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Research Article

Thermodynamic and Exergy Analysis of an Isopentane-Based Organic Rankine Cycle for Waste Heat Recovery

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Abstract

The growing need for renewable and sustainable energy technologies for efficient power production has led researchers to investigate various techniques that employ the use of low-temperature sources of energy to produce power. One of the efficient technologies that has proven to be very useful in generating power from low and medium temperature heat sources is the Organic Rankine Cycle. This paper explores the thermodynamics, exergy analysis, and economics of an ORC using Isopentane fluid. Analysis of the ORC performance is based on energy and exergy balance equations taking into consideration changes in condenser pressure, evaporator pressure, and superheating temperatures. Thermophysical properties and thermodynamics of the Isopentane cycle were determined using REFPROP software based on the environmental friendliness, safety, and thermodynamics considerations of Isopentane. Parameters that were considered include pump work, turbine work, heat added, heat rejected, thermal efficiency, exergy destruction, and second-law efficiency. Three optimization cases were investigated to improve cycle performance: decreasing condenser pressure to 0.15 MPa, increasing the superheating temperature to 455 K, and increasing evaporator pressure to 3.2 MPa. The results suggest that the thermal efficiency and second-law efficiency of the cycle are substantially enhanced by reducing the condenser pressure. At a condenser pressure of 0.15 MPa, 30 percent second-law efficiency and a maximal thermal efficiency of 16% were attained. Superheating increased the total exergy of the system, while higher evaporator pressure provided moderate improvements in efficiency. The study also demonstrates that Isopentane is a suitable working fluid because of its high specific heat capacity, favorable critical temperature, non-toxic nature, and environmentally friendly characteristics. In addition, a solar thermal collector and thermal storage system were integrated with the ORC system to enhance energy utilization and overall performance. Economic analysis showed that the system can generate approximately 460,080 kW annually with an estimated payback period of about five years. The results of this work show the capability of Isopentane-based ORC systems for efficient recovery of waste heat and sustainable power generation from renewable and low-temperature energy sources.

I. INTRODUCTION

The use of new energy conversion devices is essential to guarantee the production of electrical energy without pollution of the environment. Among those technologies, low-grade heat sources or low-power heat sources are important fields of development. ORC is a well-known technology that can be traced back to the 1980s. In this case, the ORC includes the same elements as those found in a conventional steam power plant (a boiler or evaporator, a work-conversion expansion machine, a condenser, and a pump). In contrast to a conventional steam power plant, the ORC uses an organic working substance, whose boiling temperature is lower than that of water, thus making power generation possible using low temperatures of heat sources. ORC systems have already been commercialized at the MW power level. ORCs convert low to medium temperature heat sources from 80 to 350°C into power, and small-to-medium applications across a range of temperatures.

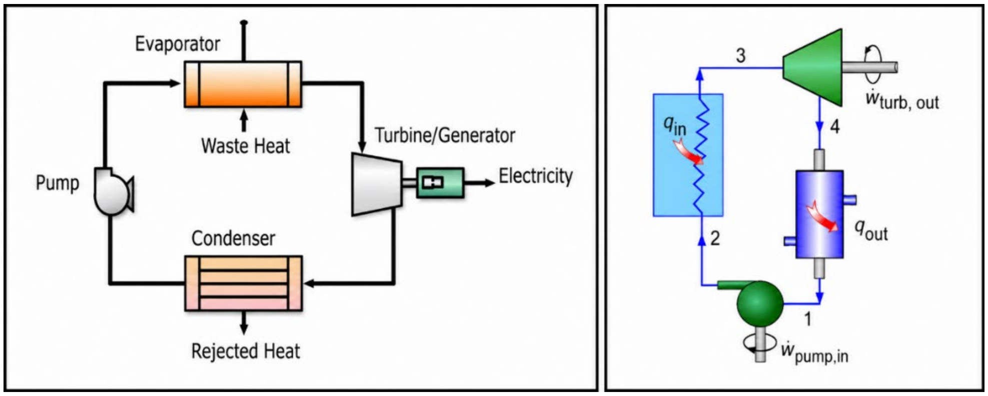

Fig.1 Schematic Diagram of Organic Rankine Cycle

ORC power plant operation works similarly to the process of electricity production used in power plants. The principle of electricity production that is employed in the organic Rankine cycle plant consists in heating and evaporating water into steam. The difference from other types of power plant is that the organic working fluid boils at a lower temperature and its vapor pressure is higher than that of water; hence, low temperature heat can be used for generating electricity. According to different properties, different organic fluids are selected. This allows using the specific heat source more effectively and obtaining higher efficiency. The working cycle starts at the pump, where refrigerant, or the fluid of the internal circuit, is pumped into the evaporator. There, with the help of available heat sources, the refrigerant is evaporated. Then the vapor produced by the evaporator is transferred to the turbine. The expansion of the vapor provides power for rotating the generator and producing electricity. Supersaturated vapor is then sent to the condenser. After condensation, it goes back to the pump.

In the Organic Rankine cycle, the working fluid passes through the four components as shown in Figure 2.

1-2: Isentropic compression in a pump
2-3: Constant pressure heat addition in a boiler or evaporator
3-4: Isentropic expansion in a turbine
4-1: Constant pressure heat rejection in a condenser

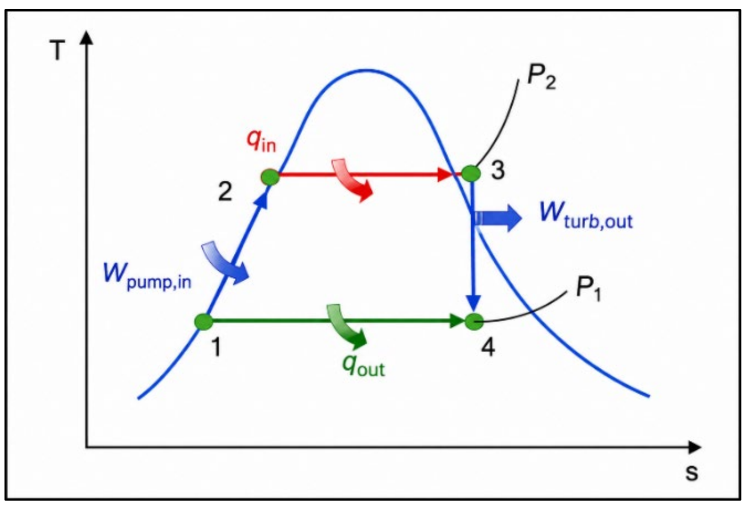

Fig.2 Temperature Entropy Diagram of Organic Rankine Cycle

On average, the compressibility of a liquid requires less energy compared to the compressibility of a gas. Consequently, the work put into the pump in the Rankine cycle is far less than that obtained from the turbine, resulting in a net power output. Efficiency is measured by comparing the net power output to the amount of heat energy supplied to the boiler/evaporator. In the case of the basic Rankine cycle illustrated in Figure 1 and 2, the efficiency is roughly 30 %.

The primary aim of this study is to analyze the optimal working fluids for the Organic Rankine Cycle (ORC) system from a thermodynamic perspective in order to improve system performance and energy utilization. The study also focuses on assessing the importance of environmental and safety criteria in the selection of suitable working fluids and system configurations. Another objective is the proper selection of the ORC system and its working fluid based on operational requirements and efficiency considerations. Furthermore, the research includes mass and energy conservation calculations to evaluate system performance accurately. The optimization of the energy system is also an important objective of this work, with the ultimate goal of enhancing the overall efficiency of the system. The ORC can be efficiently employed in various applications for the generation of mechanical work and electrical power. One of its major applications is waste heat recovery, where low-grade waste heat from industrial processes is converted into useful energy. ORC systems are also widely used in biomass power plants to generate electricity from biomass-based heat sources. In addition, ORC technology is suitable for geothermal power plants due to its capability to operate effectively with moderate-temperature geothermal resources. Another important application is in solar thermal power systems, where solar energy is converted into thermal energy and then utilized to produce electricity through the ORC process.

II. LITERATURE REVIEW

The comprehensive review of literature is presented about the various aspects of organic Rankine cycle. Many techniques are in use nowadays for the qualitative as well as quantitative analysis of energy generation. However, thermodynamic properties of working fluid and system selection have been studied. Roberto Agromayor, *et al.* examine that the choice of the working fluid may depend on various factors such as compatibility of the working fluid with the heat source and heat sink, stability, environmental concerns, safety considerations, or economic considerations. In order to fulfill this purpose, a multi-objective, thermodynamic optimization technique was applied by means of a genetic algorithm. In the process, the second law efficiency was taken as the objective function and the performance of the expander was assessed at the optimum operating condition. When a dry or isentropic working fluid is used in a cycle arrangement, the optimum is found to be when the temperature of the heat source is slightly higher than the critical temperature of the working fluid [1]. A techno-economic model has been proposed by James Freeman *et al.* to evaluate the possible performance of a residential CSHP-ORC system with a positive displacement expander in the UK. A better comparative analysis needs to be conducted between several designs of a PV and solar water heating system occupying the same roof space or between a PV-T system where the PV and solar-thermal elements are integrated into one unit. Moreover, it is necessary to select an appropriate working fluid, especially for the concentrated collector system, whose operating temperature was found to be restricted by using the working fluid R245fa [2].

Omar Aboelwafa, *et al.* have found that the Rankine cycle is the competing cycle for generating power from solar energy. CSP collectors are often employed for large scale systems due to their high efficiencies when operating at moderate and high temperatures. Non-concentrating collectors can be used in low power output systems. Normally, refrigerants such as R245fa, R134a, CO2, n-butane, i-butane and propane were extensively tested under temperatures lower than 250 °C, while toluene and water were tested at higher temperatures (toluene up to 340 °C) [3]. The ORC system with dry working fluid has a higher net power than the ORC system with moist working fluid, as demonstrated by Fan Wei *et al.* The more cost-effective option is the ORC system with moist working fluid. The IORC's thermal efficacy is not necessarily superior to that of the ORC with the same working fluid. If the equivalent exhaust temperature is exceeded by the exhaust gas temperature, the thermal efficacy of the IORC is enhanced. With the moist working fluid, the ORC system has a lesser investment cost and a lower net power output. When the temperature of exhaust gas is not restricted, then the most economic working fluid for the ORC system is R152a, while R245fa is the most economic when the temperature of exhaust gas does not exceed 90 °C [4].

Collings, P., *et al.* studied small-scale ORC cycles which are typically designed using positive-displacement expansion devices whose efficiencies are dependent on a fixed expansion ratio. The effect of an increased pressure ratio due to the adjustment of the condensation temperature to reflect cold environmental temperature during the winter season cannot be harnessed by positive displacement expanders. With the recuperator system however, it is possible for the higher temperature discharge from the expander to lead to enhanced heat recovery from the cycle resulting to lower heat input into the evaporator and improved thermal efficiency and specific power output. A study was conducted involving the exergy of the process, and it was revealed that exergy destruction occurred primarily in the evaporator [5].

According to the research carried out by Spayde, *et al.*, the authors performed an analysis to reveal the benefits that are associated with the use of solar-based ORC with electric energy storage (EES) technology to supply electrical power to commercial buildings like a large office, a small office, and a full-service restaurant. In the operation process of the solar-based ORC-EES technology, the ORC provides the electrical power generated by the ORC to the EES when the irradiation level is high, and the EES provides the electrical power to the building when the irradiation level is low. Furthermore, the authors analyse the effect of the quantity of solar panels on the capacity of the batteries, financial gains, percentage of electricity generation, and battery discharge period [6]. The performance of the organic Rankine cycle–vapor compression refrigeration integrated system is analysed using the energy and exergy approach. The proposed system working fluids are R600, R600a, R601, R601a and R1234ze(E). It can be concluded that the working fluid with the highest critical temperature has reached the highest energetic efficiency, the highest exergetic efficiency and the lowest total mass flow rate. The R602 is the most appropriate choice for integrated system for the use of renewable energy in between temperature of 70 °C and 110 °C. The total exergy destruction rate for the typical cycle is 84.1 kW. The place of maximum exergy destruction is found to be the condenser with about 34.7% of total rate [7].

Kamran Taheri *et al.* discuss the issue of increasing energy efficiency in manufacturing systems. In recent years, exergy has appeared as a useful technique from the field of thermodynamics in the evaluation of energy efficiency in addition to energy balance. The combination of exergy efficiency and exergy destruction based on the exergy approach can show the inefficiency of energy in the system. The system's proposed indicators for evaluation are energy efficiency, exergy efficiency, and exergy destruction.

Advantages of exergy approach in comparison to the conventional approach of energy efficiency evaluation through verification of thermal spray processes [8].

The authors Daniele Cocco, *et al.* make use of concentrating solar systems for providing thermal energy and electric power for industrial processes. Exergy analysis is done, and the plant exergy efficiency is selected as the index to determine the optimal configuration. A parallel arrangement of the heat generator with the ORC system becomes a sole alternative for producing high-pressure steam, while a configuration where the heat generator is placed after the ORC becomes feasible when low-pressure steam is considered, along with high power-to-heat ratios. Waste heat recovery from the ORC system can be an excellent option for hot water generation; however, it demands complete utilization of available heat to prevent substantial exergy losses [9]. The design and optimization of the ORC used as a bottoming cycle in the combined cycles Brayton/organic rankine cycle and steam rankine/organic rankine cycle are regarded as the main focus of this research. The exhaust temperature of the topping cycle is the criteria for determining the working fluids that can give the best performance of the bottoming cycle (ORC). In this regard, Iso-butane, R11 and ethanol are found to be the best working fluids which give the maximum efficiency of the ORC bottoming cycle due to its high efficiency in producing thermal energy. Working fluids with high specific heat capacity or high critical temperatures provide more thermal efficiency in the supercritical region [10]. Chao He, *et al.* aimed to find the OET (optimum evaporating temperature) of subcritical ORC from thermodynamic principle by considering net power as the objective function. To evaluate the validity of the findings, the quadratic approximation approach with EES (engineering equation solver) software is applied and the optimum evaporating temperatures are determined through simulations based on the maximization of net power. Based on the optimal value of net power, appropriate working pressures, overall heat transfer rate and size parameters of the expander are selected and R114, R245fa, R123, R601a, n-pentane, R141b and R113 are recommended as working fluids for the subcritical organic Rankine cycle [11].

Bao Junjiang, *et al.* suggests an appropriate selection of working fluid as well as a selection of expansion machines for ORC systems. The mixed working fluids have good temperature matching to enhance the efficiency, which is limited by the working fluid selection process to the operating conditions, working fluid properties, structures, and environmental safety, and all those limits are beneficial for the preliminary working fluid selection and cleaning; the selection of expansion machines requires taking into account many aspects, such as power output capability, isentropic efficiency, cost, and complexity, among other parameters; therefore, different expansion machines require different application scope [12]. Off-design situations in the study done by Changwei Liu, *et al.*, for ORC geothermal power plants can be experienced from changes in geology, environmental temperatures, and operational parameters. This results in an increased flow rate, which subsequently increases the net power generation. In case the inlet geothermal water temperature is increased, the refrigerant pump speed will become the control variable that increases, hence increasing evaporation pressure and mass flow rate; therefore, net power generation increases. However, if the cooling water inlet temperature is reduced, a combination of refrigerant pump speed and turbine guide vane angle will reduce the condensation pressure, which increases net power output [13].

Gao, P. *et al.* conducted an investigation into an ORC system that used a variable-displacement scroll expander. So, energy and exergy efficiency analysis are taken into account. The ORC system is subsequently constructed and investigated in accordance with the model. The experimental results indicate that the system's energy efficiency ranged from 1.7% to 3.2%, and its exergy efficiency fluctuated between 8.6% and 16.9% at a heat source temperature of 105°C. Using submodels of the main apparatus and experiments on the isentropic efficacy of the expander, models of thermodynamics and heat transfer were developed for organic Rankine cycles (ORC) systems. The ORC system was developed, and the energy and exergy efficiencies were investigated through experiments [14].

According to Fredy Velez, *et al.*, the subcritical Rankine cycle with organic working fluids was employed to generate electric power through the application of low-temperature heat sources, as demonstrated by the thermodynamics investigation. The research conducted has demonstrated that these systems can be effectively employed for the recovery of heat in industries and/or the exploitation of renewable energy sources at low and moderate temperatures to generate electricity. It appears that ORC technology has significant potential for the conversion of low-temperature heat sources into electrical energy by utilising renewable energy resources, such as biomass, solar, and geothermal energy, as well as residual heat from industrial processes or other processes. Nevertheless, the environmental impact, safety, stability, thermophysical properties, availability and cost of the working fluid are important factors in selecting appropriate working fluids for such processes [15]. Yuping Wang, *et al.* have done experimental analysis of the operational efficiency of an ORC system with zeotropic mixture using a small-scale experimental setup of ORC power generation system. R601a/R600a has been chosen as the working fluid for this purpose. For the heat source temperature of 115°C, the maximum power generation efficiency is obtained at the mixture ratio of 0.6/0.4. At a temperature of 115°C, the optimal mixture ratio is 0.6/0.4, which results in the highest net power. Additionally, the net powers of the majority of R601a/R600a proportions are greater than those of the unadulterated fluids. This suggests that the efficacy of the ORC system can be enhanced by the addition of R601a/R600a [16]. It is evident that Isopentane has the highest net power output value among organic fluids. This is due to the fact that its critical temperature (187.2 °C) is the closest to that of the waste heat source (160 °C). Therefore, as the critical temperature of the fluid approaches that of the waste heat, the net power increases. Isopentane possesses the greatest heat transfer capacity. Taking into account the net power and the cycle efficiency, it is evident that Isopentane has advantages with respect to other working fluids. In accordance with the model and environmental aspects, the following conclusions can be made. In accordance with current working conditions, the most suitable combination of an ORC configuration with a working fluid can be recommended as the ORC configuration with Isopentane because it is characterized by such advantages as high specific heat capacity, wide operating range, safety, and environmental friendliness.

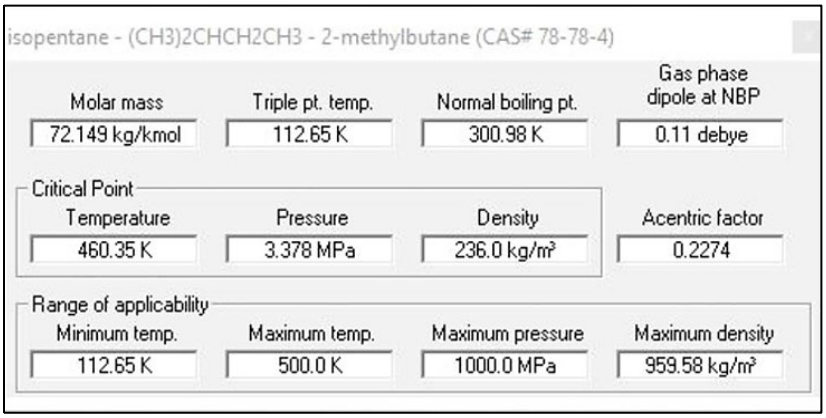

Fig.3 Limitations of Isopentane Working Fluid

III. THEORETICAL CALCULATION

To calculate the energy calculations and efficiency of the cycle. We use different formulas: For pump work at the initial stage;

1. For turbine work at the exit;
2. For energy addition;
3. For energy rejection;
4. After this, we determine the efficiency of the system;
5. The destruction of exergy by any cycle is related to the amount of heat added/subtracted by the high temperature and low temperature reservoirs and the respective temperatures. It may also be presented on a per unit mass basis as; in case of simple cycle s1=s2, s3=s4;

We are using REFPROP software to determine the thermodynamic as well as thermophysical properties of the working fluid.

$x_{dest,23} = T_o\left(s_3 - s_0 - \dfrac{Q_{in,23}}{T_{source}}\right)$	(6)
$x_{dest,41} = T_o\left(s_1 - s_4 - \dfrac{Q_{\mathrm{out},41}}{T_{\mathrm{sink}}}\right)$ The exergy of a fluid stream in any state may be calculated from;	(7)
$\mathrm{Exergy}_1 = \left(h_1 - h_2\right) - T_0\left(s_1 - s_2\right)$	(8)
$\mathrm{Exergy}_2 = \left(h_2 - h_3\right) - T_0\left(s_2 - s_3\right)$	(9)
$\mathrm{Exergy}_3 = \left(h_3 - h_4\right) - T_0\left(s_3 - s_4\right)$	(10)
$\mathrm{Exergy}_4 = \left(h_4 - h_1\right) - T_0\left(s_4 - s_1\right)$ Second law efficiency of the cycle;	(11)
$x_{expanded} = x_{heat,in} + x_{pump,in}$	(12)
$x_{heat,in} = 1 - \left(\dfrac{T_0}{T_h}\right) q_{in}$	(13)
$\eta_{th} = 1 - \dfrac{x_{destroyed}}{x_{expended}}$	

	Temperature (K)	Pressure (MPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Volume (m³/kg)	Vapor Volume (m³/kg)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)	Liquid Exergy (kJ/kg)	Vapor Exergy (kJ/kg)
1	350.61	0.43000	556.98	12.100	0.0017954	0.082646	120.79	420.44	0.36920	1.2238	164.68	209.52
2	359.34	0.53000	546.13	14.871	0.0018311	0.067246	143.64	434.14	0.43306	1.2415	168.49	217.96
3	366.92	0.63000	536.31	17.680	0.0018646	0.056562	163.91	445.98	0.48835	1.2571	172.27	225.14
4	373.65	0.73000	527.24	20.536	0.0018967	0.048695	182.26	456.43	0.53740	1.2712	176.00	231.40
5	379.73	0.83000	518.73	23.448	0.0019278	0.042647	199.13	465.77	0.58168	1.2839	179.67	236.95
6	385.29	0.93000	510.64	26.423	0.0019583	0.037845	214.82	474.22	0.62220	1.2955	183.28	241.94
7	390.42	1.0300	502.88	29.469	0.0019885	0.033934	229.55	481.92	0.65966	1.3061	186.84	246.48
8	395.19	1.1300	495.38	32.592	0.0020186	0.030682	243.47	488.98	0.69459	1.3158	190.34	250.63
9	399.66	1.2300	488.09	35.801	0.0020488	0.027932	256.70	495.47	0.72737	1.3248	193.80	254.45
10	403.87	1.3300	480.95	39.105	0.0020792	0.025572	269.35	501.45	0.75833	1.3330	197.21	257.97

Fig.4 Thermophysical Properties of Isopentane from REFPROP

TABLE I SPECIFIC STATE POINTS CALCULATION OF ISOPENTANE FLUID

States	Temperature (K)	Pressure (MPa)	Density (kg/m³)	Volume (m³/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Exergy (kJ/kg)
1	350.61	0.43000	556.99	0.0017954	120.79	0.36919	164.68
2	351.77	2.6300	561.16	0.0017820	124.73	0.36919	168.62
3	426.38	2.6300	444.82	0.0022481	338.53	0.91822	218.73
4	350.61	0.43000	18.610	0.053734	313.29	0.91822	193.48

Due to working fluid limit, we select the system pressure range from 0.43 MPa to 2.63 MPa and in the superheating case the temperature is 426.38 K.

State 1:
P1 = 0.43 MPa, h1= hf at 0.43Mpa = 120.79 kJ/kg,
v1= vf at 0.43 MPa = 0.0017954 m3/kg, s1= 0.36919 kJ/kg-K

State 2:
P2 = 2.63 MPa, s1 = s2, h2= 124.73 kJ/kg
Using formula (1) to determine the pump work; Assume mass flow rate= 2.5 kg/s

$$W_{pump,in} = 2.5\left(124.73 - 120.79\right) = 9.85kW$$

State 3:
P3 = 2.63 MPa, T3=426.38 K, h3= 338.53 kJ/kg, s3= 0.91822 kJ/kg-K

State 4:
P4= 0.43 MPa, h4= 313.29 kJ/kg, s3=s4
Using formula (2) to determine the turbine work; Assume mass flow rate= 2.5 kg/s

$$W_{Turbine,out} = 2.5\left(338.53 - 313.29\right) = 63.1kW$$

Using formula (3) to determine the addition of energy;

$$Q_{in} = 338.53 - 124.73 = 213.8kJ/kg$$

Using formula (4) to determine rejection of energy;

$$Q_{out} = 313.29 - 120.79 = 192.5kJ/kg$$

Using formula (5) to determine the system efficiency;

$$\eta_{th} = 1 - \frac{192.5}{213.8} = 10\%$$

Due to isentropic case of ideal cycle;

$$x_{dest,12} = 0,\ x_{dest,34} = 0$$

Using formula (6) to determine the exergy destruction state 2-3;
Assume the value of T_O= 300 K, Tsource= 450 K, s3= 0.91822 kJ/kg-K, s0= 0.36919 kJ/kg-K

$$x_{dest,23} = 300\left(0.918220 - 0.36919 - \frac{213.8}{300}\right)$$

$$x_{dest,23} = 22.17kJ/kg$$

Using formula (7) to determine the exergy destruction state 4-1;

$$x_{dest,41} = 300\left(0.36919 - 0.91822 + \frac{192.5}{300}\right)$$

$$x_{dest,41} = 27.791kJ/kg$$

Total Exergy destruction calculation;

$$x_{dest,cycle} = x_{dest,12} + x_{dest,23} + x_{dest,34} + x_{dest,41}$$

$$x_{dest,cycle} = 49.961kJ/kg$$

Using formula (8) to determine the exergy of state 1;

$$\text{Exergy}_1 = \left(h_1 - h_2\right) - T_0\left(s_1 - s_0\right)$$

$$\text{Exergy}_1 = \left(120.79 - 124.73\right) - 3350.61\left(0.36919 - 0.36919\right)$$

$$\text{Exergy}_1 = 164.68\ kJ/kg$$

Using formula (9) to determine the exergy of state 2;

$$\text{Exergy}_2 = (h_2 - h_3) - T_0(s_2 - s_3)$$

$$\text{Exergy}_2 = (124.73 - 338.53) - 351.77(0.36919 - 0.91822)$$

$$\text{Exergy}_2 = 168.62\ kJ/kg$$

Using formula (10) to determine the exergy of state 3;

$$\text{Exergy}_3 = (h_3 - h_4) - T_0(s_3 - s_4)$$

$$\text{Exergy}_3 = (338.53 - 313.29) - 426.38(0.91822 - 0.91822)$$

$$\text{Exergy}_3 = 218.03\ kJ/kg$$

Using formula (11) to determine the exergy of state 4;

$$\text{Exergy}_4 = (h_4 - h_1) - T_0(s_4 - s_1)$$

$$\text{Exergy}_4 = (313.29 - 120.79) - 350.61(0.91822 - 0.36919)$$

$$\text{Exergy}_4 = 193.48\ kJ/kg$$

$$\text{Total Exergy} = \text{Exergy}_1 + \text{Exergy}_2 + \text{Exergy}_3 + \text{Exergy}_4$$

$$\text{Total Exergy} = 164.68 + 168.62 + 218.73 + 193.48 = 745.51\ kJ/kg$$

Using formula (14) to find second law of efficiency of cycle;

$$\eta_{th} = 1 - \frac{x_{destroyed}}{x_{\exp anded}}$$

where;

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

Using formula (13) to find heat addition;

$$x_{heat,in} = 1 - \left(\frac{350.61}{426.38}\right)213.8$$

$$x_{heat,in} = 59.86 kJ/kg$$

Then using formula (12);

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

$$x_{expanded} = 59.86 + 2.63 = 62.49 kJ/kg$$

Then for second law efficiency;

$$\eta_{th} = 1 - \frac{49.961}{62.49}$$

$$\eta_{th} = 21\%$$

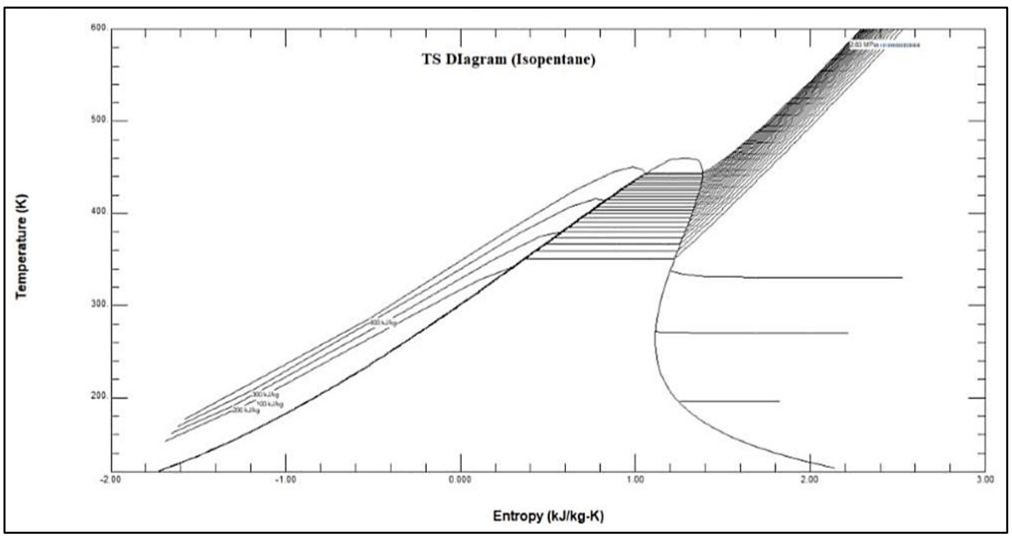

Fig.5 Temperature Entropy Plot of Isopentane

According to the hypothesis simplification, the kinetic and potential energy of the system will be zero.

IV. OPTIMIZATION AND EFFICIENCY INCREASING METHODS

Most of the electric power generation around the globe utilizes such power cycles for generating electricity. The consumption of fuels can be significantly reduced by even a minor enhancement in the thermal efficacy of these power cycles. All modifications implemented to enhance the thermal efficiency of a specific power cycle have been predicated on a single fundamental principle: increasing the average temperature at which heat is distributed to the fluid in the boiler or decreasing the average temperature at which heat is rejected from the fluid in the condenser.

Case 1: (Decrease the condenser pressure to 0.15 MPa)

State 1:
P_1 = 0.15 MPa, h_1= h_f at 0.15 MPa = 27.55 kJ/kg,
v_1= v_f at 0.15 MPa = 0.0016674 m^3/kg, s_1= 0.089509 kJ/kg-K

State 2:
P_2 = 2.63 MPa, $s_1 = s_2$, h_2= 31.675 kJ/kg

Using formula (1) to determine the pump work; Assume mass flow rate= 2.5 kg/s

$$W_{pump,in} = 2.5(31.675 - 27.55) = 10.31 kW$$

State 3:
P_3 = 2.63 MPa, T_3 = 426.38 K, h_3 = 338.53 kJ/kg, s_3 = 0.0022481 kJ/kg-K

	Temperature (K)	Pressure (MPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Volume (m³/kg)	Vapor Volume (m³/kg)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)	Liquid Exergy (kJ/kg)	Vapor Exergy (kJ/kg)
1	312.83	0.15000	599.72	4.4169	0.0016674	0.22640	27.555	361.44	0.089525	1.1568	154.83	170.50
2	329.97	0.25000	581.07	7.1671	0.0017210	0.13953	68.768	388.05	0.21723	1.1849	157.96	188.75
3	342.45	0.35000	566.75	9.9028	0.0017645	0.10098	99.900	407.62	0.30932	1.2079	161.64	201.45
4	352.47	0.45000	554.71	12.651	0.0018027	0.079042	125.62	423.36	0.38281	1.2276	165.44	211.33
5	360.94	0.55000	544.09	15.429	0.0018379	0.064812	147.88	436.64	0.44471	1.2447	169.25	219.49
6	368.33	0.65000	534.44	18.247	0.0018711	0.054804	167.71	448.17	0.49860	1.2600	173.02	226.46
7	374.92	0.75000	525.50	21.114	0.0019030	0.047362	185.74	458.37	0.54660	1.2738	176.73	232.56
8	380.88	0.85000	517.08	24.038	0.0019339	0.041601	202.36	467.53	0.59006	1.2863	180.39	237.99
9	386.35	0.95000	509.06	27.026	0.0019644	0.037001	217.84	475.82	0.62992	1.2977	183.99	242.89
10	391.40	1.0500	501.36	30.087	0.0019946	0.033237	232.39	483.38	0.66684	1.3081	187.54	247.34

Fig.6 Thermophysical Properties of Isopentane at Pressure Range 0.15 MPa to 2.63 Mpa

TABLE II SPECIFIC STATE POINTS CALCULATION AT DECREASING CONDENSER PRESSURE

States	Temperature (K)	Pressure (MPa)	Density (kg/m³)	Volume (m³/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Exergy (kJ/kg)
1	312.83	0.15000	599.72	0.0016674	27.550	0.089509	154.83
2	313.79	2.6300	602.77	0.0016590	31.675	0.089509	158.95
3	426.38	2.6300	444.82	0.0022481	338.53	0.91822	218.73
4	312.83	0.15000	5.6766	0.17616	286.80	0.91822	167.00

State 4:

P_4= 0.15 MPa, h_4= 286.80 kJ/kg, s_3=s_4

Using formula (2) to determine the turbine work; Assume mass flow rate= 2.5 kg/s

$$W_{Turbine,out} = 2.5\,(338.53 - 286.8) = 129.32kW$$

Using formula (3) to determine the addition of energy;

$$Q_{in} = 338.53 - 31.675 = 306.85\ kJ/kg$$

Using formula (4) to determine rejection of energy;

$$Q_{out} = 286.8 - 27.55 = 259.25\ kJ/kg$$

Using formula (5) to determine the system efficiency;

$$\eta_{th} = 1 - \frac{259.25}{306.85}$$

$$\eta_{th} = 16\%$$

Due to isentropic case of ideal cycle;

$$x_{dest,12} = 0,\ x_{dest,34} = 0$$

$x_{dest,12}$= 0, $x_{dest,34}$ = 0

Using formula (6) to determine the exergy destruction state 2-3;

Assume the value of T_o = 300 K, T_{source}= 450 K, s_3= 0.91822 kJ/kg-K, s_0= 0.089509 kJ/kg-K

$$x_{dest,23} = 300\left(0.918220 - 0.089509 - \frac{306.85}{450}\right)$$

$$x_{dest,23} = 41.649\ kJ/kg.K$$

Using formula (7) to determine the exergy destruction state 4-1;

$$x_{dest,41} = 300\left(0.089509 - 0.91822 + \frac{259.25}{300}\right)$$

$$x_{dest,41} = 10.636\ kJ/kg.K$$

Total Exergy destruction calculation;

$$x_{dest,cycle} = x_{dest,12} + x_{dest,23} + x_{dest,34} + x_{dest,41}$$

$$x_{dest,cycle} = 52.285\ kJ/kg$$

Using formula (8) to determine the exergy of state 1;

$$\text{Exergy}_1 = (h_1 - h_2) - T_0(s_1 - s_2)$$

$$\text{Exergy}_1 = (27.55 - 31.675) - 312.83\,(0.089509 - 0.08959)$$

$$\text{Exergy}_1 = 154.83\ kJ/kg$$

Using formula (9) to determine the exergy of state 2;

$$\text{Exergy}_2 = (h_2 - h_3) - T_0(s_2 - s_3)$$

$$\text{Exergy}_2 = (31.675 - 338.53) - 313.79(0.91822 - 0.089509)$$

$$\text{Exergy}_2 = 158.95\ kJ/kg$$

Using formula (10) to determine the exergy of state 3;

$$\text{Exergy}_3 = (h_3 - h_4) - T_0(s_3 - s_4)$$

$$\text{Exergy}_3 = (338.53 - 286.80) - 426.38\,(0.91822 - 0.91822)$$

$$\text{Exergy}_3 = 218.73\ kJ/kg$$

Using formula (11) to determine the exergy of state 4;

$$\text{Exergy}_4 = (h_4 - h_1) - T_0(s_4 - s_1)$$

$$\text{Exergy}_4 = (286.8 - 27.55) - 312.83\,(0.91822 - 0.089509)$$

$$\text{Exergy}_4 = 167\ kJ/kg$$

$$\text{Total Exergy} = \text{Exergy}_1 + \text{Exergy}_2 + \text{Exergy}_3 + \text{Exergy}_4$$

$$\text{Total Exergy} = 154.83 + 158.95 + 218.73 + 167$$

$$\text{Total Exergy} = 699.51\ kJ/kg$$

Using formula (14) to find second law of efficiency of cycle; where;

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

Using formula (13) to find heat addition;

$$x_{heat,in} = 1 - \left(\frac{312.83}{426.38}\right)306.85$$

$$x_{heat,in} = 82.84 kJ/kg$$

Then using formula (12);

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

$$x_{expanded} = 82.84 + 2.63 = 85.47 kJ/kg$$

Then for second law efficiency;

$$\eta_{th} = 1 - \frac{52.28}{85.47}$$

$$\eta_{th} = 30\%$$

Case 2: (Superheating at Temperature 455 K)

	Temperature (K)	Pressure (MPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Volume (m³/kg)	Vapor Volume (m³/kg)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)	Liquid Exergy (kJ/kg)	Vapor Exergy (kJ/kg)
1	350.61	0.43000	556.98	12.100	0.0017954	0.082646	120.79	420.44	0.36920	1.2238	164.68	209.52
2	359.34	0.53000	546.13	14.871	0.0018311	0.067246	143.64	434.14	0.43306	1.2415	168.49	217.96
3	366.92	0.63000	536.31	17.680	0.0018646	0.056562	163.91	445.98	0.48835	1.2571	172.27	225.14
4	373.65	0.73000	527.24	20.536	0.0018967	0.048695	182.26	456.43	0.53740	1.2712	176.00	231.40
5	379.73	0.83000	518.73	23.448	0.0019278	0.042647	199.13	465.77	0.58168	1.2839	179.67	236.95
6	385.29	0.93000	510.64	26.423	0.0019583	0.037845	214.82	474.22	0.62220	1.2955	183.28	241.94
7	390.42	1.0300	502.88	29.469	0.0019885	0.033934	229.55	481.92	0.65966	1.3061	186.84	246.48
8	395.19	1.1300	495.38	32.592	0.0020186	0.030682	243.47	488.98	0.69459	1.3158	190.34	250.63
9	399.66	1.2300	488.09	35.801	0.0020488	0.027932	256.70	495.47	0.72737	1.3248	193.80	254.45
10	403.87	1.3300	480.95	39.105	0.0020792	0.025572	269.35	501.45	0.75833	1.3330	197.21	257.97

Fig.7 Thermophysical Properties of Isopentane at Temperature of 455 k Range

TABLE III SPECIFIC STATE POINTS CALCULATION AT SUPERHEATING

States	Temperature (K)	Pressure (Mpa)	Density (kg/m³)	Volume (m³/kg)	Enthalpy (Kj/kg)	Entropy (Kj/kg-K)	Exergy (Kj/kg)
1	350.61	0.43000	556.99	0.0017954	120.79	0.36919	164.68
2	351.77	2.6300	561.16	0.0017820	124.73	0.36919	168.62
3	455.00	2.6300	81.755	0.012232	586.17	1.4776	299.59
4	395.55	0.43000	10.217	0.097880	515.06	1.4776	228.48

State 1:

$P_1 = 0.43$ MPa, $h_1 = h_f$ at 0.43 MPa = 120.79 kJ/kg,
$v_1 = v_f$ at 0.43 MPa = 0.0017954 m³/kg, $s_1 = 0.36919$ kJ/kg-K

State 2:

$P_2 = 2.63$ MPa, $s_1 = s_2$, $h_2 = 124.73$ kJ/kg

Using formula (1) to determine the pump work; Assume mass flow rate= 2.5 kg/s

$$W_{pump,in} = 2.5\ (124.73 - 120.79) = 9.85 kW$$

State 3:

$P_3 = 2.63$ MPa, T_3=455 K, $h_3 = 586.17$ kJ/kg, $s_3 = 1.4776$ kJ/kg-K

State 4:

$P_4 = 0.43$ MPa, $h_4 = 515.06$ kJ/kg, $s_3 = s_4$

Using formula (2) to determine the turbine work; Assume mass flow rate= 2.5 kg/s

$$W_{Turbine,out} = 2.5\ (586.17 - 515.06) = 177.77 kW$$

Using formula (3) to determine the addition of energy;

Q_{in}= 586.17 – 124.73 = 461.44 kJ/kg

Using formula (4) to determine rejection of energy;

$$Q_{out} = 515.06 - 120.79 = 394.27\ kJ/kg$$

Using formula (5) to determine the system efficiency;

$$\eta_{th} = 1 - \frac{394.27}{461.44}$$

$$\eta_{th} = 15\%$$

Due to isentropic case of ideal cycle;

$$x_{dest,12} = 0, \quad x_{dest,34} = 0$$

Assume the value of T_o= 300 K, T_{source}= 450 K, s_3= 1.4776 kJ/kg-K, s_0= 0.36919 kJ/kg-K Using formula (6) to determine the exergy destruction state 2-3;

$$x_{dest,23} = 300\left(1.4776 - 0.36919 - \frac{461.44}{450}\right)$$

$$x_{dest,23} = 24.9\ kJ/kg.K$$

Using formula (7) to determine the exergy destruction state 4-1;

$$x_{dest,41} = 300\left(0.36919 - 1.4776 + \frac{394.27}{300}\right)$$

$$x_{dest,41} = 61.747 kJ/kg$$

Total Exergy destruction calculation;

$$x_{dest,cycle} = x_{dest,12} + x_{dest,23} + x_{dest,34} + x_{dest,41}$$

$$x_{dest,cycle} = 86.647 kJ/kg$$

Using formula (8) to determine the exergy of state 1;

$$\text{Exergy}_1 = (h_1 - h_2) - T_0(s_1 - s_0)$$

$$\text{Exergy}_1 = (120.79 - 124.73) - 350.61(0.36919 - 0.36919)$$

$$\text{Exergy}_1 = 164.68 \ kJ/kg$$

Using formula (9) to determine the exergy of state 2;

$$\text{Exergy}_2 = (h_2 - h_3) - T_0(s_2 - s_3)$$

$$\text{Exergy}_2 = (124.73 - 586.17) - 351.77(0.36919 - 1.4776)$$

$$\text{Exergy}_2 = 168.62 \ kJ/kg$$

Using formula (10) to determine the exergy of state 3;

$$\text{Exergy}_3 = (h_3 - h_4) - T_0(s_3 - s_4)$$

$$\text{Exergy}_3 = (586.17 - 515.06) - 455(1.4776 - 1.4776)$$

$$\text{Exergy}_3 = 299.59 \ kJ/kg$$

Using formula (11) to determine the exergy of state 4;

$$\text{Exergy}_4 = (h_4 - h_1) - T_0(s_4 - s_1)$$

$$\text{Exergy}_4 = (515.06 - 120.79) - 395.55(1.4776 - 0.36919)$$

$$\text{Exergy}_4 = 228.48 \ kJ/kg$$

$$\text{Total Exergy} = \text{Exergy}_1 + \text{Exergy}_2 + \text{Exergy}_3 + \text{Exergy}_4$$

$$\text{Total Exergy} = 164.68 + 168.62 + 299.59 + 228.48 = 861.37 \ kJ/kg$$

Using formula (14) to find second law of efficiency of cycle;

$$\eta_{th} = 1 - \frac{x_{destroyed}}{x_{\exp anded}}$$

where;

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

Using formula (13) to find heat addition;

$$x_{heat,in} = 1 - \left(\frac{350.61}{455}\right)461.44$$

$$x_{heat,in} = 106.13 kJ/kg$$

Then using formula (12);

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

$$x_{expanded} = 106.13 + 2.63 = 108.7 kJ/kg$$

Then for second law efficiency;

$$\eta_{th} = 1 - \frac{86.47}{108.7}$$

$$\eta_{th} = 21\%$$

Case 3: (Increasing Evaporator Pressure to 3.32 MPa)

	Temperature (K)	Pressure (MPa)	Liquid Density (kg/m³)	Vapor Density (kg/m³)	Liquid Volume (m³/kg)	Vapor Volume (m³/kg)	Liquid Enthalpy (kJ/kg)	Vapor Enthalpy (kJ/kg)	Liquid Entropy (kJ/kg-K)	Vapor Entropy (kJ/kg-K)	Liquid Exergy (kJ/kg)	Vapor Exergy (kJ/kg)
1	350.61	0.43000	556.98	12.100	0.0017954	0.082646	120.79	420.44	0.36920	1.2238	164.68	209.52
2	359.34	0.53000	546.13	14.871	0.0018311	0.067246	143.64	434.14	0.43306	1.2415	168.49	217.96
3	366.92	0.63000	536.31	17.680	0.0018646	0.056562	163.91	445.98	0.48835	1.2571	172.27	225.14
4	373.65	0.73000	527.24	20.536	0.0018967	0.048695	182.26	456.43	0.53740	1.2712	176.00	231.40
5	379.73	0.83000	518.73	23.448	0.0019278	0.042647	199.13	465.77	0.58168	1.2839	179.67	236.95
6	385.29	0.93000	510.64	26.423	0.0019583	0.037845	214.82	474.22	0.62220	1.2955	183.28	241.94
7	390.42	1.0300	502.88	29.469	0.0019885	0.033934	229.55	481.92	0.65966	1.3061	186.84	246.48
8	395.19	1.1300	495.38	32.592	0.0020186	0.030682	243.47	488.98	0.69459	1.3158	190.34	250.63
9	399.66	1.2300	488.09	35.801	0.0020488	0.027932	256.70	495.47	0.72737	1.3248	193.80	254.45
10	403.87	1.3300	480.95	39.105	0.0020792	0.025572	269.35	501.45	0.75833	1.3330	197.21	257.97

Fig.8 Thermophysical Properties of Isopentane at Pressure Range 0.43 MPa to 3.32 MPa

TABLE IV SPECIFIC STATE POINTS CALCULATION AT INCREASING BOILER OR EVAPORATOR PRESSURE

States	Temperature (K)	Pressure (MPa)	Density (kg/m³)	Volume (m³/kg)	Enthalpy (kJ/kg)	Entropy (kJ/kg-K)	Exergy (kJ/kg)
1	350.61	0.43000	556.99	0.0017954	120.79	0.36919	164.68
2	352.06	3.2000	562.19	0.0017787	125.74	0.36919	169.63
3	426.38	3.2000	450.81	0.0022182	337.06	0.91180	219.18
4	350.61	0.43000	18.823	0.053126	311.04	0.91180	193.15

State 1:

$P_1 = 0.43$ MPa, $h_1 = h_f$ at 0.43 MPa = 120.79 kJ/kg,
$v_1 = v_f$ at 0.43 MPa = 0.0017954 m³/kg, $s_1 = 0.36919$ kJ/kg-K

State 2:

$P_2 = 3.2$ MPa, $s_1 = s_2$, $h_2 = 125.74$ kJ/kg

Using formula (1) to determine the pump work; Assume mass flow rate = 2.5 kg/s

$$W_{pump,in} = 2.5\,(125.74 - 120.79) = 12.375 kW$$

State 3:

P_3 = 3.2 MPa, T_3=426.38 K, h_3 = 337.06 kJ/kg, s_3= 0.91180 kJ/kg-K

State 4:

P_4= 0.43 MPa, h_4= 311.04 kJ/kg, s_3=s_4

Using formula (2) to determine the turbine work; Assume mass flow rate= 2.5 kg/s

$$W_{Turbine,out} = 2.5\left(337.06 - 311.04\right) = 65.05 kW$$

Using formula (3) to determine the addition of energy;

$$Q_{in} = 337.06 - 125.74 = 211.32\ kJ/kg$$

Using formula (4) to determine rejection of energy;

$$Q_{out} = 311.04 - 120.79 = 190.25\ kJ/kg$$

Using formula (5) to determine the system efficiency;

$$\eta_{th} = 1 - \frac{190.25}{211.32}$$

$$\eta_{th} = 11\%$$

Due to isentropic case of ideal cycle;

$$x_{dest,12} = 0,\ x_{dest,34} = 0$$

Assume the value of T_o= 300 K, T_{source}= 450 K, s_3= 0.91180 kJ/kg-K, s_0= 0.36919 kJ/kg-K

Using formula (6) to determine the exergy destruction state 2-3;

$$x_{dest,23} = 300\left(0.91180 - 0.36919 - \frac{211.32}{450}\right)$$

$$x_{dest,23} = 21.903\ kJ/kg$$

Using formula (7) to determine the exergy destruction state 4-1;

$$x_{dest,41} = 300\left(0.36919 - 0.91180 + \frac{190.25}{300}\right)$$

$$x_{dest,41} = 27.467\ kJ/kg$$

Total Exergy destruction calculation;

$$x_{dest,cycle} = x_{dest,12} + x_{dest,23} + x_{dest,34} + x_{dest,41}$$

$$x_{dest,cycle} = 49.37\ kJ/kg$$

Using formula (8) to determine the exergy of state 1;

$$\text{Exergy}_1 = \left(h_1 - h_2\right) - T_0\left(s_1 - s_0\right)$$

$$\text{Exergy}_1 = \left(120.79 - 124.73\right) - 350.61\left(0.36919 - 0.36919\right)$$

$$\text{Exergy}_1 = 164.68\ kJ/kg$$

Using formula (9) to determine the exergy of state 2;

$$\text{Exergy}_2 = \left(h_2 - h_3\right) - T_0\left(s_2 - s_3\right)$$

$$\text{Exergy}_2 = \left(124.73 - 337.06\right) - 352.06\left(0.36919 - 0.91180\right)$$

$$\text{Exergy}_2 = 169.63\ kJ/kg$$

Using formula (10) to determine the exergy of state 3;

$$\text{Exergy}_3 = \left(h_3 - h_4\right) - T_0\left(s_3 - s_4\right)$$

$$\text{Exergy}_3 = \left(337.06 - 311.04\right) - 426.38\left(0.91180 - 0.91180\right)$$

$$\text{Exergy}_3 = 219.18\ kJ/kg$$

Using formula (11) to determine the exergy of state 4;

$$\text{Exergy}_4 = \left(h_4 - h_1\right) - T_0\left(s_4 - s_1\right)$$

$$\text{Exergy}_4 = \left(311.04 - 120.79\right) - 350.61\left(0.91180 - 0.36919\right)$$

$$\text{Exergy}_4 = 193.15\ kJ/kg$$

$$\text{Total Exergy} = \text{Exergy}_1 + \text{Exergy}_2 + \text{Exergy}_3 + \text{Exergy}_4$$

$$\text{Total Exergy} = 164.68 + 169.63 + 219.18 + 193.15 = 746.64\ kJ/kg$$

Using formula (14) to find second law of efficiency of cycle;

$$\eta_{th} = 1 - \frac{x_{destroyed}}{x_{expanded}}$$

where;

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

Using formula (13) to find heat addition;

$$x_{heat,in} = 1 - \left(\frac{350.61}{426.38}\right)211.32$$

$$x_{heat,in} = 59.16 kJ/kg$$

Then using formula (12);

$$x_{expanded} = x_{heat,in} + x_{pump,in}$$

$$x_{expanded} = 59.16 + 3.2 = 62.36 kJ/kg$$

Then for second law efficiency;

$$\eta_{th} = 1 - \frac{49.37}{62.36}$$

$$\eta_{th} = 21\%$$

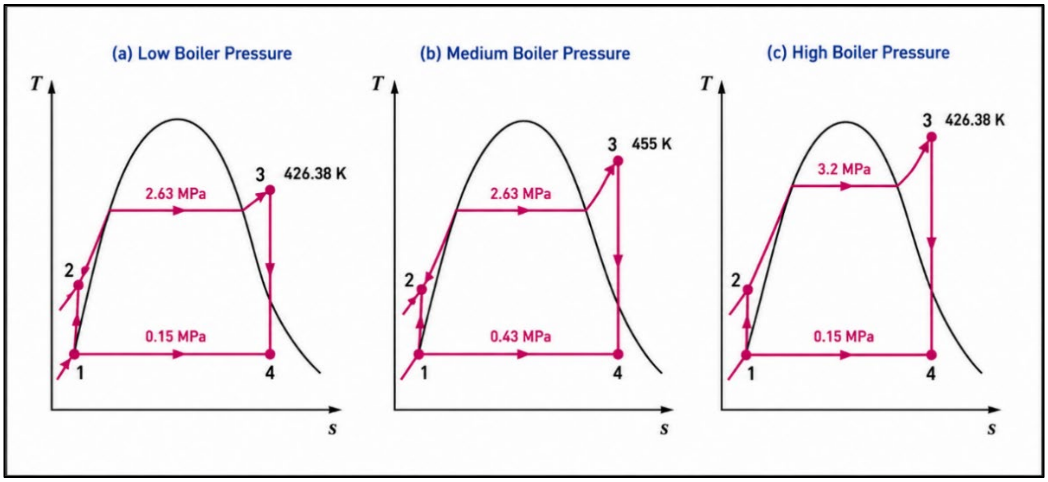

Fig.9 Temperature Entropy Plot of Isopentane: (a) Low Condenser Pressure; p=0.15MPa; (b) Superheating; t=455 k, (c) High Boiler or Evaporator Pressure; p=3.2 MPa

V. SOLAR ENERGY ADDITION

The Rankine cycle is widely regarded as the most efficient and preferable power cycle for the generation of electric power from solar thermal energy. The components employed in a solar thermal Rankine power plant are as follows: (1) a solar collector, (2) a thermal energy storage device, and (3) a Rankine cycle. It should be noted that water is the most appropriate working fluid for sources of heat that operate at elevated temperatures (above 370°C). At the same time, steam Rankine power plants show lower effectiveness and higher cost for operation when heat sources operate at lower temperatures. In this regard, organic Rankine cycles make use of organic working fluid (hydrocarbons, refrigerants, siloxanes) that ensures much higher effectiveness in the process of heat utilization from sources operating at low and medium temperatures due to their low boiling point. Examples of solar ORC system include the solar thermal power plant, which produces 9.85 kW of electric power, and its collector field area is exceeding.

The HTF inlet temperature increases as a result of the decrease in the HTF mass flow rate at the pinch-point temperature (ΔTPP). However, the HTF's exhaust temperature rises as the mass flow rate increases. Consequently, it is evident that the reduction of the average temperature of the HTF is a challenging issue, and it is one of the most significant drawbacks of operating the solar collector at high temperatures rather than low temperatures. The Rankine cycle generates 9.85 kW of inlet energy, and then we connect the solar thermal collector to the storage tank. Water serves as a working fluid which is ideal for use in solar Rankine cycle. Therefore, in this case, we reverse the process, and the inlet energy generated by the pump remains the same as that of the solar panel. The solar collector heats up the storage tank fluid during the day, and at night, the heating fluid performs well. This cycle works efficiently with an efficiency of around 35%, and the payback period of this system is 3-4 years.

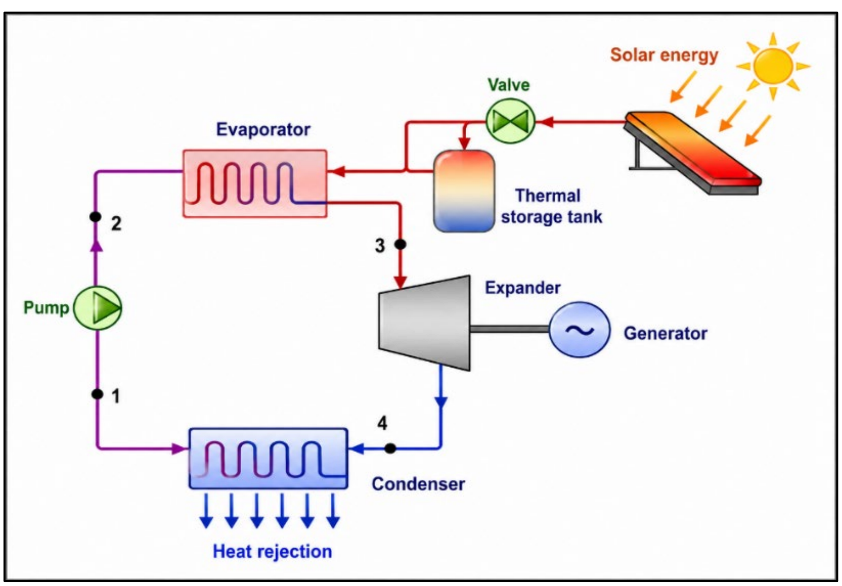

Fig.10 Schematic Diagram of Solar Energy Addition in the Organic Rankine Cycle

VI. ECONOMIC ANALYSIS

Economic evaluation of the system to determine the production and the cost of per kilowatt according to the different organization.

$W_{net} = W_{turbine, out} – W_{pump, in}$

$W_{net} = 63.1 – 9.85 = 53.25$ kW

To determine the annual production of electricity or power:

Annual Production= $W_{net} \times$ 24hours × 30days × 12months

= 53.25 × 24 × 30 × 12 = 460080 kW

Monthly production of electricity = 38340 kW

According to the cost and annual production of electricity. per kilowatt price of the energy generation is;

TABLE V COST ANALYSIS

Components	Specification	Cost in $
Working Fluid (Isopentane)	15Litres	2000
Solar Collector	2kW	7000
Pump	3MPa	3000
Evaporator	-	3000
Generator or Turbine	100kW	4000
Condenser	-	8000
Other Components	-	6000
Labor Cost	-	6000
Grand Total		39000

TABLE VI ENERGY GENERATION

Energy Production (kW)	460080
Monthly (kW)	38340
1kW Cost ($)	1000
Payback Time (years)	5

The efficiencies that could be realized will be roughly double the case of the drying application, with an efficiency of 21%. The power output of an appropriately configured ORC unit will be 63 kW, with the total cost of installation being $3.9 million. With this configuration running not more than 3,000 hours per year, this would mean saving $562,500 annually (assuming $0.15/kWh) leading to an overall five years payback period. Ideally, in the event that this configuration runs continuously during the year, then the payback period will reduce by more than half. Currently, the thermal energy produced at the Rutgers combined heat and power plant is relied on for heating and cooling periods, thus reducing the availability of waste heat and thereby elongating the payback period.

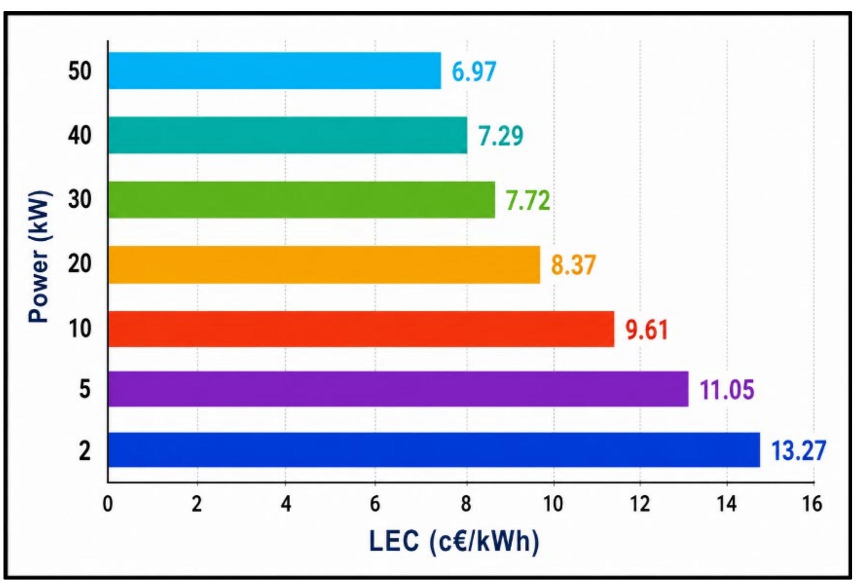

Fig.11 Cost Evaluation

VII. RESULTS AND DISCUSSION

From the result, it is seen that improvement of the cycle depends upon the degree of superheating as well as other cases. From the table, it is noticed that there exists exergy and exergy destruction in the system, while the system has relatively high efficiency as far as isopentane as a working fluid. In addition, taking into consideration the high design value for Tcond, the efficiency of the ORC power system is more than 30% and the efficiency of SF is around 25%. As the steam power plants are relatively large in size and use indirect thermal storage system, their efficiencies are 21%, whereas there is no estimation of any kind of annual performance index in the case of ORC power system. Thus, thermal losses and exergy losses due to the degradation of the segmentation are neglected.

TABLE VII ENERGY CALCULATION AND ITS RESULTS

Case	Thermal Efficiency (%)	Total Exergy Destruction (kJ/kg)	Total Exergy (kJ/kg)	2nd Law of Efficiency (%)
P_1= 0.43, P_2= 2.63MPa, T_3= 426.38K	10	49.961	745.51	21
P_1= 0.15, P_2= 2.63MPa, T_3= 426.38K	16	52.285	699.51	30
P_1= 0.43, P_2= 2.63MPa, T_3= 455K	15	86.647	861.37	21
P_1= 0.43, P_2= 3.2MPa, T_3= 426.38K	11	49.37	746.64	21

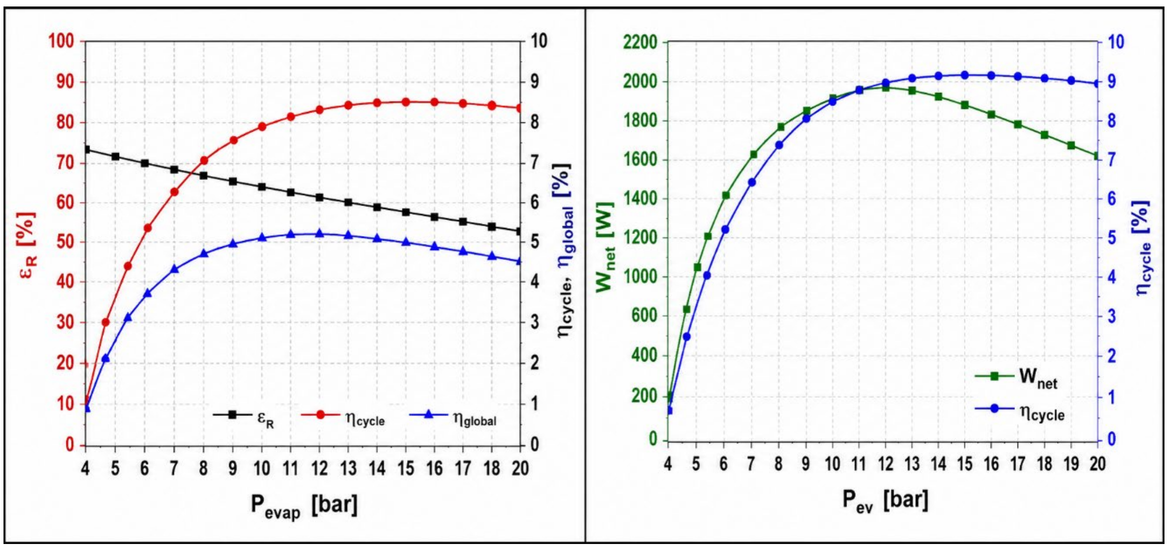

Fig.12 Graph of Efficiency vs Pressure and Work Net vs Pressure

VIII. CONCLUSION AND RECOMMENDATIONS

One more application which was investigated in the past and could be an interesting subject for the future is the transformation of solar radiation in outer space. The development of the International Space Station along with the use of the Stirling engine and Closed Brayton Cycle gas turbine was accompanied by intensive research of ORC systems, including experimental investigations. One can not exclude that with the growing number of artificial objects in space and the intensification of space exploration, special solar or even nuclear-powered ORC systems may appear, where the key advantage over photovoltaics could be efficient production of energy necessary for heating or cooling the on-board equipment.

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