluid and green machine 4000 series by Electra-Therm Inc. is installed in Louisiana, generating 50kWe [66].

4.2. Recuperated ORC cycle

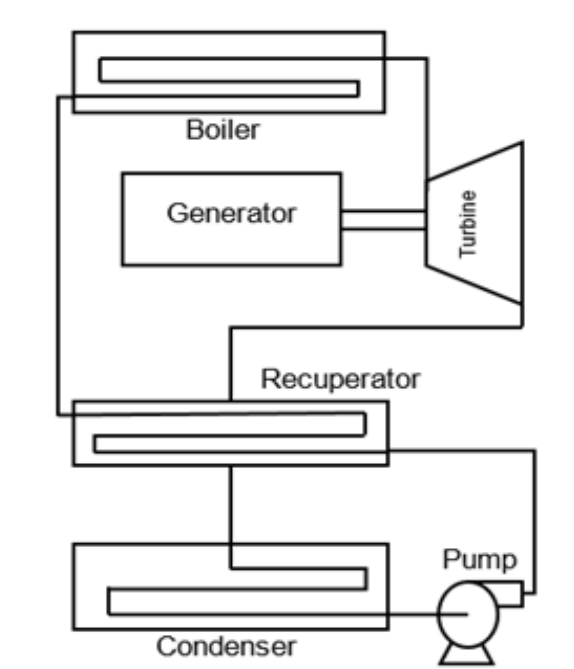


Figure 4: Schematics of RORC

Recuperated cycle is an advanced version of the simple cycle. The heat still available in the exhaust of the turbine is extracted by the introduction of a recuperator,

to heat the inlet of the boiler as shown in figure 4. The cycle is similar to simple cycle i.e. the saturated or superheated working fluid enters the turbine, work is done on the turbine blade generating mechanical power to be converted into electrical power with the help of coupled generator. The exhaust of the turbine is superheated and still have potential to work as the preheating fluid for the low temperature saturated liquid available at the condenser exhaust. Thus a heat exchanger termed as recuperator or internal heat exchanger is used to transfer heat to the liquid entering the boiler before going to condenser. The vapor is condensed into liquid in condenser, which is pumped to boiler for heating and recirculation.

The T-S chart of the cycle is shown in Figure 5. The process 1-2 and 4-5 is constant pressure heat exchange process in the recuperator. The other four processes are two isentropic processes i.e. expansion in turbine (6-1) and compression in pump (3-4) and two constant pressure heat addition in boiler (5-6) and heat rejection in condenser(2-3). Recuperated cycle works best with dry working fluids whereas the wet and isentropic fluids needs to be in the superheated region before expansion to work on the recuperated cycle which in turn increases the burden on the boiler.

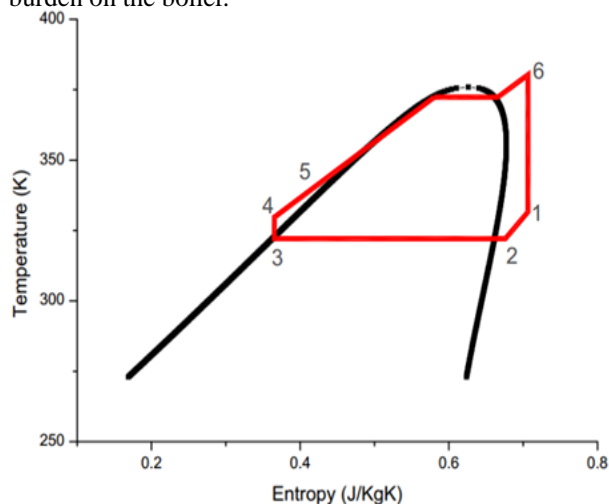


Figure 5: T-S chart of RORC

Recuperated cycle was utilized to generate power from low grade waste heat in ABB Switzerland Ltd. cement plant. The plant raise the electrical efficiency up to 20% of plant capacity [29]. In another application of recuperated cycle it was utilize in combined heat and power application using biomass boiler as the source of heat. The system was designed for the economic feasibility and appears to be an attractive option for alternative power generation [67]. Different expansion devices have been studied for the bottoming of the design parameters of a recuperated cycle. The study reveals that the axial and radial turbine architecture is best suited with R-245fa, iso-pentane and iso-butane, whereas the scroll machines have limited applicability due to high expansion ratio [33].

4.3. Reheat ORC cycle

In reheated cycle, as shown in Figure 6, there are two turbines namely high pressure and low pressure turbine depicted by T1 and T2 respectively were used. The exhaust of the high pressure turbine T1 is fed to the re-heater for increasing the heat content at constant pressure, and then it is fed to low pressure turbine T2 to extract the power to its fullest extent. The same generator is coupled to both the turbine so the RPM of both the turbine should remain same. The vapor leaving from the low pressure turbine T2 is condensed in condenser before isentropic pumping to boiler.

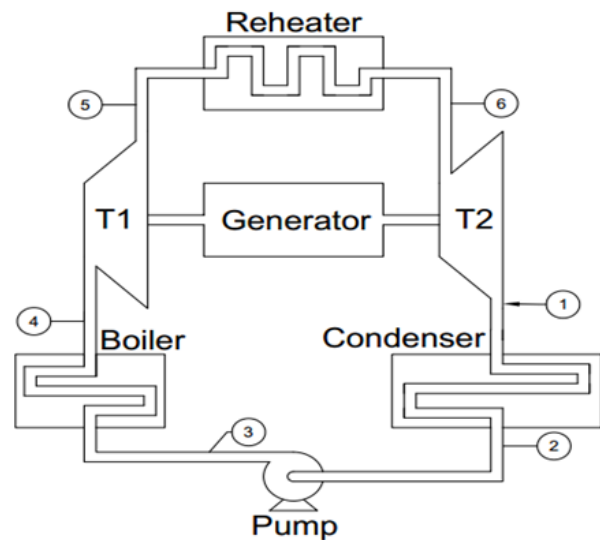


Figure 6: Schematics of ReORC

The reheat process is shown in Figure 7 on a T-S Chart by process 5-6. The remaining process is same as simple cycle except that there are two isentropic expansion processes i.e. 4-5 and 6-1. Process 1-2 and 3-4 are constant pressure heat rejection and heat addition process in condenser and boiler respectively. The isentropic pumping is process 2-3.

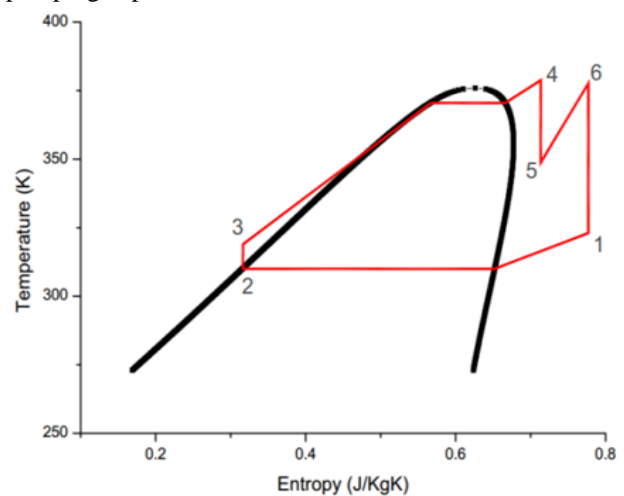


Figure 7: T-S Chart of ReORC

First law analysis of coupled power refrigeration suggest that the use of reheat cycle will improve the COP of the plant by 80% with economizer being used [47].

The analysis compared different cycles for replacement of the condenser in the vapor compression refrigeration yielding 60% improvement by recuperated cycle. Reheat cycle is also used in process industries with waste heat sources at various positions to be used as reheating sources as in case of post combustion capture process of carbon dioxide. Reheat cycle achieves 46% higher second law efficiency than simple cycle and generate more electricity[68].

4.4. Reheat and recuperated ORC cycle

Recuperated and reheat cycle is combination of the two cycle, the isentropic working fluid from the reheat cycle also have some heat still available that goes waste in condenser. Thus to utilize the heat content available at the exhaust of the low pressure turbine T2 recuperator is introduced as shown in the Figure 8.

The liquid thus leaving the condenser is pumped to boiler through the recuperator working as a pre-heater to heat the working fluid. Thus reducing the heating load of the boiler. The saturated vapor leaving the boiler is fed to turbine T1 for isentropic expansion to pressure P2 and

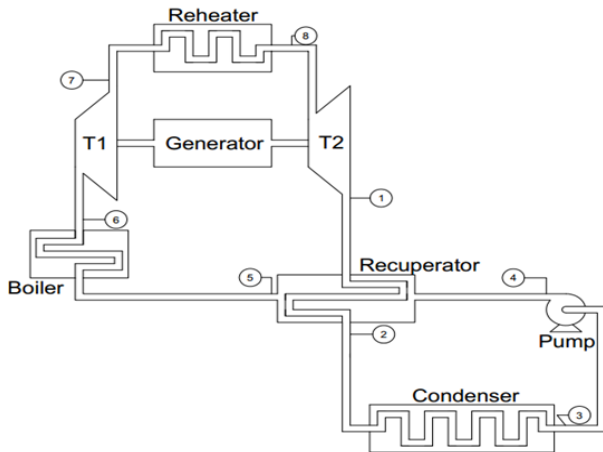


Figure 8: Schematics of RRORC

then heated to same temperature in re-heater. The superheated vapor at low pressure is fed to turbine T2 to complete the cycle by isentropic expansion. The dry working is most suitable for this cycle though the use of reheat and recuperated cycle is dependent on various factors like the availability of source for the reheating and cost of the equipment and space availability etc.

Performance analysis of Different ORC's have been studied and it is found that a conventional steam Rankine cycle can achieve 20.6% efficiency whereas the combined reheat and recuperated ORC cycle can achieve as high as 28.5% for complex heat source as Fisher Tropsch Plant [69].

5. Mathematical Model for Parametric Analysis

The analysis of various cycle configuration is done to find the best suited cycle configuration for the following

assumptions

1. Turbine Inlet and outlet Pressure of 2.5 and 1 MPa.
2. Turbine Intermediate pressure of 2 MPa.
3. Turbine Inlet Temperature varying from 100°C to 150°C.
4. The efficiency of turbine and pump is 85% and 60% respectively.
5. The effectiveness of recuperator is 0.80.
6. The mass flow rate is kept as 1 kg/sec.

The equations used in the mathematical model are general heat balance equation written for different components, namely turbine, recuperator, condenser, pump, boiler and reheater, as listed below:

Turbine work and efficiency from eqn. (1) and (2) resp., is used to calculate the exit condition of the working fluid

$$\dot{W}_t = \dot{m}_f (h_o - h_i) \quad (1)$$

$$\eta_t = \frac{(h_i - h_{o,a})}{(h_i - h_o)} \quad (2)$$

Condenser heat is used for calculating mass flow rate of the cooling fluid as shown in eqn. (4).

$$Q_c = \dot{m}_f (h_{i,a} - h_o) \quad (3)$$

$$Q_c = \dot{m}_c C_{pc} \Delta t \quad (4)$$

The work input to pump is given in eqn. (5)

$$\dot{W}_p = \dot{v}_f (P_b - P_c) \quad (5)$$

The heat input in the boiler and re-heater is evaluated using eqn. (6)

$$Q_b = \dot{m}_f (h_o - h_i) \quad (6)$$

The re-cuperator have a cold side (i.e. from pump to boiler) and a hot side (i.e. from turbine to condenser). The heat transfer from cold side to hot side is given in eqn. (7)

$$Q_r = (h_o - h_i)_{co} = (h_o - h_i)_{ho} \quad (7)$$

The efficiency of the overall cycle is given in eqn. (9)

$$\eta_l = (\dot{W}_t - \dot{W}_p) / (Q_b) \quad (8)$$

where,

\dot{W}_t work from both turbines

\dot{W}_p work from pump

Q_b Heat added in boiler and re-heater

6. Results and Discussions

Different working fluids have been tested for different configuration. The turbine inlet temperature is varied to find the optimal temperature and cycle configuration.

6.1. Effect on heat addition in the boiler:

With the change in turbine inlet temperature, heat addition in the boiler is also increased to meet the requirement of higher temperature. The Figure 9 shows the heat addition for R-134a. The trend seems to be similar in all the cycles but the slope of reheat cycle is the maximum and thus is the main reason for decrease of efficiency in reheat cycle. The reheat cycle is expected to increase the work output but the expense of energy to gain the small change in the network is more. It can be observed that the recuperation is a good option as it reduces the heat required to achieve the outlet temperature significantly, whether it be simple cycle or be the reheat cycle. The trend for the recuperation is almost parallel. Thus the reheat and recuperated cycle is suggested over the other cycles.

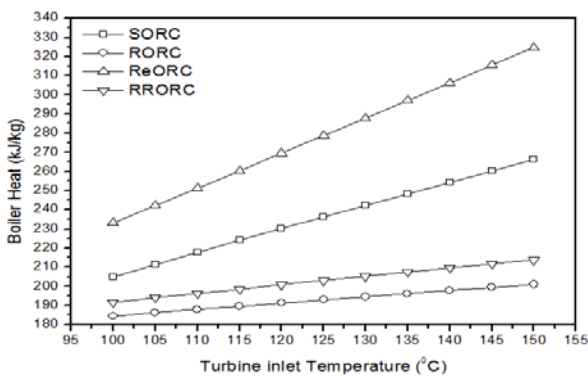


Figure 9: Heat addition in the boiler

6.2. Effect on the heat removal in condenser

The trend is repeated in the condenser while rejecting heat as can be seen in Figure 10. The maximum amount of heat is rejected in the reheat cycle configuration. The slope of recuperated cycle and reheat and recuperated cycle is almost same thus indicating that the recuperation is a good option for reduction in the size of the condenser plant. The reheat cycle is not suitable as the size of the condenser plant will increase on account of large amount of heat to be rejected. The maximum amount of heat to be rejected is 300 kJ/kg for refrigerant R-134a in reheat cycle and the minimum amount of heat to be rejected is 170 kJ/kg at TIT of 100°C for recuperated organic cycle.

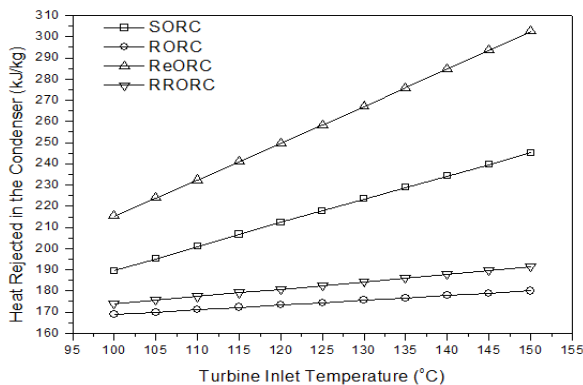


Figure 10: Heat rejection in Condenser

6.3. Effect on turbine work with heat in boiler

In order to analyze the effect of reheat with the increase of turbine inlet temperature, turbine work and heat addition in the boiler are plotted as shown in the Figure 11. It is observed that with the increase in turbine inlet pressure heat addition in the boiler and turbine work also increases. With the increase in TIT, the net change in the work of turbine is decreases, whereas the heat addition in the boiler increases and thus is the reason for decrease in the efficiency of the cycle. The recuperation in both the cases reduces the heat addition in boiler significantly.

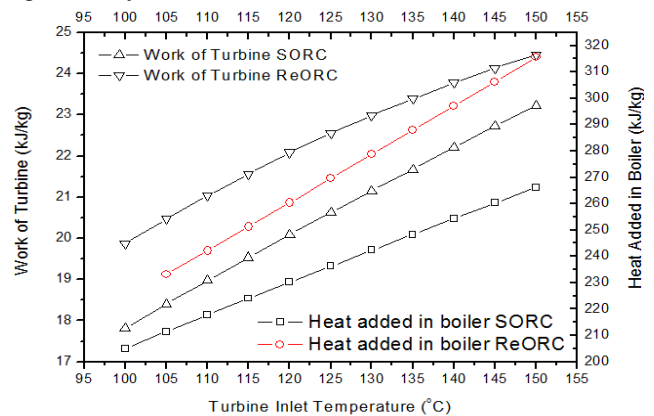


Figure 11: Effect of Reheat

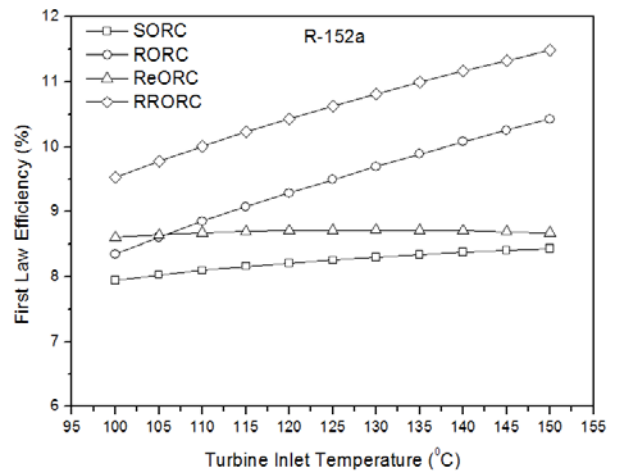


Figure 12: Efficiency V/S TIT for R-152a

6.4. Effect on first law Efficiency:

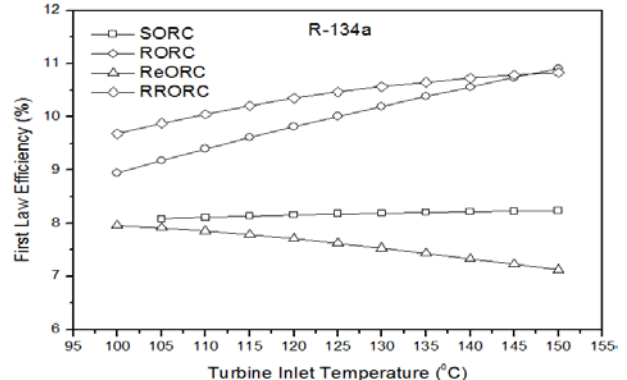


Figure 13: Efficiency V/S TIT for R-134a

The impact of change in turbine inlet temperature is shown for different working fluid. The results show that the most preferred cycle is Reheat and recuperated ORC in all working fluids i.e. dry, isentropic and wet. Figure 12, Figure 13 and Figure 14 shows the variation of first law efficiency for a wet fluid R-152a, R-134a and R-227ea resp.

It can be seen that with the increase in the TIP, the efficiency also increases for all cycle configurations except for Reheat ORC. The reheat and recuperated ORC gives the maximum efficiency for 150°C. For isentropic fluid i.e. R-134a the trend remains the same except for the reheat cycle where the reheat cycle does not increase the efficiency but decrease the efficiency. This is due to the loss of energy available in high pressure turbine. Similar pattern is repeated for the dry working fluid (R-227ea). The reheat cycle again does not increase the efficiency but reduce it. The results show that for any working fluid the best cycle is reheat and recuperated ORC configurations.

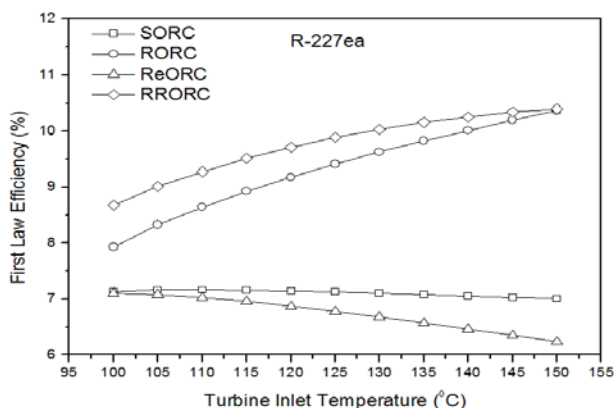


Figure 14: Efficiency V/S TIT for R-227ea

A comparative plot of Reheat is prepared for the first law efficiency with increasing TIT as shown in the Figure 15. The maximum efficiency is achieved at 150°C for R-227ea in this pressure range. The literature suggest that the mostly used working fluid id R-245fa but in the lower pressure range. In the higher pressure range the saturation temperature is also high and thus the efficiency is reduced.

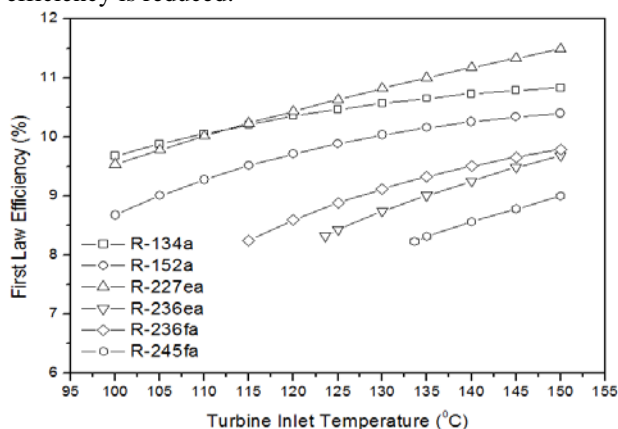


Figure 15: Comparative First Law Efficiency

7. Conclusions

A comprehensive review of 45 working fluids used in Organic Rankine Cycle in literature have been made. The most suitable working fluid in all category have been selected as R-227ea in dry, R-134a in isentropic and R-152a in wet region for selected range of 100 to 150°C. The various applications of the ORC like solar thermal power, combined heat and process, desalination, geothermal, waste heat recovery and ocean thermal energy conversion applications have been studied and the effect of working fluid have also been analyzed. The second law efficiency analysis is suggested to get a better insight of the rise in efficiency in wet working fluid than dry and isentropic fluids, yet some of the concluded points are listed below:

1. The use of working fluid R-245fa is found in most of the applications.
2. The different configuration suggested is also analyzed for finding the best configuration for low temperature applications such as solar thermal power generation.
3. The reheat and recuperated ORC configuration is found to be the best for all types of working fluids based on the first law efficiency.
4. The maximum efficiency of the RRORC is found to be 10.39%, 10.83% and 11.48% for dry, isentropic and wet working fluids.

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Nomenclature

\dot{W}_t	→ Work of turbine
\dot{W}_p	→ Work of pump
h	→ Enthalpy
\dot{m}_f	→ Mass flow rate of working fluid
\dot{m}_{cf}	→ Mass flow rate of cooling fluid
Q_c	→ Heat rejected in Condenser
Q_b	→ Heat added in the boiler
Δt	→ Temperature difference in cooling fluid
v_f	→ Specific volume of liquid working fluid
P_b	→ Boiler pressure
P_c	→ Condenser pressure

Subscripts

i	→ inlet condition
o	→ outlet condition
a	→ actual condition
t	→ turbine
c	→ condenser
b	→ boiler
p	→ pump