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Thermodynamic analysis of solar integrated waste heat recovery systems for power generation industry using eco-friendly fluids

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Abstract

This paper explores eco-friendly material R134a and activated carbon-methanol based organic Rankine model and solar integrated vapor adsorption cooling system for waste heat utilization of condenser unit of steam power plant. The results of proposed model analysis is to explain the environmental parameters of used material, effect of operating parameters and cooling-power output. The proposed analysis is concern with the condenser heat recovery of reheating-Rankine steam power plant by using eco-friendly material based ORC and vapor adsorption cooling system for simultaneously power and cooling generation. Two models have been analyzed in this paper, model-1 (Fig-1) integrated with solar ORC and model-2(fFg-2) equipped with double bed adsorption refrigeration system.Model-2 have two different adsorbed bed. Bed-1 is connected with condenser outlet of reheating-Rankine steam plant and AC-Methanol material pair is used and bed-2 integrated with solar parabolic collector for condenser water heating, that heated water circulating to bed-2 for generator purpose. The main objective of this analysis to estimate the multiple thermal effect (heating-power and cooling) from available un-covered heat of power plant by using different novel thermal system. The application of R134a and AC-Methanol type material helps to produce required cooling effect at low grade of thermal availability with environmental safety aspect.

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Keywords: Organic Rankine Cycle, Vapor Adsorption System, Waste Heat Recovery

1. Introduction

The uncovered heat from power plants and heavy industry is dumped into atmosphere and it causes of severe impact on atmosphere in terms of greenhouse effect. The US Dept. of Energy reported that the cement/ captive power plants have, nearly 35% heat is lost, and this corresponds to around 70 to 75 MW of thermal energy.

EPA estimated that the 65% of world CO_2 emission is recorded from greenhouse gas emission from fissile fuel burning during intensive industrial process. This un-used or waste heat have tremendous potential to generate heating-power and cooling generation for industrial process by using eco-friendly material based novel thermodynamic system like, organic Rankine system (ORC), kalina system for cooling-power generation, vapor absorption and adsorption refrigeration techniques. The employment of solar energy will enhance the system performance. All these systems work as waste heat recovery generator (WHRG). For a 6000 ton per day (tpd) capacity of cement plant can be reduced around 70,000 ton/annum of CO_2 by installation of WHRG.

The proposed analysis is concern with the condenser heat recovery of reheating-Rankine steam power plant by using eco-friendly material based ORC and vapor adsorption cooling system for simultaneously power and cooling generation. Two models have been analyzed in this paper, model-1 (Fig-1) and part description is given in Table-1(a) respectively.

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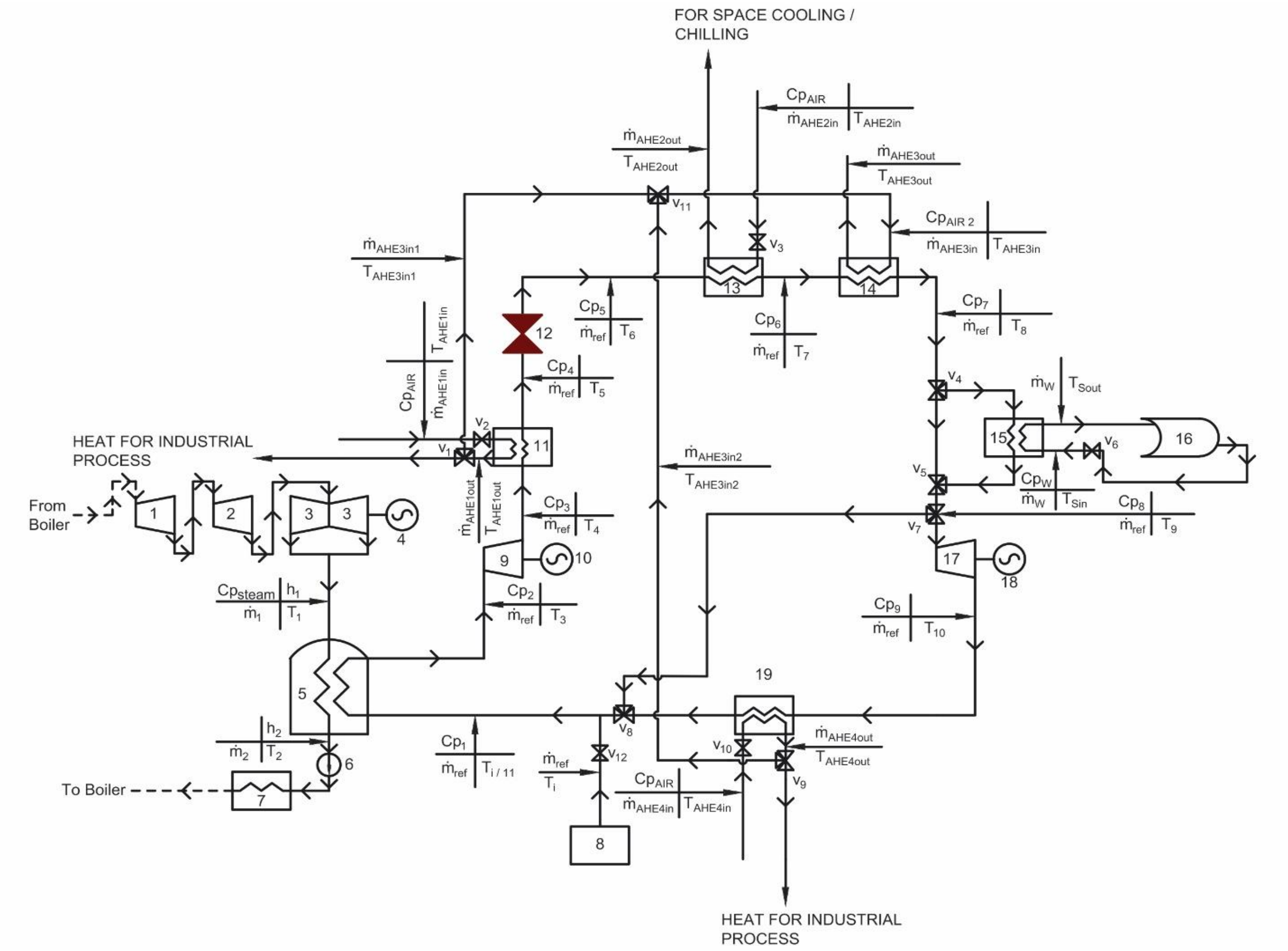

Figure 1: Schematic of R134a Based ORC heat recovery Unit

Table 1(a): Description of systems used in the schematic (Fig. - 1)

System number	System description	System number	System description	System number	System description
1	High pressure turbine	12	Throttle	V4	Three way flow control valve
2	Intermediate pressure turbine	13	Heat exchanger 2	V5	Three way flow control valve
3	Low pressure turbine	14	Heat exchanger 3	V6	Two way flow control valve
4	Power generator	15	Solar heat exchanger	V7	Three way flow control valve
5	Condenser	16	Solar heater	V8	Three way flow control valve
6	Circulation Extraction pump	17	Expander 2	V9	Three way flow control valve
7	Feedwater Heater	18	Power generator	V10	Two way flow control valve
8	Refrigerant tank	19	Heat exchanger 4	V11	Three way flow control valve
9	Expander 1	V1	Three way flow control valve	V12	Two way flow control valve
10	Power generator	V2	Two way flow control valve		
11	Heat exchanger 1	V3	Two way flow control valve		

Integrated with solar ORC and model-2(Fig-2) equipped with double bed adsorption refrigeration system. And part description is given in Table-1(b) respectively.

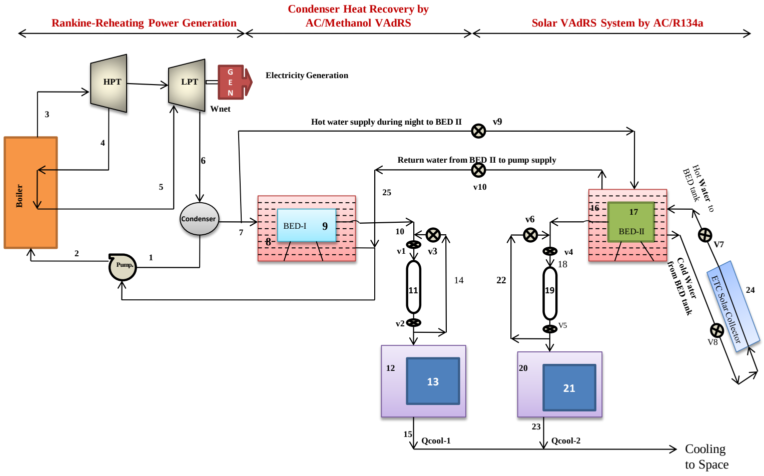

Figure 2: Solar Integrated Vapor Adsorption System for Condenser Heat Recovery

Table-1(b)-Parts Description of Model-2

S.N	Specification	S.N	Specification	S.N	Specification
1	Condensed steam feed to pump	15	Chilled air supply to space where cooling is required through cooling line.	v3	Valve 3 open for adsorption of vapor refrigerant into adsorbent bed after evaporation.(v1 & v2 remain closed when adsorption is happening)
2	Feed water supply to boiler for steam formation	16	Outer chamber of BED-2 for hot water collection, (need of hot water for adsorption of heat from adsorbate).	v4	Valve 4 open and allow to vaporized refrigerant into condenser for condensation
3	Steam from boiler supply to high pressure turbine (HPT)	17	Inner chamber of bed -2, where Activated carbon-R134a pair is filled, Its utilize condenser heat.	v5	Valve 5 allow Condensed refrigerant into evaporator and chiller unit.
4	Exhaust steam from HPT supply for reheating process	18	Vapor refrigerant (R134a) supply to condenser of VAdRS after desorption process.	v6	Valve 6 open for adsorption of vapor refrigerant into adsorbent bed after evaporation.(v1 & v2 remain closed when adsorption is happening)
5	Reheated steam supply to low pressure turbine (LPT) for more work output.	19	Refrigerant liquefy by using condenser	v7	Valve 7 open for hot water supply from ETC to BED-2 for desorption of refrigerant.(V9 & V10 are closed)
6	Exhaust steam from LPT supply to condenser unit for condensation process	20	Refrigerant liquefy by using condenser	v8	Valve 8 open for cold water return in to ETC after desorption. (V9 & V10 are closed)
7	Hot water proceed to BED-1 and only during night to BED-2 for adsorption cooling process.	21	Chiller get to be cooled during evaporation	v9	V9 works during night, whenETC does not work. It open for hot water supply from condenser exhaust to BED-2.
8	Outer chamber of BED-1 for hot water collection, (need of hot water for adsorption of heat from adsorbate).	22	Evaporated refrigerant adsorbed in adsorbent bed	v10	V 10 allow return cold water from BED-2 after adsorption and water mixed with water line for pumping into boiler.
9	Inner chamber of bed -1, where Activated carbon-Methanol pair is filled, Its utilize condenser heat.	23	Chilled air supply to space where cooling is required through cooling line.	CON DEN SER	Condensation of exhaust steam and refrigerant vapor condensation process
10	Vapor refrigerant (Methanol) supply to condenser of VAdRS after desorption process.	24	ETC solar thermal water heater for hot water supply to BED-2.	HPT & LPT	High pressure turbine and low pressure turbine. LPT use for reheated steam expansion.

11	Refrigerant liquefy by using condenser	25	Return cold water from BED-2 and mixed with water line for pumping to boiler.	GEN	Generator for electricity generation.
12	Liquid refrigerant get evaporation by evaporator	v1	Valve 1 open and allow to vaporized refrigerant into condenser for condensation	BOI LER	Steam formation thermal utilities
13	Chiller get to be cooled during evaporation	v2	Valve 2 allow Condensed refrigerant into evaporator and chiller unit.	14	Evaporated refrigerant adsorbed in adsorbent bed

Model-2 have two different adsorbed bed. Bed-1 is connected with condenser outlet of reheating-Rankine steam plant and AC-Methanol material pair is used and bed-2 integrated with solar parabolic collector for condenser water heating, that heated water circulating to bed-2 for generator purpose. The main objective of this analysis to estimate the multiple thermal effect (heating-power and cooling) from available un-covered heat of power plant by using different novel thermal system. The application of R134a and AC-Methanol type material helps to produce required cooling effect at low grade of thermal availability with environmental safety aspect.

2. Literature Review

The worldwide two major issues are being discussed in these days, CO2 concentration in atmosphere on one hand and energy crisis on other hand. All possible savings of reducing CO2 generation. Fossil fuels depletion, control in energy prices increments, achievable by adaptation of eco-friendly energy efficient material based technology. The un-arrested heat (waste heat) discharge from energy intensive industries like captive power plants, cement production, steel industry, Oil-refineries, etc have tremendous potential for multiple energy generation (heating-power & cooling) for further industrial process with dumped heat recovery [1-4]

The numerous researches have been developed and analyzed eco-friendly material based novel thermodynamic model in environmental concern. The selection of a suitable working material depends on its great influence in the design of the process, application, heat source and the temperature level. The used material must have optimum thermodynamic properties at the lowest possible temperatures and pressures, and also satisfy several criteria such as economical, nontoxic, nonflammable, environmentally-friendly. An extensive literature review of R134a was chosen as working fluid. This selection has been done on the basis that R134a: is a nontoxic and nonflammable fluid and its Ozone Depletion Potential (ODP) is zero. R134a has a high molecular mass (chemical formula: CF3CH2F, MM=102kg/kmol).It has a temperature and critical pressure of 101.1ºC and 40.6 bar, respectively, R134a based heat recovery ORC of condenser operates at a higher pressure than atmospheric, and therefore air in-leakages do not occur. [1,2-4]. Several researchers have investigated the application and performance of ORC with R134a as a working fluid, the efficiency of the ORC using benzene, ammonia, R134a, R113, R11 and R12 was analyzed [5-7]. An analysis of a regenerative ORC based on the parametric optimization, using R12, R123, R134a, and R717 as working fluids superheated at constant pressure was carried out and. results revealed that selection of a regenerative ORC during overheating using R123 as working fluid appears to be a good system for converting low-grade waste heat to power [8]. In a low-temperature solar organic Rankine cycle system was designed and built with R134a as working fluid that works between 35^{0o}C and 75.8ºC for reverse osmosis desalination in Greece, the results showed a system efficiency of about 7% and 4%, respectively [8-10]. Other studies that have analyzed the use of R134a as working fluid in the ORC cycles for reverse osmosis desalination at an experimental level [11], and also as a theoretical manner showed a simulation to estimate the increase in the efficiency and the energy available for desalination of an upper ORC coupled with a lower ORC with R134a, obtaining an efficiency for the latter of 4.2%. Other cycles with R134a for applications for geothermal sources are reported also and R134a integrated with internal combustion engine for heat recovery as bottoming cycle [12-21].The overall screening of all heat recovery fluids R134a was found to be the most suitable in terms of most eco-friendly, non-toxic, high thermal performance for cooling and power generation with optimized system.

The other category of energy recovery material for green refrigeration technology as vapor adsorption refrigeration. Activated carbon is a form of carbon that has been processed to make it extremely porous, and it has a large surface area available for adsorption. Methanol and ammonia are the most common refrigerants paired with activated carbon. Activated carbon−methanol is one of the most promising working pairs in practical systems because of its large adsorption quantity and low adsorption heat (about 1800 to 2000 kJ/kg [22]. Low adsorption heat is beneficial to the system's performance because the majority of heat consumption in the desorption phase is the adsorption heat. Another advantage of activated carbon−methanol is low desorption temperature (about 100°C), which is within a suitable temperature range for using solar energy as a heat source[26].Activated carbon−ammonia has almost the same adsorption heat as the activated carbon−methanol working pair. The main difference is the much higher operating pressure (about 1600 kPa when the condensing temperature is 40°C of activated carbon−ammonia. The high operating pressure leads to rather small pipe diameters and relatively compact heat exchangers, as compared to activated carbon− methanol. Another advantage of activated carbon−ammonia is the possibility of using heat sources at 200°C or above [23]. The drawbacks of this working pair are the toxicity and pungent smell of ammonia. Silica gel is a granular, highly porous form of silica made synthetically from sodium silicate. For the silica gel−water working pair, the adsorption heat is about 2500 kJ/kg and the desorption temperature could be as low as 50°C [22]. Such a low desorption temperature makes it suitable for solar energy use with availability of 80-100 0C. One of the drawbacks of the silica gel−water working pair is its low adsorption quantity (about 0.2 kg water/kg silica gel). Another drawback is the limitation of evaporating temperature due to the freezing point of water [22-23]. Zeolite is a type of alumina silicate crystal composed of alkali or alkali soil. The adsorption heat of zeolite−water is higher than that of silica gel−water, at about 3300 to 4200 kJ/kg. The desorption temperature of zeolite−water is higher than 200°C due to its stable performance at high temperatures. The drawbacks of zeolite−water are the same as for silica gel−water, low adsorption quantity and inability to produce evaporating temperatures below 0°C [22].

3. Thermodynamic Analysis of R134a Based Heat Recovery System

Fig-1 shows the Schematic of R134a Based ORC heat recovery Unit and description of systems used in the schematic is shown in Table-1(a), The steam from the outlet of low pressure turbine of the Rankine cycle enters the condenser at temperature T_1, having mass flow rate $\dot{m}_1$, enthalpy h_1 and temperature T_1. Condensed water leaves the condenser at temperature T_2 having mass flow rate $\dot{m}_2$ (= $\dot{m}_1$) and enthalpy h_2, the heat rejection by the steam in the condenser heat exchanger is calculated using heat balance equation.

$$Q_{COND} = \dot{m}_1 Cp_{steam} (T_1 - T_2) \qquad (1)$$

Taking the latent heat rejected by the steam in the condenser heat exchanger as Q_{LH} and the condenser heat loss 15%, the total heat available to the refrigerant vapour is calculated from

$$Q_{Total} = \{Q_{COND} + (Q_{LH} * \dot{m}_1)\} - [\{Q_{COND} + (Q_{LH} * \dot{m}_1)\} * 0.15] \qquad (2)$$

Superheated refrigerant (R 134a) vapour from the refrigerant storage is introduced into the system at a known temperature T_i for the startup cycle. In all operation cycle the refrigerant vapour enters the condenser heat exchanger at temperature T_{11}.the superheated refrigerant vapour is calculated using heat balance equation

$$Q_{Total} = \dot{m}_{ref} Cp_1 (T_1 - T_{i/11}) \qquad (3)$$

The efficiency of expander 1 (η_{E1}), T_4 and the work done by expander 1 (W_{E1}) is calculated using the equations.

$$\eta_{E1} = W_{E1} / Q_{Total} \qquad (4)$$

Where,

$$W_{E1} = \dot{m}_{ref} Cp_2 (T_3 - T_4) \qquad (5)$$

Heat available in heat exchanger 1 (Q_{HE1}) for different values of mass flow rate, $\dot{m}_{AHE1in}$, are calculated using heat balance equation.

$$Q_{HE1} = \dot{m}_{ref} Cp_3 (T_4 - T_5) - [\{\dot{m}_{ref} Cp_3 (T_4 - T_5)\} * 0.15] \qquad (6)$$

$$Q_{HE1} = \dot{m}_{AHE1in} Cp_{AIR} (T_{AHE1out} - T_{AHE1in}) \qquad (7)$$

The heat available in heat exchanger 2 (Q_{HE2}) for different values of mass flow rate of air ($\dot{m}_{AHE2in}$) are calculated using heat balance equation

$$Q_{HE2} = \dot{m}_{ref} Cp_5 (T_7 - T_6) - [\{\dot{m}_{ref} Cp_5 (T_7 - T_6)\} * 0.15] \qquad (8)$$

$$Q_{HE2} = \dot{m}_{AHE2in} Cp_{AIR} (T_{AHE2in} - T_{AHE2out}) \qquad (9)$$

Where

$$\dot{m}_{AHE3in} = \dot{m}_{AHE3in1} + \dot{m}_{AHE3in2} \qquad (10)$$

The values of T_7, $\dot{m}_{ref}$, Cp_6 and Cp_{AIR2} are known. T_{AHE3in} is calculated using the equation

$$\dot{m}_{AHE3in1} Cp_{AIR} (T_{AHE3in1} - T_{AHE3in}) = \dot{m}_{AHE3in2} Cp_{AIR} (T_{AHE3in} - T_{AHE3in2}) \qquad (11)$$

Taking the heat exchanger heat loss as 15%, T_8, $T_{AHE3out}$ and heat available in heat exchanger 3 (Q_{HE3}) for different values of mass flow rate of air ($\dot{m}_{AHE3in}$) are calculated using the equation.

$$Q_{HE2} = \dot{m}_{AHE3in} Cp_{AIR2} (T_{AHE3in} - T_{AHE3out}) - [\{\dot{m}_{AHE3in} Cp_{AIR2} (T_{AHE3in} - T_{AHE3out}) * 0.15] \qquad (12)$$

$$Q_{HE2} = \dot{m}_{ref} Cp_6 (T_8 - T_7) \qquad (13)$$

$\dot{m}_{AHE3in1}$and $\dot{m}_{AHE3in2}$ are optimized from the data generated.

3.1 Solar thermal Generator Energy Equation

$$Q_{Solar} = \dot{m}_W Cp_W (T_{Sin} - TSout) - [\{\dot{m}_W Cp_W (T_{Sin} - T_{Sout})\} * 0.15] \qquad (14)$$

$$Q_{Solar} = \dot{m}_{ref} Cp_7 (T_9 - T_8) \qquad (15)$$

Superheated refrigerant vapour leaves the condenser heat exchanger and enters expander 2 at temperature T_9, having mass flow rate $\dot{m}_{ref}$ and specific heat capacity Cp_8. It leaves expander 2 at temperature T_{10}, having mass flow rate $\dot{m}_{ref}$ and specific heat capacity Cp_9. Knowing the value of the specific heat capacity, Cp_8, of the superheated refrigerant vapour at the inlet of expander 1 and taking different values for efficiency of expander 1(η_{E1}) T_{10} and the work done by expander 2 (W_{E2}) is calculated using the equations.

$$\eta_{E2} = W_{E2} / Q_{Solar} \quad (16)$$

$$\text{Where, } W_{E2} = \dot{m}_{ref}\, Cp_8\, (T_9 - T_{10}) \quad (17)$$

Superheated refrigerant vapour leaves expander 2 and enters heat exchanger 4 as the hot fluid at temperature T_{10} and having mass flow rate $\dot{m}_{ref}$ and specific heat capacity Cp_9. It leaves heat exchanger 4 at temperature T_{11}, having mass flow rate $\dot{m}_{ref}$ and specific heat capacity Cp_1.The value of Cp_1 will change in the transition from startup cycle to operation cycle. Air enters heat exchanger 2 as the cold fluid at temperature T_{AHE4in}, having mass flow rate $\dot{m}_{AHE4in}$ and specific heat capacity Cp_{air}. It leaves heat exchanger 4 at temperature $T_{AHE4out}$ and having mass flow rate $\dot{m}_{AHE4out}$ (= $\dot{m}_{AHE4in}$). The values of T_{10}, $\dot{m}_{ref}$, Cp_9 and Cp_{air} are known. T_{AHE4in} is taken as 288.15 K as per standard atmosphere data. Taking the heat exchanger heat loss as 15%, assuming that no makeup is required, T_{11}, $T_{AHE4out}$ and heat available in heat exchanger 4 (Q_{HE4}) for different values of mass flow rate of air ($\dot{m}_{AHE4in}$) are calculated using heat balance equation.

$$Q_{HE4} = \dot{m}_{ref}\, Cp_9\, (T_{11} - T_{10}) - [\{\dot{m}_{ref}\, Cp_9\, (T_{11} - T_{10})\} * 0.15] \quad (18)$$

$$Q_{HE4} = \dot{m}_{AHE4in}\, Cp_{air}\, (T_{AHE4out} - T_{AHE4in}) \quad (19)$$

3.2 Model-2-Activated Carbon based Solar Integrated Vapor Adsorption Cooling System for Condenser Waste Heat Recovery.

The proposed model which is shown in fig-2 of double bed vapor adsorption system is adopted for heat recovery of condenser of reheating-Rankine steam power cycle. In this system two eco-friendly materials-pair have been used Activated carbon (AC) methanol and activated carbon-R134a. 1st bed is equipped with AC-Methanol and 2nd bed is running with AC-R134a.The solar parabolic trough collector is integrated with 2nd bed for more heating at generator of 2nd bed. In the beginning of the desorption process in both bed, the desorbed is heated by the waste heat of Reheating Rankine condenser and solar collector to begin desorbing, and the condensation process occurs firstly in the evaporator because the evaporator temperature is lower than the condenser temperature at that time. Such process will continue until the evaporator temperature rises higher than the condenser temperature. Thereafter, the condenser will be in condensing mode, and the evaporator will be idle.Table-4 describe all parts of model-2.

4. Thermodynamics model of Double bed VADRS

The total energy input to the system is given by, The total heat input to the system is estimation of energy liberated during adsorption is called isostatic heating and heat due to desorption The equations for thermal analysis of model2 is adopted from Mahesh and Kausik model[32].

$$Q_{heat\_total} = Q_{isosteric} + Q_{desorption} \quad (20)$$

Mahesh and Kaushik [32] is written as

$$Q_{heat\_total} = \sum m_{ad}\,[(Cp_{ad} + Cp_r * X_{max})(T_g - T_{ad}) + (Cp_{ad} + \Delta X/2)(T_{des} - T_g) + \Delta X * H_D]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (21)$$

After condensation of desorbed refrigerant, the liquid refrigerant in the evaporator will absorb the heat of vaporization from liquid to be cooled, this cause the refrigeration effect and it can be estimate as

$$Q_{Ref} = \sum [(m_{ad} * \Delta X * L_E)]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (22)$$

The amount of energy used in cooling the evaporated adsorbate when passing through the condenser is given by

$$Q_{Cool} = \sum [(m_{ad} * \Delta X * Cp_r\, (T_C - T_E)]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (23)$$

The useful cooling of vapor adsorption system can be estimated by

$$Q_{cool\ useful} = \sum m_{ad}\,[(L_E * \Delta X - Cp_r\, (T_C - T_E)]_{BED\text{-}1\ \&\ BED\text{-}2} \quad (24)$$

The compactness or size of system is important parameter, Specific cooling power (Useful cooling per unit mass) of adsorption system decide the compactness. Desorption and Adsorption time (second) play key role here for cooling effect of VAdRS and its SCP.

$$SCP_{VADRS} = Q_{cooluseful} / m_{ad} \quad (25)$$

The ratio of useful cooling and total heat input for adsorption and desorption account the refrigeration capacity of cooling machine.

$$COP_{VADRS} = Q_{cooluseful} / Q_{heat\_total} \quad (26)$$

The refrigeration effect of solar vapor adsorption refrigeration system account by COP_{SOLAR}, The ratio of net cooling heat and heat gain by solar thermal system is express as

$$COP_{SOLAR} = Q_{cooluseful} / Q_G = Q_{cool\ useful} / (I * Ac) \quad (27)$$

5. Results and Discussions

The numerical computation were carried out for Energy Recovery using eco-friendly materials for ORC and Adsorption cycle and the energy recovery heat grades, and efficient materials have been shown in table 2-4 respectively. Similarly all parametric results are shown in Table 5 to 8 respectively.

Table-2 Energy recovery and its sources [26]

Heat Categories	Heat Sources	Temperature in 0C	Suggested recovery technology
High Grade (>650^{0}C)	Solid waste	650-1000	Air preheating
	Nickel refining furnace	1370-1650	Steam generation for heating
	Copper reverberatory furnace	760-815	Thermoelectric and thermal PV
	Glass melting furnace	1000-1550	Heat exchanger for preheating
	Hydrogen plant	650-1000	Thermal PV
Medium Grade (230-650^{0}C)	Steam boiler exhaust	230-480	Steam rankine cycle
	Gas turbine exhaust	370-540	Organic rankine cycle
	Drying and baking ovens	230-600	Thermal PV
	Catalytic crackers	425-650	Thermal PV
	Reciprocating engine exhaust	315-600	Thermoelectric
Low Grade (>230^{0}C)	Welding and injection molding	32-88	Kalina cycle
	Hot processed liquids and solids	32-233	Organic rankine cycle
	Drying,Baking and Curing ovens	93-230	Absorption and adsorption cooling
	Bearing	32-88	Piezoelectric

Table-3: Summary of Commercialized Working fluids [27-29]

S. N	Working Fluids	Physical Data				Safety Data	Environmental Data		
		Molecular Mass (Kg/K mol)	Normal Boiling Point Temp (Tbp) in ^{0}C	Critical Temp (Tcriti) in 0C	Critical Pressure (Pcriti) in MPa	ASHRAE Group	Atmospheric Life Time (ALT) in years	Ozone Depletion potential (ODP)	Global Warming Potential (GWP) of 100 years
1	RC118	200.03	-6.0	115.2	2.778	A1	3200	0	10225
2	R600a	58.12	-11.7	135	3.647	A3	0.017	0	˜˜20
3	R114	170.92	3.6	145.7	3.289	A1	300	1.00	10040
4	R600	58.12	-0.5	152	3.796	A3	0.018	0	˜˜20
5	R601	72.15	36.1	196.5	3.364	NA	0.01	0	˜˜20
6	R113	187.38	47.6	214.1	3.439	A1	85	1.000	6130
7	Cyclohexane`	84.16	80.7	280.5	4.075	A3	NA	NA	NA
8	R290	44.10	-42.1	96.68	4.247	A3	0.041	0	~20
9	R407C	86.20	-43.6	86.79	4.597	A1	NA	0	1800
10	R32	52.02	-51.7	78.11	5.784	A2	4.9	0	675
11	R500	99.30	-33.0	105.5	4.455	A1	NA	0.738	8100
12	R152a	66.05	-24.0	113.3	4.520	A2	1.40	0	124
13	R717 Amonia)	17.03	-33.3	132.3	11.333	B2	0.1	0	<1
14	Ethanol	46.07	78.4	240.8	6.148	NA	NA	NA	NA
15	Methanol	32.04	64.4	240.2	8.104	NA	NA	NA	NA
16	R718 (Water)	10.2	100	374	22.064	A1	NA	0	<1
17	R134a	102.03	-26.1	101	4.059	A1	14.0	0	1430
18	R12	120.91	-29.8	112	4.114	A1	100	1.000	10890
19	R123	152.93	27.8	183.7	3.668	B1	1.3	0.02	77
20	R141b	116.95	32.0	204.2	4.249	NA	9.3	0.120	725
21	R245fa	134.05	15.3	154.1	3.64	B1	8.8	0	820
22	R236fa	152.0	-1.5	124.0	3.20	-	209	0	6300
23	R227ea	170.0	-17.5	102.0	2.95	-	36.5	0	2900
24	R1234yf	114.02	-29.45	94.7	3.382	A1	NA	0	4

Table-4-Energy Recovery Materials for Adsorption Refrigeration Techniques of Heat Recovery [24]

Material	Heat Source	Performance Index(COP)	Application
AC-Methanol	20 MJ/m^2 per Day	0.12	Ice Making
	18 to 19.2 MJ/m^2 per Day	0.12 to 0.14	
	17 to 20 MJ/m^2 per Day	0.13 to 0.15	
	<120 degree C	0.18	
AC-NH3	105 degree C	0.10	
Silica Gel-Water	15.4 MJ/m^2 per Day	0.16	
AC-Blanked steel-Methanol	20 MJ/m^2 per Day	0.18	
AC+CaCl2-NH3	115 degree C	0.39	
Silica Gel-Water	55 degree C	0.36	Chilled Water
	65 degree C	0.28	
	75-90 MJ/m^2 per Day	0.35 to 0.60	
	80 to95 MJ/m^2 per Day	0.35 to 0.60	
	80 MJ/m^2 per Day	0.33to 0.50	
AC-NH3	232 degree C	0.42 to 1.19	Air Conditioning
	100^0 C	0.20	
Zeolite-Water	204^0 C	0.60 to 1.60	
	230^0 C	0.41	
	310^0C	0.38	
	230 to 300^0C	0.20 o 0.21	

Table 5: Parametric results of orc heat recovery system

Parametric Results	Cycle without solar	Cycle with solar
$\eta_{thermal}$	8-15%	12-27%
$\eta_{expander}$	10-25%	10-25%
Process Heat ($Q_{process\ heat}$)	900-1100 kW	1200-1500 kW
Cooling Effect ($Q_{Cooling}$)	14- 19.25 kW	35- 38.5 kW
Expander work ($W_{expander}$)	200-470 kW	260-650 kW
Solar Heating (Q_{Solar})	N/A	460 kW
Solar Collector Efficiency	N/A	20-25%
Water Temp through solar field	N/A	343.15-363.15 K

Table 6: Performance of Adsorption bed-1

COP_{TH}	Q_{COOL}(kJ)	Q_{HEAT}(kJ)	Q_{REF}(kJ)	SCP(kW/kg)	T_{GEN} (K)
0.488	854.6	33543	17222	23.32	328
0.4852	1688	58643	30139	40.53	333
0.4803	2687	84050	43056	57.51	338
0.4749	3850	109765	55973	74.25	343
0.4692	5179	135787	68890	90.76	348

Table 7: Performance of Adsorption bed-2

COP_{SOLAR}	COP_{TH}	I_G (W/m^2)	Q_{COOL}(kJ)	Q_{HEAT}(kJ)	Q_{REF}(kJ)	SCP(kW/kg)	T_{GEN}(K)
0.4811	0.507	750	1570	19858	11638	38.18	333
0.5529	0.482	775	1865	24804	13820	45.34	340.5
0.6202	0.4593	800	2159	30136	16002	52.49	348
0.6834	0.4387	825	2454	35855	18184	59.65	355.5
0.7429	0.4199	850	2748	41961	20366	66.81	363

Table 8: Parametric Results of combined double bed vapor adsorption System

Performance Parameter	Resultant Values	Performance Parameter	Resultant Values	Performance Parameter	Resultant Values
Turbine work output (W_{HPT}& W_{LPT})	294.7 & 1072 kJ/kg	Condenser heat exhaust (Q_{OUT})	2133 kJ/kg	COP of combined VAdRS ($COP_{COMBINEDVADRS}$)	0.46 to 0.51 (13-15% increment in overall COP)
Pump Work (W_{PUMP})	16.23 kJ/kg	Plant Efficiency (η_P)	0.388	Sp. Cooling Power ($SCP_{COMBINEDVADRS}$)	70-90 W/kg (10-16% increment in overall SCP)
Boiler Heat Generation (Q_{IN})	3484 kJ/kg	Net Work Output (W_{NET})	1350 kJ/kg	COP gain by Solar energy (COP_{SOLAR})	0.48-0.74 (most effective COP)

Fig -3 depicts the efficiency of scroll type expander is in the range of 14% to 25% of model-1. The expander in case of cycle without solar integration will do more work. In case of solar integration, more heat for industrial process will be obtained. By increasing the mass flow rate of air in HE-2 increases the heat obtained thereby reducing the cooling effect and heating the air more. In operation cycle with solar heater the change in heat obtained and the rise in air temperature for a small increase in mass flow rate of air is much more than that in the operation cycle without solar heater. Fig 4 shows the integrating solar heating with operation cycle increases the process heat obtained for industrial purpose. When solar heating is not implemented, the heat for process obtained from HE 1 will be more in both cycles than that obtained when solar heating is implemented. Moreover, the gain in the heat obtained from HE 1, for process, per unit increase of mass flow rate of air passing through the HE-1, in case of operation cycle, is greater when solar heating is not implemented than when solar heating is implemented. Fig 5 shows the expander work obtained and the thermal efficiency will be more in case of startup and operation cycle without solar heating. This is because in case of startup and operation cycle without solar heating, work output is being provided by only one expander and also heat obtained for industrial process is only from one heat exchanger as opposed to the case of startup and operation cycle with solar heating where the work output is being provided by two expanders and the heat obtained for industrial process is only from two heat exchanges with additional heat input to the cycle through the utilization of solar energy. The variation of COP with variation of source temperature of VADRS is shown in Fig-6 and it is found that COP o of both Beds are decreasing with increasing source temperature of VADRS. However the COP of bed A is higher than COP of Bed B. Similarly the variation of combined COP with variation of source temperature of VADRS is shown in Fig-7 respectively. It is found that combined COP is increasing order as source temperature of VADRS is increased.

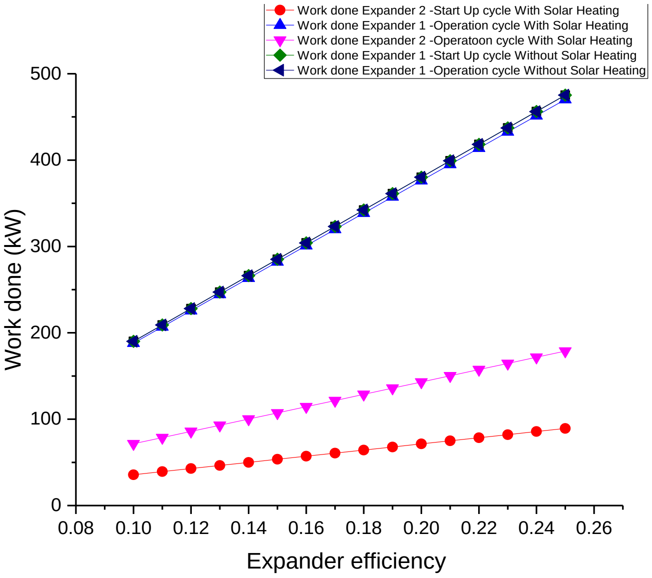

Figure 3: Effect of expander efficiency on work done by expander

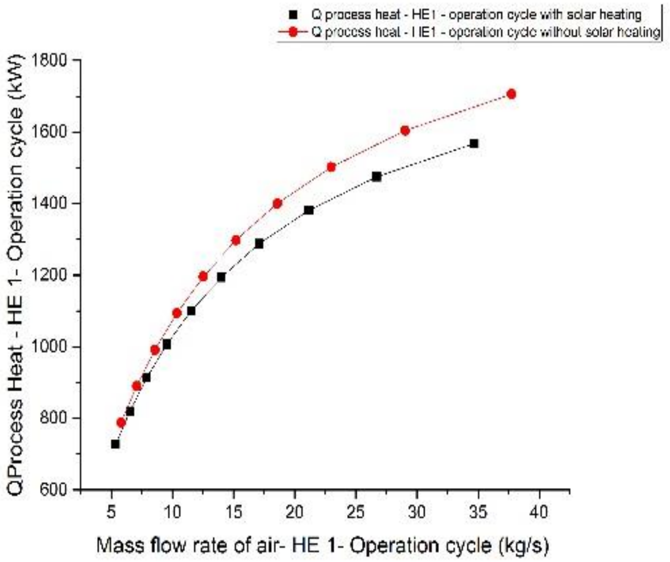

Figure 4: Effect of mass flow rate of air, flowing through HE 1, on Q process heat in operation cycle and startup cycle

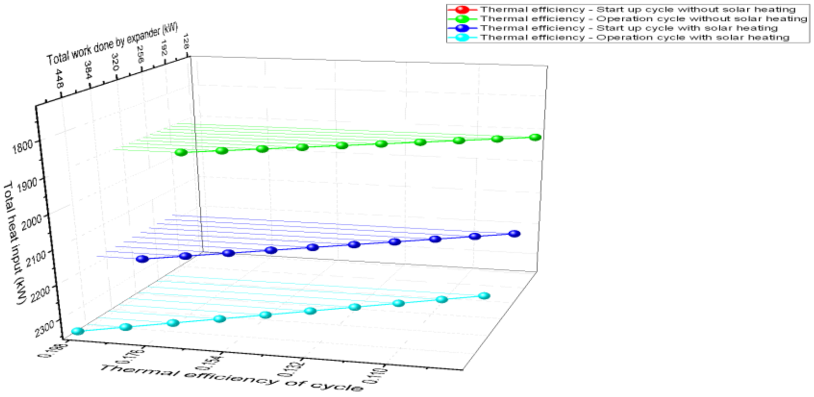

Figure 5: Total work done by expander v/s total heat input V/s thermal efficiency

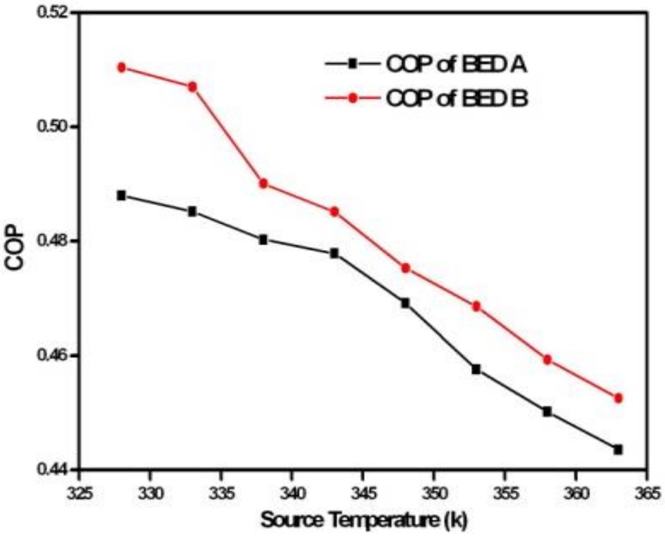

Figure 6: Variation of COP with variation of Source Temp of VADRS

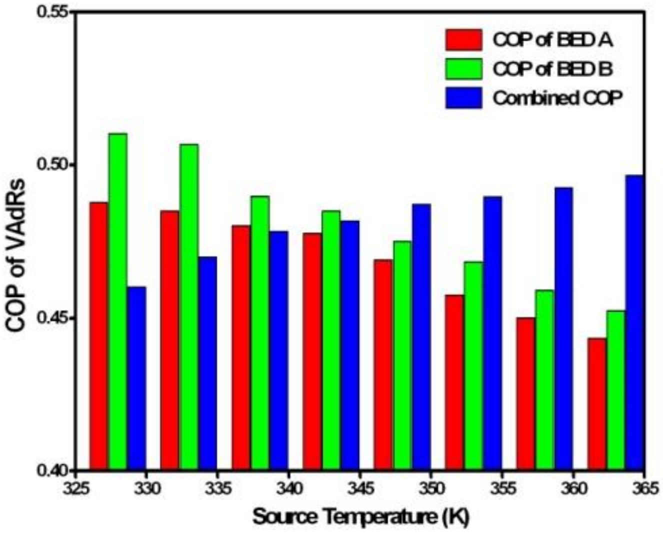

Figure 7: Variation of Combined COP Vs variation of Source Temp of VADRS

6. Conclusion

The selection of a suitable working material depends on its process, application, heat source and the temperature level. The used material must have optimum thermodynamic properties at the lowest possible temperatures and pressures, and also satisfy the several criteria such as being economical, nontoxic, nonflammable, environmentally-friendly.

All conventional refrigeration systems consume precious fuel or electricity to attain refrigeration. ORC and Adsorption systems are heat-operated units that need only heat in terms of un-covered heat from the energy intensive sectors, so they can utilize waste heat and generate multiple effects like heating, power and cooling.

The major conclusion are drawn from both model analysis are given below

1. Rankine Cycle with solar coupling provides more heat for process. This is nearly two times that of the heat provided for process by the cycle without solar coupling, but the total work achieved in case of solar heating is less than that achieved without solar heating.
2. More power from the expanders is achieved when solar heating is not implemented. This is due to the the heat generated by the solar heater is converted into process heat. Therefore, solar integration is recommended, for the purpose of obtaining more process heat, in summer and for the purpose of power generation in winter.
3. Solar integrated thermal circuit in combined power generation system enhance the cooling effect due to more heat addition through solar heating in R134a ORC system.R134a will attain superheating, and produce cooling effect.
4. The parametric table-5 concludes that the performance indicators (thermal efficiency, heating and cooling effect) of proposed system is increased by 30-40% using solar system.
5. Parametric results which is shown in table-8 of double bed vapor adsorption refrigeration system for heat recovery of Reheating-Rankine condenser, the overall COP and SCP enhanced by 10-16% shown in figure 7.
6. Solar connected vapor adsorption bed gives more COP individually as compare to condenser connected vapor adsorption bed. The specific cooling power (SCP) of solar vapor adsorption bed is increasing but increasing of evaporation temperature will decrease the SCP of solar vapor adsorption bed due to conversion of solar gain into heating (Qheat-solar) by double rate.
7. The present combination of adsorbent-adsorbate material is suitable for low grade heat recovery with solar thermal integration for continuous cooling effect generation and applicable for space cooling purpose. The used material R134a and methanol with activated carbon is most suitable pair under the category of eco-friendly material as per AHRAE guidelines of environmental safety protocol.
8. The different materials have reviewed in present analysis. The two parameters are very important for environmental aspect, like GWP and ODP values. Minimum value of GWP &OPD shows non-toxic and eco-friendly nature of material for power and cooling generation system.
9. The proposed material also applicable for low grade heat recovery purpose due to its low boiling and freezing temp thermal property.

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