Chapter 5: Case Study for Nuclear Power Plant

5.1. Selection of Location – Feasibility Study

The real data of working NPP, provided by BRNS, India, have been used for analysis. The possible waste heat locations are mentioned in Table 5.1.

A steam and feed water system are a device that enhances a steam turbine's ability to control a feedwater pump and plant operation characteristics when water is delivered slowly. The main feedwater duct on the discharge side of the feedwater pumps is equipped with a cut-off valve and is connected parallel with a bypass duct containing a pressure-compensated flow control valve in a nuclear power plant where the reactor's feedwater pumps are driven by a steam turbine. This system allows water to be fed to the reactor through the main feedwater duct from feedwater pumps powered by the steam turbine when the rate of feedwater is high and the cut-off valve is closed when the rate is low. In this system, the main steam condenser is the essential component in which steam is condensed and the feed water pump is used to supply a sufficient amount of feed water by steam and feed water system.

The purpose of the Main Heat Transport (MHT) purification system is to maintain high reactor water purity in order to minimize the reduction of heat transfer in the nuclear reactor core by minimizing the deposition of water impurities on the fuel surfaces and to minimize secondary sources of beta and gamma radiations by removing the corrosion products, fission products and impurities in the primary system. The purification system comprises of pumps, heat exchangers such as regenerative coolers, isolation heat exchanger, non-regenerative coolers, filters, ion exchange columns and strainers storage tank and the associated instrumentation, piping and valves. The purification flow from MHT is passed through heat exchangers to cool it before passing it through an ion-exchange column and filters for purification and returned back to MHT after reheating in non-regenerative coolers.

The moderator in a nuclear power plant is a material, usually heavy water or graphite, that slows down neutrons, enabling nuclear reactions to occur in a controlled manner. The moderator is a crucial component of a nuclear reactor, and it is essential to maintain its temperature within a safe range. The moderator cooling system is responsible for removing heat generated by the nuclear reaction and maintaining the moderator's temperature. The moderator cooling system typically uses a closed-loop cooling system, which circulates water or another coolant through the moderator vessel to remove heat. The system comprises a moderator vessel, a heat exchanger, pumps, and a cooling tower. The moderator vessel is the primary component of the cooling system, where the moderator is located. The vessel is made of a suitable material, such as stainless steel, and is designed to withstand high temperatures and pressures. The coolant circulates through the vessel, removing heat from the moderator. The heat exchanger is another critical component of the moderator cooling system. The heat exchanger transfers heat from the coolant to the secondary coolant, which carries heat away from the power plant. The secondary coolant can be air, water, or another suitable fluid. Pumps are used to circulate the coolant. One of the critical challenges in the moderator cooling system is maintaining the temperature of the moderator within a safe range. If the moderator's temperature is too high, it can cause the nuclear reaction to become unstable, leading to a potential nuclear meltdown. Conversely, if the moderator's temperature is too low, it can reduce the efficiency of the reactor. To prevent these issues, the moderator cooling system is designed to maintain the moderator's temperature within a safe range.

The author has done a feasible study with the help of necessary input data and identified locations provided by BRNS (Board of Research in Nuclear Sciences). Within nine waste heat sources, the feasible cases to use ORC have been discussed in the Table 5.2.

Table 5.1. Waste heat disposal components and its temperature of Indian NPP data provided by BRNS

System	Component	Waste Heat Source [MWth]	Hot fluid inlet temperature [°C]	Hot fluid outlet temperature [°C]	Cold fluid inlet temperature [°C]	Cold fluid outlet temperature [°C]	Hot fluid flow rate [kg/s]	Cold fluid flow rate [kg/s]
Steam and Feed Water System	Main Steam Condenser	635	Heat Source Temp: 42.3 °C Steam condenses at a temp. of 42.3 °C at a pressure of 62.5/0.085 mm Hg/ kg/cm^2.		29	36	323	21600
MHT Purification System	Isolation Heat Exchanger	4.2	98	68	65	78	33.33	77
	Non-regenerative Heat Exchanger	3.2	68	45	35	42	33.33	110
Main Moderator Cooling System	Moderator Active Process Water Heat Exchanger No-1	25	75	50	35	47	235.5	498
	Moderator- Active Process Water Heat Exchanger No-2	25	75	50	35	47	235.5	498

Table 5.2. Input condition data to check the feasibility of running ORC

S.No.	System and component for waste heat source	Modification required in existing system or not	Source of Input condition	Comment
1.	Main steam condenser of steam and feed water system	Modification not required	$T_{hf,i}$=36 °C, $\dot{m}_{hf}$=21600 kg/s	(Cycle can not run) Hot fluid inlet temperature is very less (below condenser temperature). No cycle is possible at this condition.
2.	Main steam condenser of steam and feed water system	Modification required	$T_{hf,i}$=42.3 °C, $T_{hf,o}$=42.3 °C, P_{hf}=8.33 kPa, $\dot{m}_{hf}$=323 kg/s	(Cycle can not run) Hot fluid inlet temperature is very less to produce any work output from given condition. No cycle is possible at this condition.
3.	Isolation heat exchanger of MHT purification system	Modification not required	$T_{hf,i}$=78 °C, $\dot{m}_{hf}$=77 kg/s	(Cycle can not run) This case is technically not possible because waste heat from isolation condenser is taken in use for producing desalination water. Thus no direct waste heat is available to produce the power. Thus, no cycle is possible at this condition.
4.	Isolation heat exchanger of MHT purification system	Modification required	$T_{hf,i}$=98 °C, $T_{hf,o}$=68 °C, $\dot{m}_{hf}$=33.33 kg/s	**Case-1 (Cycle can run)** Cycle can run on the given condition to produce the work due to availability of hot fluid inlet temperature in adequate

				amount. Cycle is possible at this condition.
5.	Non-regenerative heat exchanger of MHT purification system when isolation heat exchanger is not considered	Modification required	$T_{hf,i}$=98 °C, $T_{hf,o}$=42 °C, $\dot{m}_{hf}$=33.33 kg/s	(Cycle can not run) Boiler pressure is lowered to the maximum possible (near lowest condenser temperature, i.e., 37°C) value to achieve 42°C boiler hot fluid (water) outlet temperature, but not. Further, by decreasing condenser temperature below 35°C, boiler hot fluid (water) outlet temperature 42°C is achieved by small amount of work output. At 35°C condenser temperature, corresponding cold fluid (water) inlet temperature of condenser must be below 27°C, which is hard to maintain ambient temperature in India throughout the year.
6.	Non-regenerative heat exchanger of MHT purification system	Modification not required	$T_{hf,i}$=42 °C, $\dot{m}_{hf}$=110 kg/s	(Cycle can not run) Hot fluid inlet temperature is very less to produce any work output from given condition. No cycle is possible at this condition.
7.	Non-regenerative heat exchanger of MHT purification system	Modification required	$T_{hf,i}$=68 °C, $T_{hf,o}$=45 °C, $\dot{m}_{hf}$=33.33 kg/s	(Cycle can not run) Hot fluid outlet temperature is very low, so to achieve 45°C boiler hot fluid (water) outlet temperature, evaporator pressure falls below 40°C saturation condenser temperature. If further condenser temperature is

				decreased then small amount of work can be obtained which serves no purpose. Hence, no cycle is possible at this condition.
8.	Moderator active process water heat exchanger of main moderator cooling system (when No-1 and No-2 both in use)	Modification not required	$T_{hf,i}$=47 °C, $\dot{m}_{hf}$=996 kg/s	(Cycle can not run) Hot fluid inlet temperature is very less to produce any work output from given condition because of evaporator temperature becomes equal to condenser temperature after applying PPTD value of heat exchanger. If further condenser temperature is decreased then small amount of work can be obtained which serves no purpose. Hence, no cycle is possible at this condition.
9.	Moderator active process water heat exchanger of main moderator cooling system (when No-1 and No-2 both in use)	Modification required	$T_{hf,i}$=75 °C, $T_{hf,o}$=50 °C, $\dot{m}_{hf}$=471 kg/s	**Case-2 (Cycle can run)** Cycle can run on the given condition to produce the work due to availability of hot fluid inlet temperature in sufficient amount. Also, hot fluid outlet temperature value is favorable for lower condenser temperature condition in which evaporator temperature can have enough value to produce work after considering PPTD. There will be no/very small amount of work output if condenser temperature is increased which serves no purpose. Cycle is possible at

				given condition but at lower condenser temperature (below general value of condenser).

The feasibility of using ORC for all nine locations is discussed in Table 5.2. Two potential locations (cases 1 and 2) are thermodynamically identified based on the feasibility study for direct application of simple ORC to generate extra power. A few systems and components have been identified that require modification but can not run on simple ORC. To run this system, further changes have to be implemented in the heat exchanger system to get the desired boiler pressure.

Isopentane, R245fa and R236ea are taken as working fluids and water is used as a hot and cold fluid. Input parameters are as follows: General Condenser temperature = 40°C, Ambient temperature = 29°C, Cold fluid inlet temperature = 29°C, PPTD of the plate type evaporator and condenser = 7°C [114,123], Isentropic reciprocating pump efficiency = 0.8 , Isentropic scroll turbine efficiency = 0.6 [124], Alternator mechanical efficiency= 0.96 [123], Electricity cost = 0.078 USD/kWh [125,126] and Annual running time = 7000 h.

5.2. Modelling and Simulation

The feasible location of waste heat recovery from the real operating NPPs data can be identified based on the thermodynamics (mass, momentum and energy equations) point of view. After identifying a feasible location and system's component, it is very necessary to check its viability from economic and environmental points of view. The required thermodynamic and design mathematical modelling equations, (2.1 - 2.27) are taken from chapter 2. Other than the above, additional equations and correlations are mentioned below.

5.2.1. Mathematical Modelling

Table 5.3 shows the market price of the alternators of different capacities and different manufacturers. This cost data, along with capacity, will be helpful in building a correlation for an alternator that can be used to convert mechanical energy into electrical energy.

Table 5.3. The market price of the alternator of different capacities and different manufacturer

Company (Manufacturer)	**Capacity (kVA)**	**Capacity (kW)**	**Market Price (INR)**	**Market Price (USD)**
Start Up Atop	5	4	22,800	274.5995

Prem	5	4	22,500	270.9864
MD Bharat	5	4	15,500	186.6795
Bharat Ac Alternator	7.5	6	23,500	283.0302
Tech Power	7.5	6	27,500	331.2056
Sunfield	10	8	27,000	325.1837
Kirloskar	10	8	22,500	270.9864
Ashok	15	12	44,000	529.9289
Gravis	20	16	49,500	596.1701
Power Gold	25	20	42,000	505.8413
Electron	25	20	42,000	505.8413
Start Up Atop	25	20	48,500	584.1262
Kishanmaster	25	20	42,000	505.8413
Mahima Enterprise	25	20	44,000	529.9289
Ms Power	35	28	56,000	674.455
Crompton	35	28	58,000	698.5427
Chandra Agriculture Industry	40	32	58,000	698.5427
Mahima Enterprise	50	40	60,000	722.6304

A trendline analysis of the 3-phase alternator's capital cost correlation has been done with capacity (kW) and cost (USD), as shown in Figure 5.1. The general value of the power factor is taken as 0.8 to convert alternator capacity kVA to kW. The R^2 value is accepted as 0.9085 because of the market price difference for the same capacity and different manufacturers.

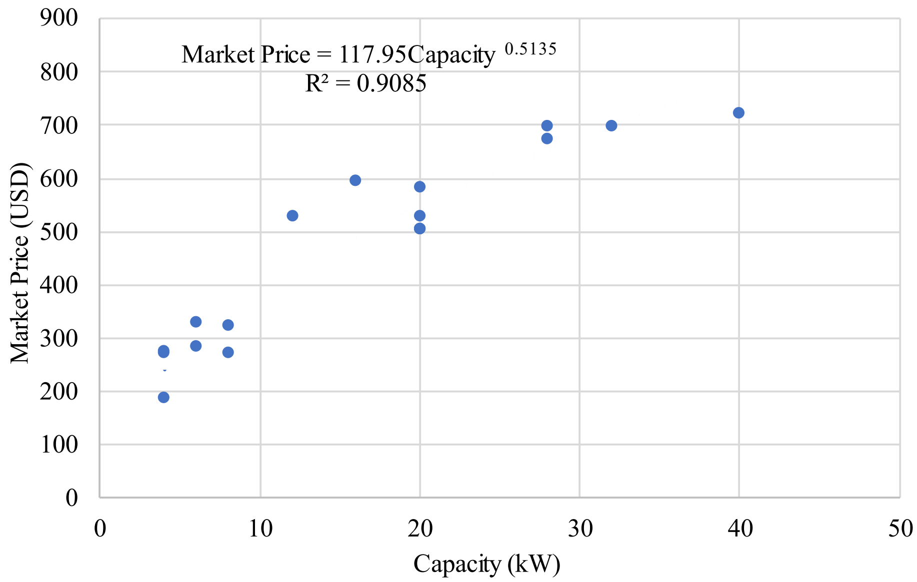

Figure 5.1. Alternator's capital cost correlation by trendline analysis between capacity (kW) and market price (USD)

Alternator's capital cost correlation is deduced from Figure 5.1 as:

Capital investment (USD),

$$Z_{alt}^{CI} = 117.95(W_{net})^{0.5135} \tag{5.1}$$

5.2.2. Simulation and Validation

While simulating the ORC systems to identify and evaluate the possible location to generate power, input and ambient thermodynamic parameters are chosen based on the problem defined and NPPs data (Table 5.1). PPTD is defined effectively when designing evaporator and condenser [114]. The simulation is done using EES software [127]. All physical and thermodynamic properties of the different fluids are calculated using the REFPROP function built into EES. DPP, LCOE, NPV, and IRR are calculated using the formula given on the spreadsheet. The heat exchanger area is calculated by Muley's and Yan-Lin's correlations [102] to determine the capital investment. Figure 5.2 represents the flow chart followed in the simulation process. Validation is the same as discussed in chapter 2.

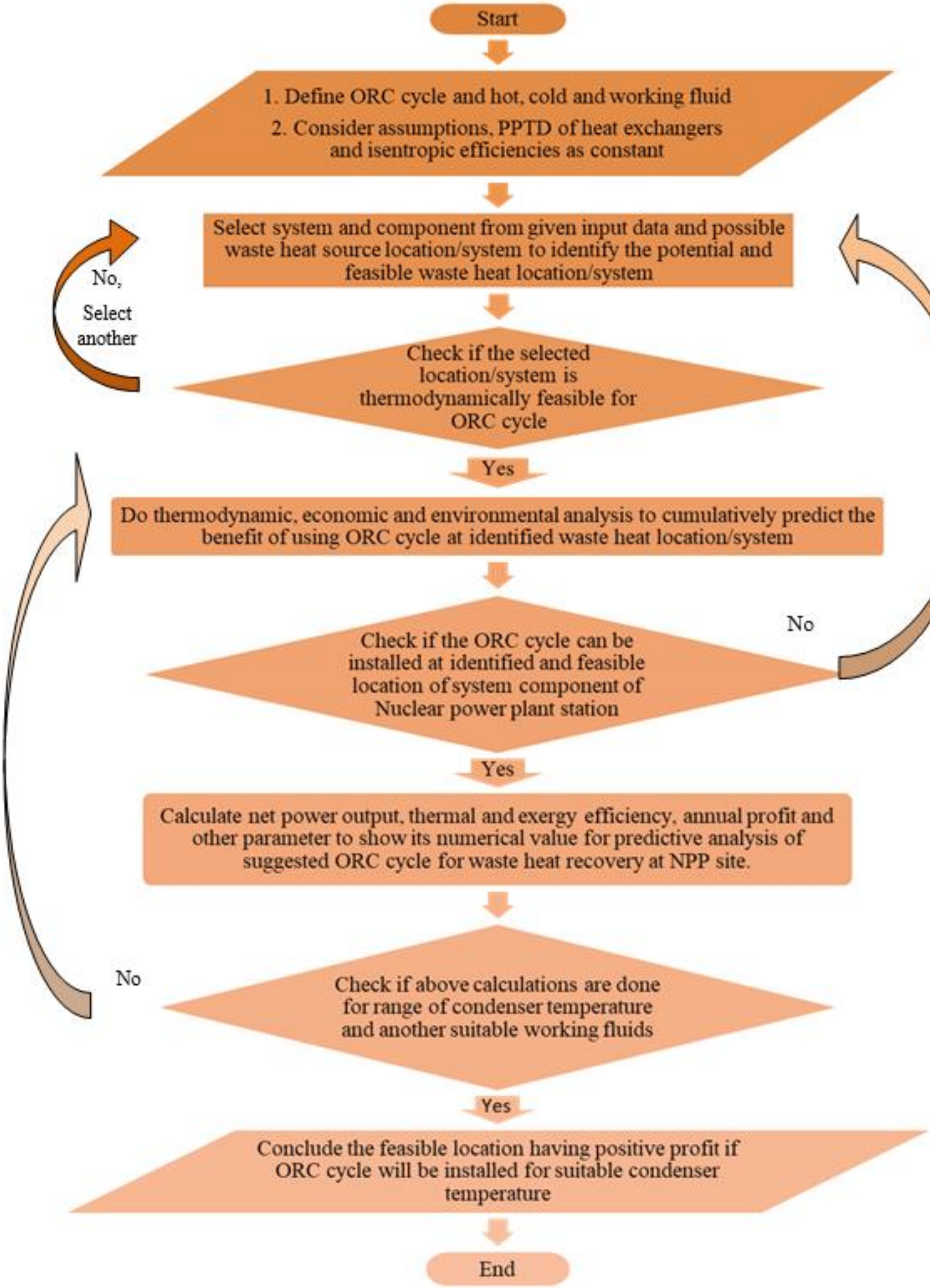

Figure 5.2. Flowchart of simulation process for feasible location in NPP to install ORC

5.3. Result and Discussion

5.3.1. Comparison of two feasible cases

The two waste heat source locations are identified by a feasibility test on EES (Engineering equation solver) for ORC applicability based on the data and drawings provided by BRNS for the nuclear power plant's waste heat recovery installation. The above 2 cases are suggested for ORC installation and recommended for the nuclear power plant's waste heat recovery, based on the requirement. The above two cases are selected without altering the existing system and components, as well as for thermal performance and cost-to-profit analysis. These two cases will be recommended for the upgrading of existing nuclear power plants to generate more electricity with the consideration of profit and the environment. Figure 5.3 shows the schematic diagram of the suggested component to be installed at NPP waste heat recovery site for further electricity generation. To upgrade the existing nuclear power plant, the following equipment is mentioned in Table 5.4, are designed and recommended for upgradation purposes to harness waste heat into electricity generation.

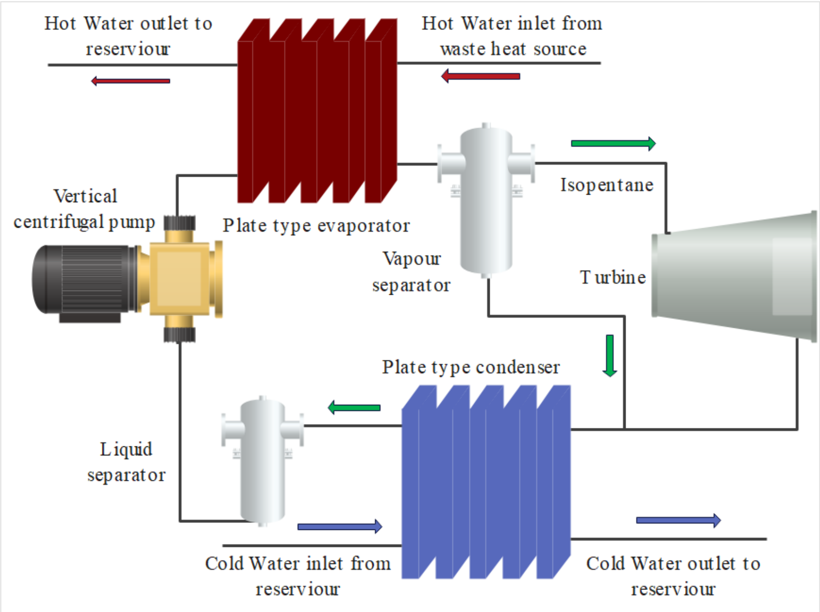

Figure 5.3. Schematic diagram of suggested ORC to be installed at NPP waste heat recovery site

Table 5.4. Recommended component design and specification for ORC installation

Sr. No.	Equipment	Type	Case-1: Design and Cost Specification	Case-2: Design and Cost Specification

1.	Pump	Vertical centrifugal pump	Capacity: 4.5 kW, Liquid flow rate: 1,128 litre/min, Cost: ₹ 7,94,724	Capacity: 6.1 kW, Liquid flow rate: 14,587 litre/min, Cost: ₹ 9,98,592
2.	Turbine	Case 1: Radial (Centrifugal), Case 2: Axial	Capacity: 199 kW, Vapour flow rate: 73,922 litre/min, Cost: ₹ 1,90,13,904	Capacity: 374 kW, Vapour flow rate: 1,760,566, litre/min, Cost: ₹ 2,96,45,112
3.	Evaporator	Plate heat exchanger	Heat transfer Area: $21m^2$, Design pressure: 3.21 bar, Capacity: 4,194 kW, Cost: ₹ 7,45,584	Heat transfer Area: $136.1m^2$, Design pressure: 1.7 bar, Capacity: 49,262 kW, Cost: ₹ 32,15,856
4.	Condenser	Plate heat exchanger	Heat transfer Area: $32m^2$, Design pressure: 1.52 bar, Capacity: 4,024 kW, Cost: ₹ 10,25,304	Heat transfer Area: $232.3m^2$, Design pressure: 1.52 bar, Capacity: 48,954 kW, Cost: ₹ 48,78,468
5.	Liquid vapour separator	Vertical cylinder	Separation velocity: 1.2 m/s, Capacity: 0.34 m^3, Cost: ₹ 1,87,120	Separation velocity: 0.97 m/s, Capacity: 10.12 m^3, Cost: ₹ 10,43,645

Table 5.5. Various input and output parameters for case 1 and case 2

Output Parameter	**Case -1**	**Case -2**
Net power (work) [kW]	167.8	318.4
Thermal efficiency (%)	4	0.65
Pump work [kW]	4	5.5
Turbine work [kW]	178.8	337.2
Condenser pressure [kPa]	151.3	151.3
Boiler pressure [kPa]	321.1	169.4
Exergy efficiency (%)	26.5	6.5
Exergy in [kW]	633.8	4896

Total irreversibility [kW]	459	4564
Condenser cold fluid (water) mass flow rate [kg/s]	224.1	2898
Working fluid mass flow rate (cold side of boiler and hot side of condenser) [kg/s]	11.3	145.7
*Boiler hot fluid (water) mass flow rate [kg/s]	33.3	471
Heat inlet from boiler [kW]	4199	49286
*Hot fluid (water) inlet temperature [°C]	98	75
*Hot fluid (water) outlet temperature [°C]	68	50
Cold fluid (water) inlet temperature [°C]	29	29
Cold fluid (water) outlet temperature [°C]	33.3	33
Condenser area [m^2]	31.5	232.3
Evaporator area [m^2]	21	136.1
Capital investment Cost [in thousand USD]	259.42	464.32
NPV of total cost (CI and O&M for 40 years) [in thousand USD]	472.26	845.27
NPV of total energy output (for 40 years) [GWh]	12.86	24.41
Total capital and O&M cost [USD/h]	3.29	5.89
Total revenue earned [USD/h]	13.09	24.83
Yearly profit (in thousand USD)	68.56	132.58
Payback period [years]	6.15	5.86
Discounted payback period [years]	8.67	8.08
LCOE (USD/kWh)	0.037	0.035
Internal rate of return (%)	17	17
Annually saving in CO_2 emission from NPP [ton]	21.14	40.12

*Selected input parameters of the feasible location of NPP for waste heat recovery

For fixed operating conditions (Table 5.2), all necessary input and output parameters are tabulated in Table 5.5 for both cases. Comparison is made easy from thermal and economic points of view. For the analysis, two feasible cases (case-1 and case-2) are deduced. Case-2 produces maximum power because of the availability of a high hot fluid mass flow rate in the cycle, but due to the availability of waste heat at a lower grade, it has the lowest efficiency. Case-1 has the highest thermal efficiency because it operates at a higher boiler temperature when compared to case-2. LCOE obtained a value of 0.038 USD /kWh for case-1 and 0.036 USD /kWh for case-2, which is lower than the electricity selling cost (0.061 USD/kWh) from power generated from Indian NPPs. So, for both cases, ORC upgradation is worth venturing into the current NPP site. It is worth noting that ORC's built-up cost in India is calculated as 1.20-1.51 thousand USD/kWe, which is the lowest among South Korea, China, Japan, Russia, and many more countries [28]. It is clear that ORC installation cost for the upgradation of the current NPP site for case -1 is 1.46 thousand USD/kWe and for case-2 is 1.38 thousand USD/kWe, which is lower and in the range of India's expected per kWe expense.

T-s diagram is shown in Figure 5.4 for both cases discussed in Table 5.5. It is clearly shown that case-1 has the maximum turbine enthalpy difference, but power output is maximum for case-2 because of the high mass flow rate of working fluid (Table 5.5). Figure 5.4 also shows the exact temperature and specific entropy node value at different stages of the ORC power plant for both cases.

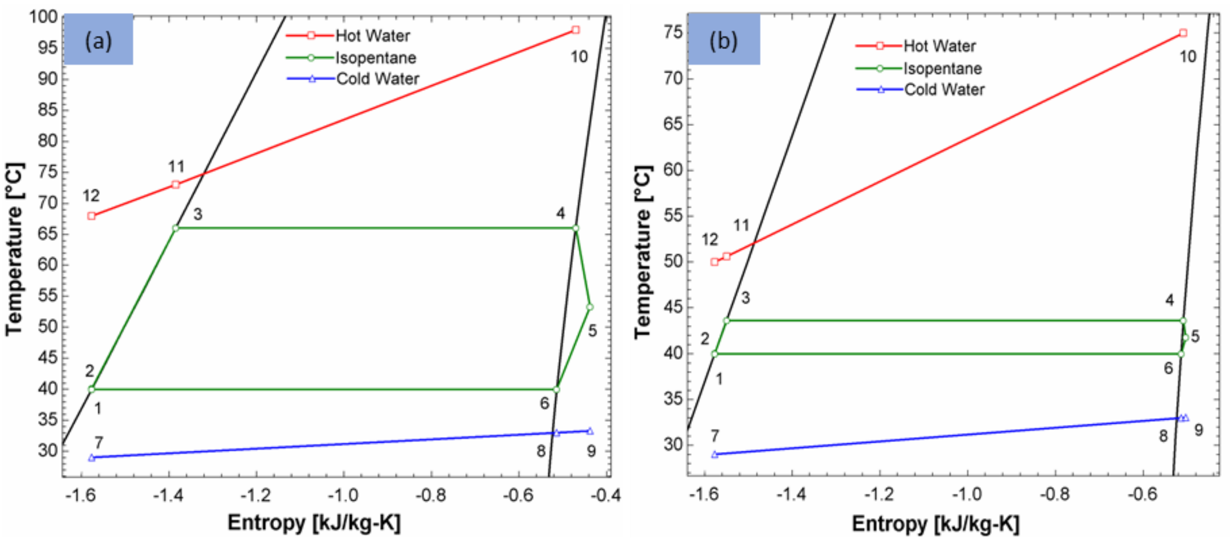

Figure 5.4. (a) ORC T-s diagram of case-1, (b) ORC T-s diagram of case-2

5.3.2. Working fluid behaviour on the output parameter

Table 5.6 shows the working fluid behaviour for different output parameters for both feasible cycles. Results show that R245fa performed better for case-1 and R236ea performed better for case-2 in terms of yearly profit when compared with R601a (Isopentane) working fluid. Other necessary parameters are compared and tabulated in Table 5.6.

Table 5.6. Effect on output parameter with working fluid for both feasible cases

Output Parameter	Case -1				Case -2			
	R601a	R245fa	R236ea	R123	R601a	R245fa	R236ea	R123
Net power (work) [kW]	167.8	169.5	169.5	168.1	318.4	321.7	329.8	314.8
Thermal efficiency (%)	4.01	4.03	4.03	4.00	0.65	0.65	0.67	0.64
Boiler pressure [kPa]	321.1	554.9	737.3	334.9	169.4	281.4	380.4	173.7
Exergy efficiency (%)	26.50	26.74	26.74	26.53	6.5	6.57	6.74	6.43
Capital investment [in thousand USD]	259.42	267.21	275.10	259.53	464.32	473.83	489.86	460.83
Yearly profit (in thousand USD)	68.56	68.79	68.07	68.74	132.58	133.54	136.55	130.92
NPV of the total cost [in thousand USD]	472.26	486.43	500.80	472.45	845.27	862.57	891.75	838.90
Discounted payback period [years]	8.67	8.87	9.19	8.65	8.08	8.18	8.26	8.12
LCOE (USD/kWh)	0.037	0.037	0.039	0.037	0.034	0.035	0.035	0.034

5.3.3. Parametric study

The net work output of the cycle will be increased when condenser temperature and cold fluid inlet temperature are lowered, as shown in Figure 5.5. Amount of work obtained for different cases will vary with evaporator and condenser pressure that is directly affected by waste heat source input temperature and condenser temperature. From Figure 5.5, it can be observed that case-1 cycle work production capacity does not much changes with condenser temperature as it drastically changes with case-2 due to the amount of working fluid mass flow rate. Case-2 produces maximum power when both condenser temperature and cold fluid inlet temperature are lowered because of the large amount of working fluid mass flow rate and turbine enthalpy difference. All graphs have some missing values because PPTD criteria cannot be met at some condenser temperature and cold fluid inlet temperature values.

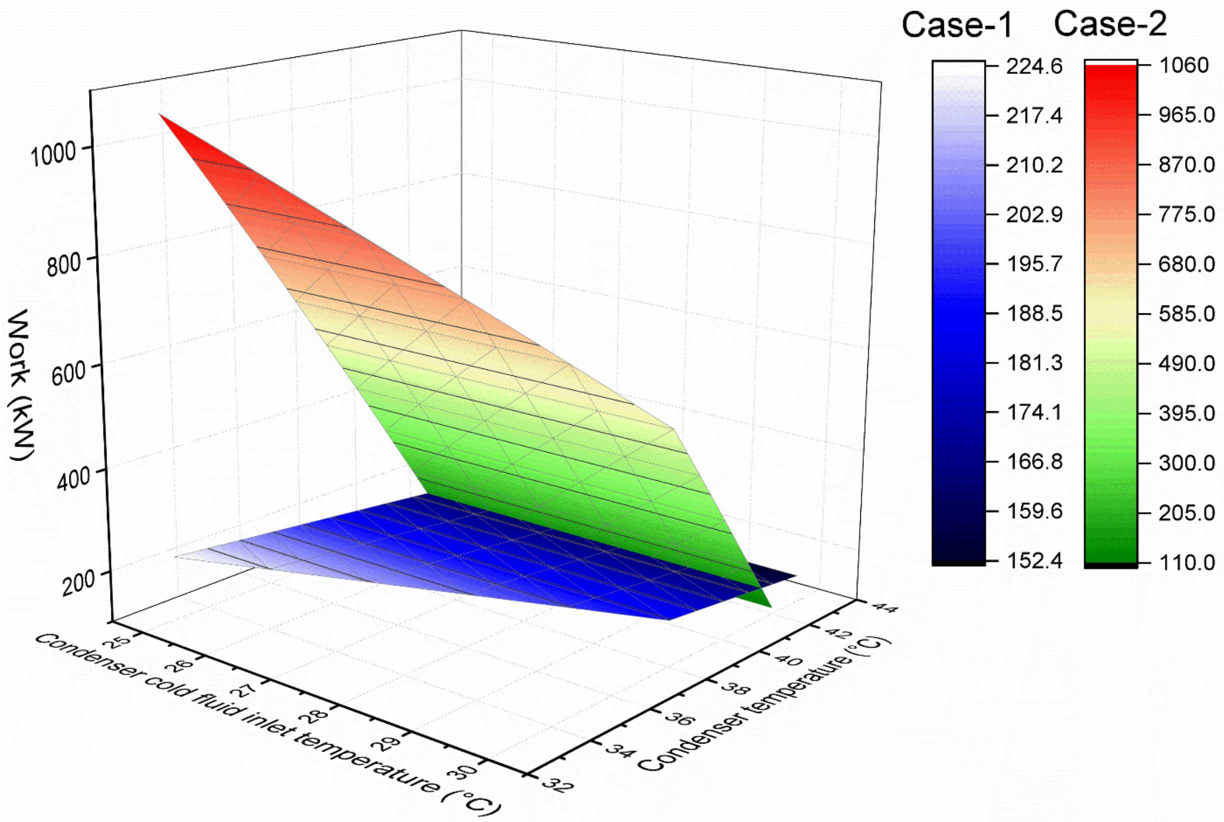

Figure 5.5. Effect of condenser and cold fluid inlet temperatures on net work output

Thermal efficiency is dependent on the type of working fluid used and the pressure ratio of the power cycle. The temperature grade of the heat source plays a significant effect on thermal efficiency. High grade of evaporator /waste heat source temperature gives higher thermal efficiency. Case-1 cycle has maximum thermal efficiency because of its high-pressure ratio of evaporator and condenser when compared to case-2, as shown in Figure 5.6. Both cases show the same decremental trend in thermal efficiency when the condenser temperature increases. Thermal efficiency is not significantly affected by the variation in the condenser's cold fluid inlet temperature.

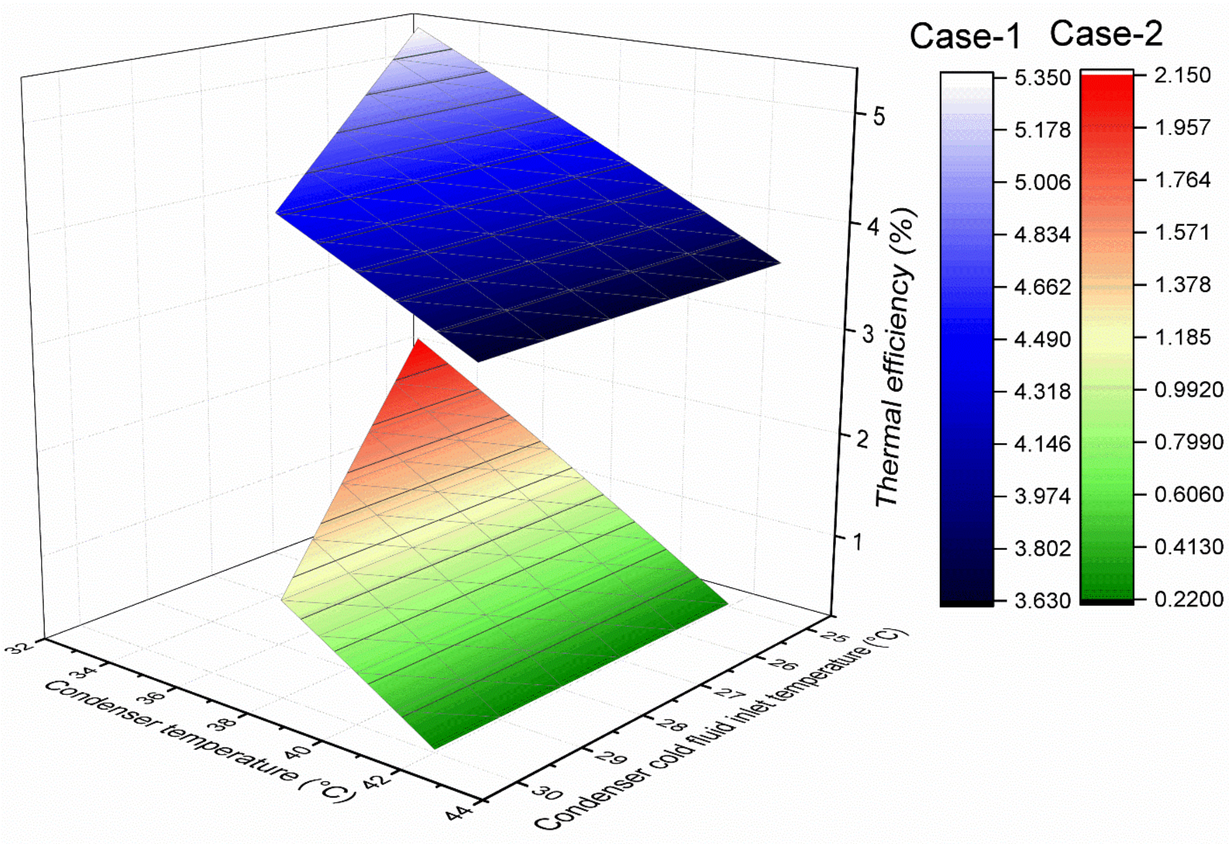

Figure 5.6. Effect of condenser and cold fluid inlet temperatures on thermal efficiency

The total investment cost is affected by the sum of individual components' capital cost and their maintenance cost up to the project lifespan, which is further dependent on the intensive property (pressure and temperature) and extensive property (mass flow rate) of the fluid. Capital cost is directly influenced by material and manufacturing costs, whereas operation and maintenance costs are dependent on the raw material used (fuel) and labor expenses for successive years. In successive years, labor and other expenses will not be the same and the time value of money also comes into the picture when calculating project feasibility at present time. To withstand the component at higher pressure and temperature, thick material is used and to handle more mass flow rate, a large size component is used. Figure 5.7 shows variations in total capital investment and O&M cost (operation and maintenance cost) for the 20-year lifespan of both cases, with variations in condenser temperature and cold fluid inlet temperature. If condenser temperature and cold fluid inlet temperature are lowered, then both cycles will produce higher work output at a larger pressure ratio; thus, net present value (NPV) of total investment becomes maximum for both cases, Figure 5.7. NPV of total investment is larger for case-2 due to big size component requires for higher mass flow rates. Due to the PPTD design constraint of condenser temperature, the difference of cold fluid inlet and outlet temperature will be less and so, more amount of cold fluid mass flow rate will be required for the same amount of heat rejection. At lower condenser temperatures, condenser size will be increased for the same mass flow rate due to lower LMTD.

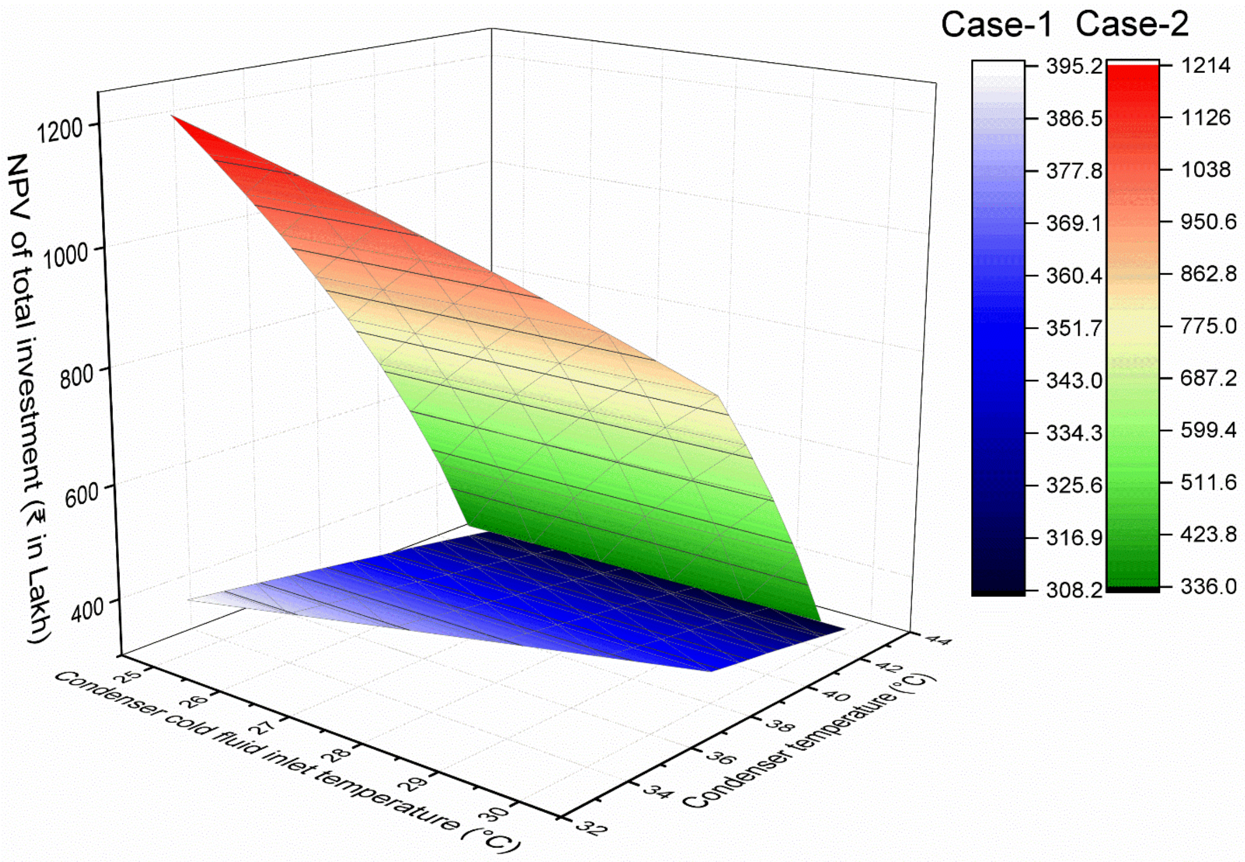

Figure 5.7. Effect of condenser and cold fluid inlet temperatures on NPV of total investment

Although case 2 has the highest capital investment and NPV of total investment, it also possesses the highest net annual profit at the same input condition due to the maximum production of the net work. Figure 5.8 shows that the annual profit increment trend with condenser temperature and cold fluid inlet temperature is the same as the work production trend because of its relative dependencies on work output and capital investment. Case-2 is not favourable concerning case-1 at a condenser temperature higher than 41°C. Although case-2 has a lower pressure ratio and thermal efficiency because of increased mass flow rate, it gives the best profit for a condenser temperature below 41°C.

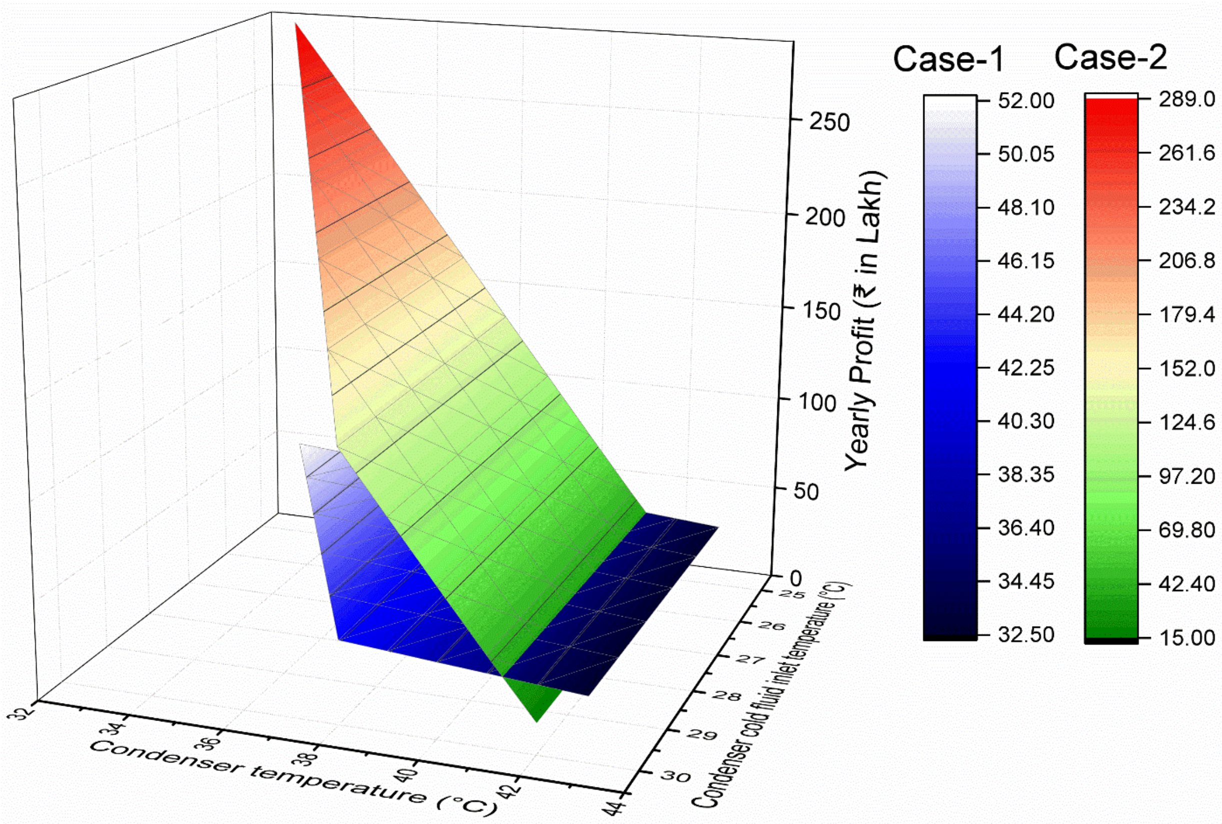

Figure 5.8. Effect of condenser and cold fluid inlet temperatures on yearly profit

The lower value of LCOE is favorable for the selection of power plant projects. Figure 5.9 shows the effect of condenser temperature and cold fluid inlet temperature on LCOE for different case cycles. Figure 5.9 clearly shows that case-1 will be worth venturing at above 41°C condenser temperature and case-2 will be worth venturing at below 41°C condenser temperature. LCOE value is quite low, thus favorable for the case-2 cycle when operating below 41°C condenser temperature. At a high value of LCOE, the discounted payback period may be greater than the project lifespan, which means the project will surely be in loss-making.

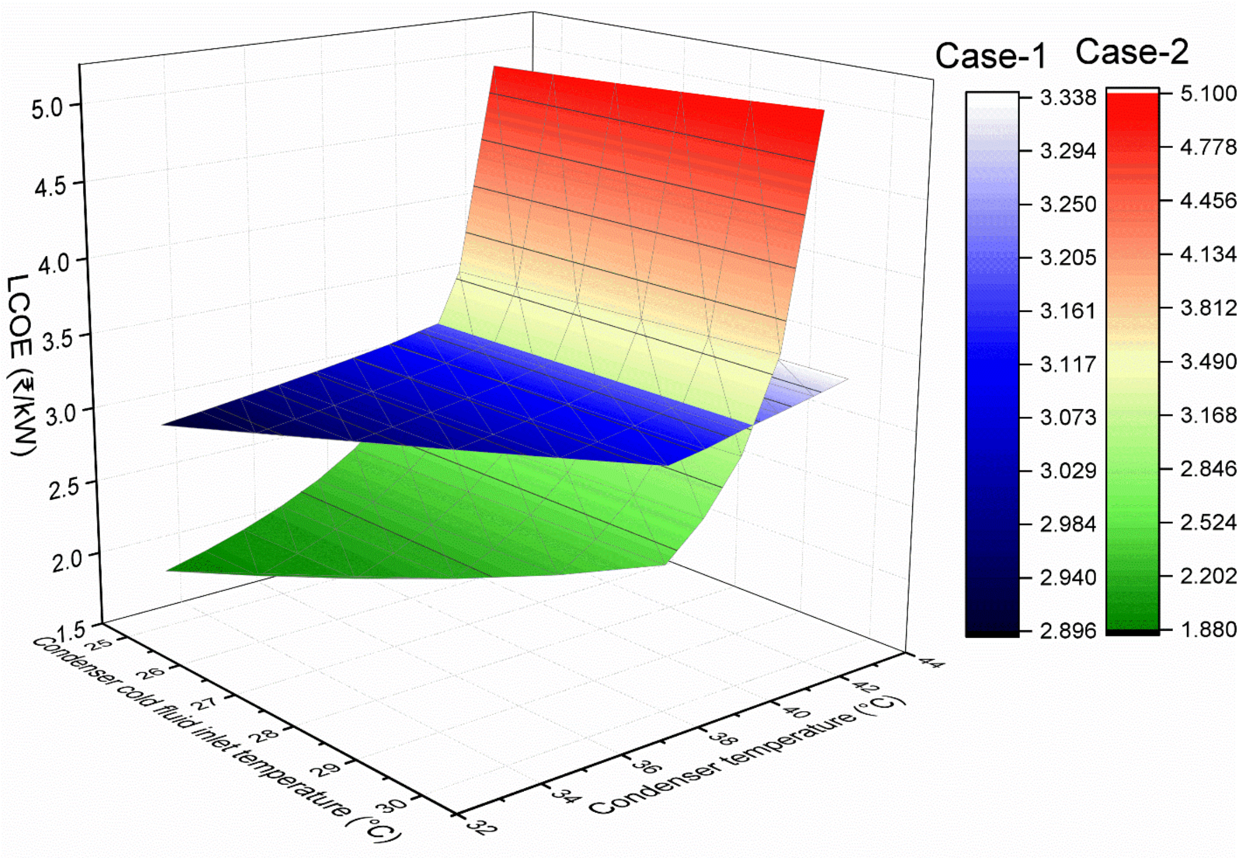

Figure 5.9. Effect of condenser and cold fluid inlet temperatures on LCOE

Discounted payback period reveals how fast invested money can be paid back when the time value of money is considered. Lower DPP is favorable for identifying potentially benefited projects. Figure 5.10 shows that case-2 is favorable for condenser temperature operating below 41°C. Figure 5.10 has some missing values of case-2 for condenser temperature operating above 41°C because, at that temperature intersection, DPP will be greater than the estimated life span of the project (40 years).

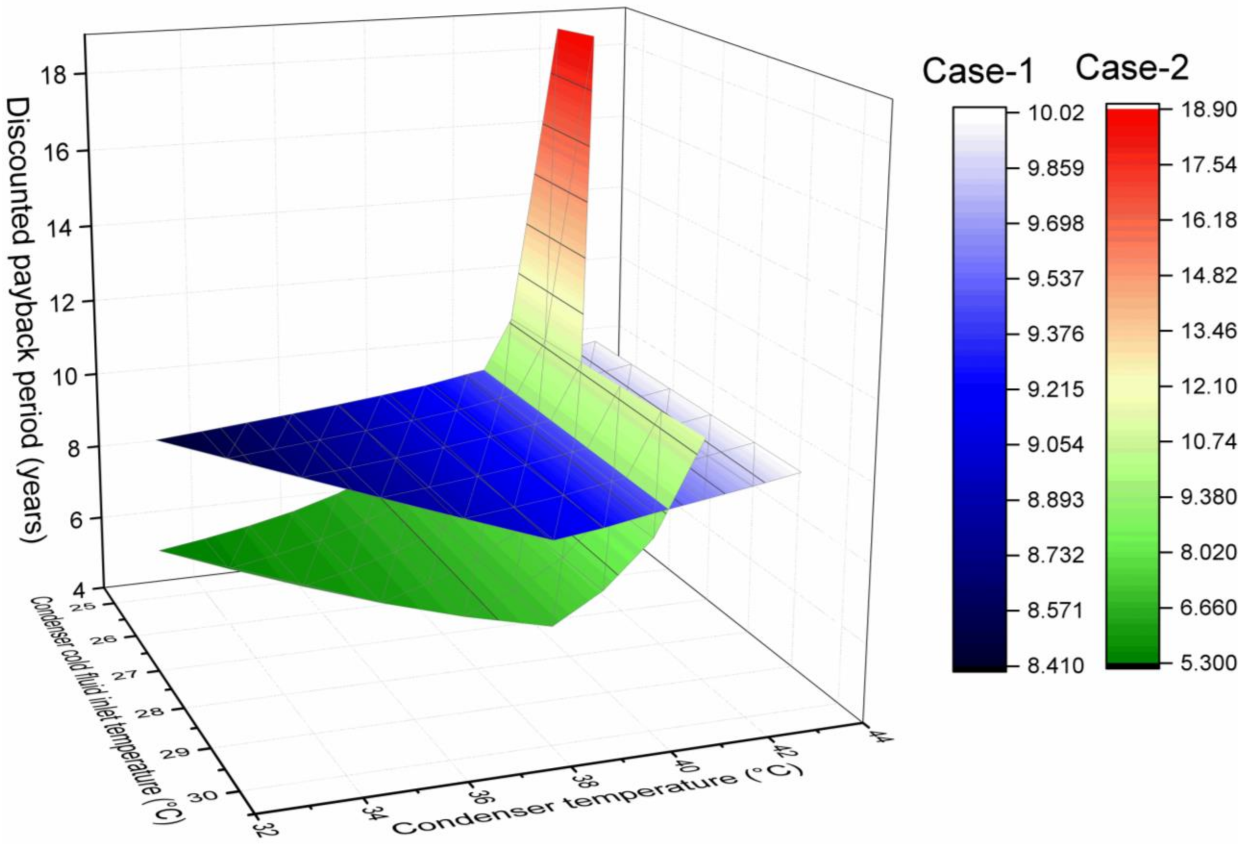

Figure 5.10. Effect of condenser and cold fluid inlet temperatures on DPP

To decide whether the project should be taken or not, internal rate of return (IRR) plays an important role in decision-making. If IRR> discounted rate (8%), then the project can be profitable. The decision is further dependent on IRR value and other parameters also (LCOE, DPP). It is clear from Figure 5.11 that case-1 would be feasible to opt at a condenser temperature less than 43°C and case-2 would be feasible to opt at a condenser temperature less than 42°C. The higher value of IRR is favorable when choosing between two projects. It is clearly shown in Figure 5.11 that the project below 41°C condenser temperature case-2 and above 41°C condenser temperature case-1 is worth venturing for all ranges of condenser cold fluid inlet temperature.

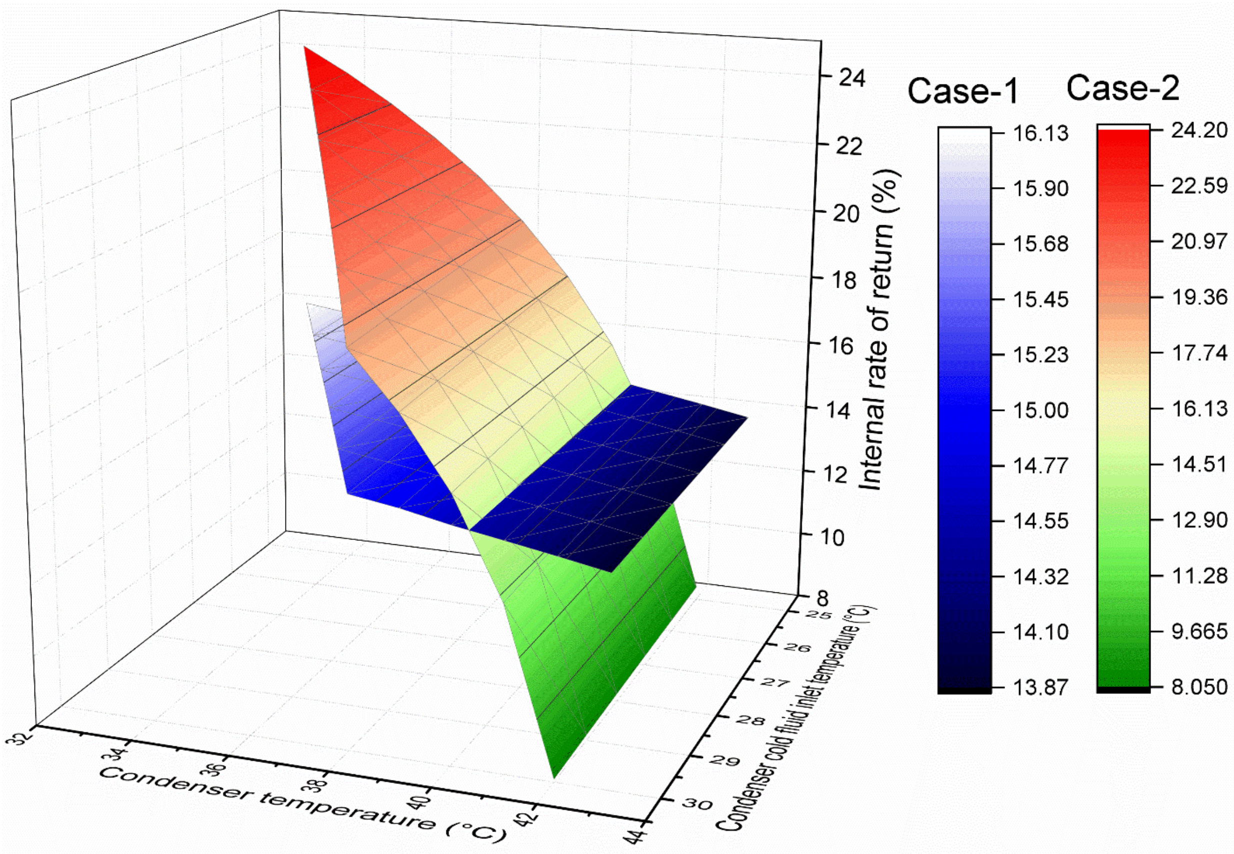

Figure 5.11. Effect of condenser and cold fluid inlet temperature on IRR for different case cycles

Although CO_2 saving from NPP is much lower than that of conventional power plants (coal-based power plants), it is worth mentioning when compared with renewable-based power plants (solar, hydro, wind). Figure 5.12 shows the effect of condenser temperature and cold fluid inlet temperature on annual CO_2 saving for both case cycles. It is clearly shown in Figure 5.12 that the project below 42.5°C condenser temperature case-2 and above 42.5°C condenser temperature case-1 will save the maximum amount of CO_2 emission for all ranges of condenser cold fluid inlet temperature. This CO_2 emission comes into the picture when the same amount of additional power generated is achieved from ORC upgradation at the NPP site and not from the NPP itself.

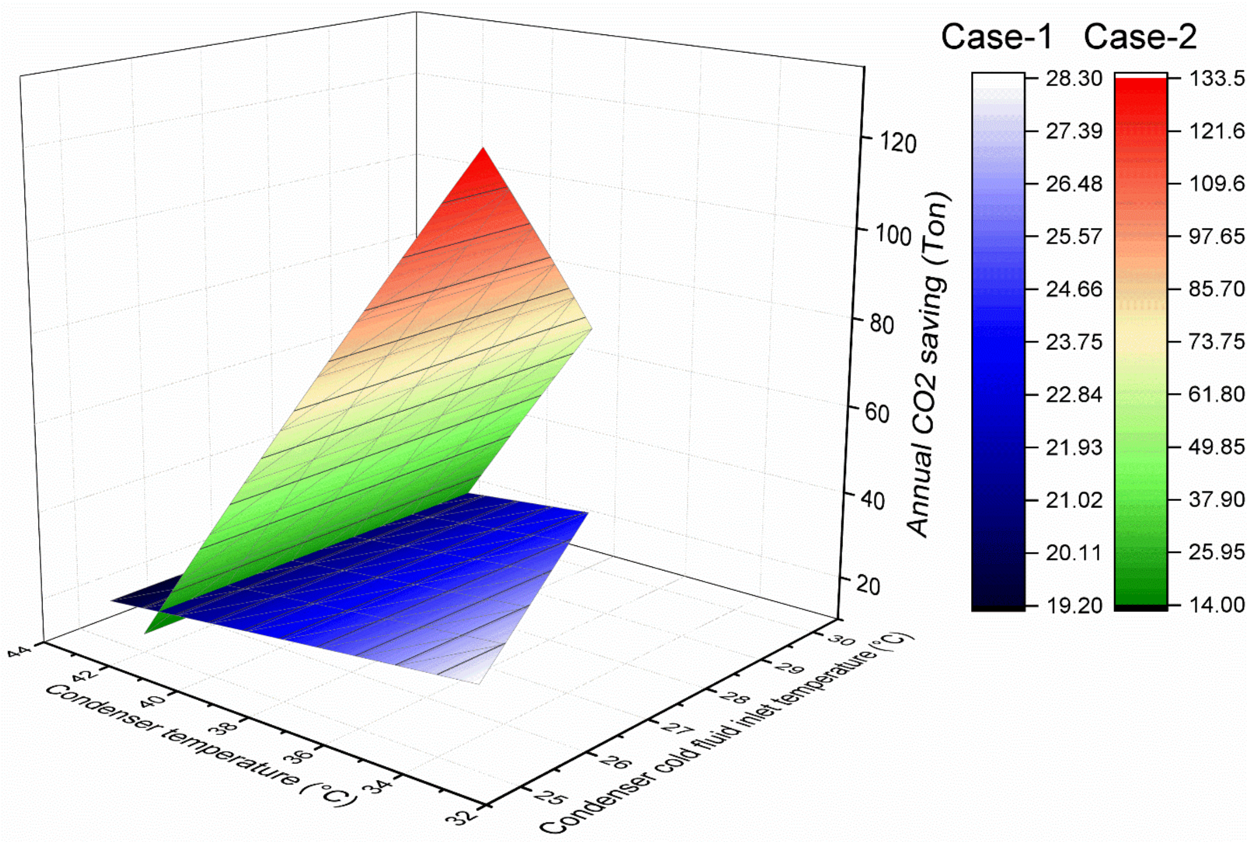

Figure 5.12. Effect of the condenser and cold fluid inlet temperature on annual CO_2 emission saving

5.3.4. Simulation vs Experimental

The simulation and experimental variation of data provided by BRNS for NPP upgradation and feasibility studies are tabulated in Table 5.7 for case-1 and Table 5.8 for case-2. For simulation conditions, cold fluid is taken as water, while for experimental purposes, the air is taken as cold fluid to avoid further complexities of water supply, drainage and leak. The hot fluid inlet temperature was matched with simulation data (data provided by BRNS, Table 5.1) by adjusting the heating load for both cases. The mean condition of the volume flow rate was taken, and the mass flow rate value for therminol VP1 was obtained.

Table 5.7. Simulation and experimental variation of Case-1 NPP

Case-1		
Parameter	**Simulation**	**Experimental**
***Hot fluid Inlet Temperature [°C]**	98.00	98.72
***Cold fluid inlet temperature [°C]**	29.00 (Air)	28.82 (Air)

Hot fluid Outlet Temperature [°C]	68.00	96.10
***Mass flow rate of hot fluid [kg/s]**	33.33 (Water)	0.833 (Therminol VP1)
Mass flow rate of working fluid [kg/s]	11.26 (Isopentane)	0.0052 (Isopentane)
***Heat Input [kW]**	4199	5
Net Work [kW]	174.8	0.106
Thermal efficiency (%)	4.2	2.13
Pump inlet pressure [bar]	1.51	1.60
Pump outlet pressure [bar]	3.21	4.12
Turbine inlet pressure [bar]	3.21	3.76
Turbine outlet pressure [bar]	1.51	1.89
Pump inlet temperature [°C]	40.00	30.92
Pump outlet temperature [°C]	40.10	31.19
Turbine inlet temperature [°C]	66.08	87.45
Turbine outlet temperature [°C]	53.32	74.61

*Selected input parameters

Table 5.8. Simulation and experimental variation of Case-2 NPP

Case-2		
Parameter	**Simulation**	**Experimental (Threshold condition)**
***Hot fluid Inlet Temperature [°C]**	75	90.7**
***Cold fluid inlet temperature [°C]**	29 (Air)	29.32 (Air)
Hot fluid Outlet Temperature [°C]	50	88.88
***Mass flow rate of hot fluid [kg/s]**	471 (Water)	1.003 (Therminol VP1)
Mass flow rate of working fluid [kg/s]	145.6 (Isopentane)	0.0050 (Isopentane)
***Heat Input [kW]**	49286	4.0
Net Work [kW]	318.4	0.09449 (94.49 W)
Thermal efficiency (%)	0.65	2.34
Pump inlet pressure [bar]	1.51	1.63
Pump outlet pressure [bar]	1.69	3.68

Turbine inlet pressure [bar]	1.69	3.35
Turbine outlet pressure [bar]	1.51	1.88
Pump inlet temperature [°C]	40.0	30.8
Pump outlet temperature [°C]	40.01	31.01
Turbine inlet temperature [°C]	43.63	81.67
Turbine outlet temperature [°C]	41.79	71.9

*Selected input parameters

**Current fabricated and installed ORC experimental setup in HMT lab of IIT BHU has design constraint (Evaporator undersize) due to which turbine movement with desired RPM is only possible at above 90°C heat source inlet temperature condition. If evaporator size is increased then it can be possible to run the experimental setup at 75°C heat source inlet temperature condition. Moreover, operating condition (Fixed hot fluid outlet temperature) is also mazor constraint to get more power at higher thermal (cycle) efficiency.

Table 5.7 and Table 5.8 show the input and output condition of both selected feasible cases and based on that, scalability scope can be identified to get installed at the site location for further generation of electricity. If desired hot fluid outlet temperature (for case-1, 68°C and for case-2, 50°C) can be maintained separately after the ORC evaporator, then by optimizing at given condition, more turbine power and cycle efficiency can be obtained.

5.3.5. Nanofluid-enhanced WHR for ORC

Waste heat-driven ORC using nanofluid as a recovery medium is taken as a case study. ORC can convert low-to-medium-temperature waste heat into useful work. Incorporating nanofluids as the hot fluid in the ORC enhances heat transfer efficiency, potentially boosting the cycle's overall performance, as shown in Figure 5.13. The studied system includes the following key components: evaporator, organic working fluid, nanofluid, turbine, condenser, and pump. The nanofluid behaviour on ORC affects the performance parameter. Hot fluids such as water and therminol VP1 are taken for comparison purposes with Al_2O_3 nanoparticles. This hot side nanofluid behaviour is analyzed with six organic fluids. The following table shows the properties of nanofluid and organic fluids used.

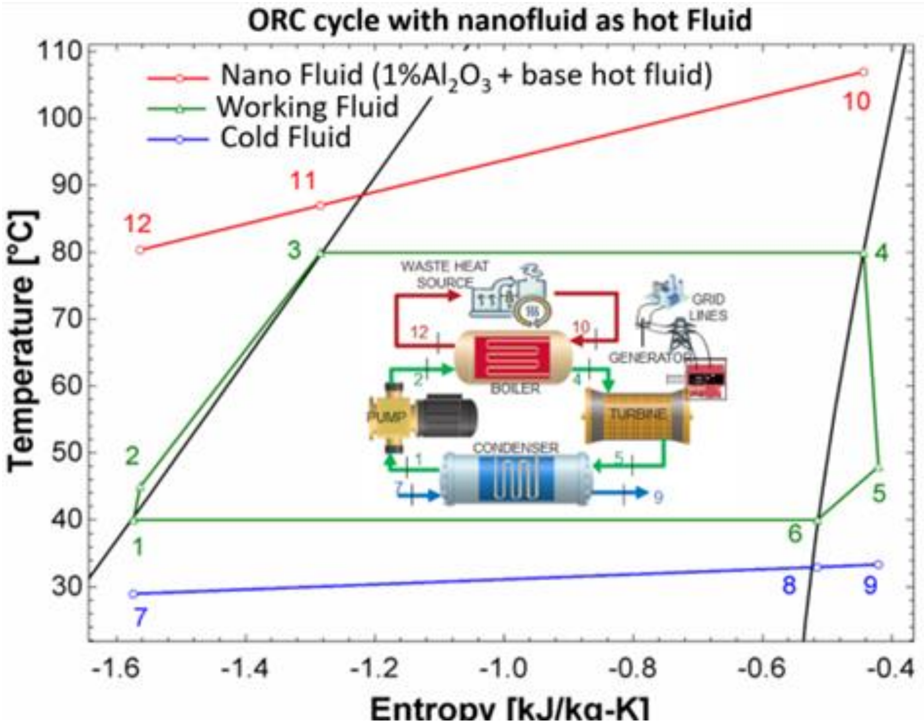

Figure 5.13. T-s Diagram and layout of nanofluid based ORC

The simulation code based on the above model has been developed on the EES platform. The input parameters are listed in Table 5.1. These feasible input parameters (Table 5.2) help in the simulation for performance comparison of waste heat recovery ORC for various nanofluid and organic working fluid combinations. The properties of used nanofluids (Al_2O_3/water and Al_2O_3/Thermonol-VP1) have been calculated using suitable correlations [128] based on individual data given in Table 5.10. Table 5.11 shows the properties of organic working fluids.

Table 5.9. Input parameters used

Input parameter	Value	Input parameter	Value
Alternator efficiency	0.96	Plate heat exchanger's length	150 mm
Cold fluid inlet	29°C	Dimension of storage tank	30 cm × 70 cm
Condenser temperature	40°C	Operational year	40 year
Hot (Nanofluid) inlet	75°C	PPTD	7°C
Hot (Nanofluid) outlet	50°C	Electricity cost	0.078 $/kWh
Nanofluid mass flow rate	471 kg/s	Yearly working hour	7000 h

Nanoparticle volume percentage	1%	Turbine efficiency	0.6
Pump efficiency	0.8		

Table 5.10. Thermophysical properties relevant to nanofluids [99,128]

Thermophysical property	Water	Therminol VP1	Al_2O_3
Density (kg/m^3)	998.2	1059	3970
Dynamic viscosity (Pa.s)	0.001003	0.003599	-
Specific heat (J/kgK)	4182	1565	765
Thermal conductivity (W/mK)	0.6	0.1357	40

Table 5.11. Organic working fluid properties

Working fluid	Isopentane	R123	R236fa	R236ea	R141b	R143m
Global warming potential	4	77	9,810	1370	725	4800
Normal boiling point value [°C]	27.85	27.78	-1.492	6.105	32.06	-23.57
Critical temperature [°C]	187.2	183.7	124.9	139.3	204.2	104.8
Ozone depletion potential	0	0.02	0	0	0.12	0
Critical pressure [bar]	33.7	36.68	32	34.29	42.49	36.35

Figure 5.14 compares the net work obtained with different hot fluids and working fluids viz. R143m, R141b, R236ea, R236fa, R123 and R601a. Water-based nanofluid and water show greater value for net work because of greater specific heat value, same temperature difference and hot fluid mass flow rate, which ultimately requires a very large amount of heat (almost 250% higher) to be used to operate the turbine and thus large turbine work is obtained. It has been clearly shown in the figure that net work output performance for nanofluid is lower than respective base fluid (hot fluid) for all organic working fluids because nanofluid lowers the specific heat value for nanofluid, which ultimately lowers the amount of heat transfer to the

boiler for same temperature difference (input condition constraint). Further, a lower amount of heat interaction requires less organic working fluid mass flow rate, which results in lower turbine work being obtained. Generally nanofluid application requires redesigning of the heat exchanger and PPTD go down for nanofluid applied boiler. From the organic working fluids, R236fa showed the best performance while R141b showed the lowest.

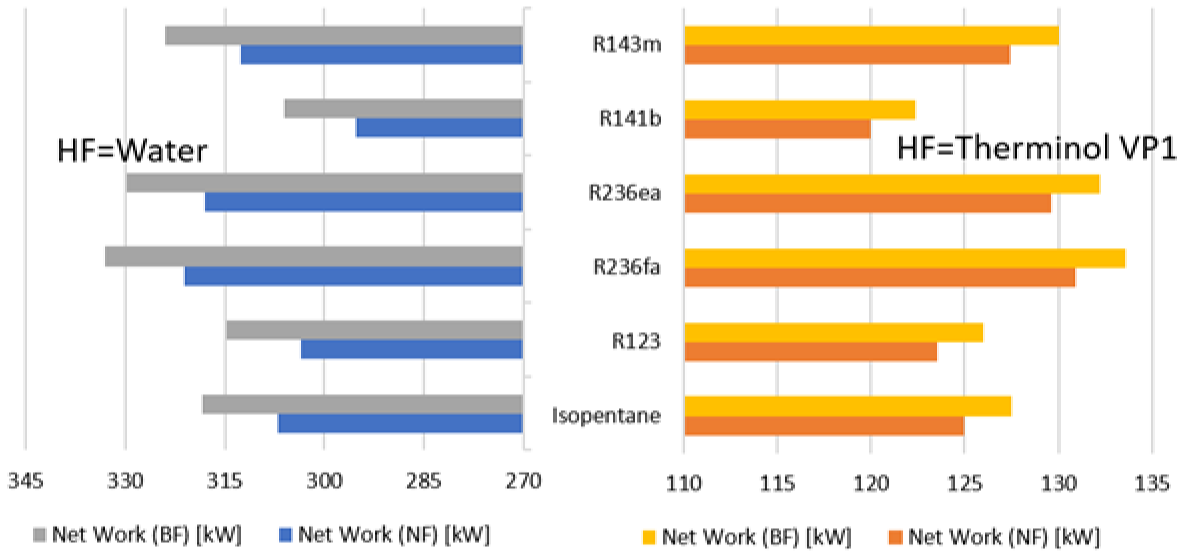

Figure 5.14. Net work comparison with different hot fluids and working fluids

Figure 5.15 compares the thermal efficiency obtained using both base hot fluids and nanofluids. Thermal efficiency depends upon net work and boiler's heat interaction. The study found that water and therminol-VP1 showed almost the same thermal efficiency with little increment with therminol-VP1 base fluid and nanofluid. Interaction of therminol-VP1-based nanofluid showed almost same or little increase in thermal efficiency with its base fluid, while water and its nanofluid slowed adverse effect. This is because of the mutual interaction between net work and boiler's heat value. Redesigning the heat exchanger is generally needed for nanofluids to optimize performance and thus, PPTD will affect the other parameters' performance. It can also be concluded that among the organic working fluids, R236fa gives the best thermal efficiency, while R141b performs the worst.

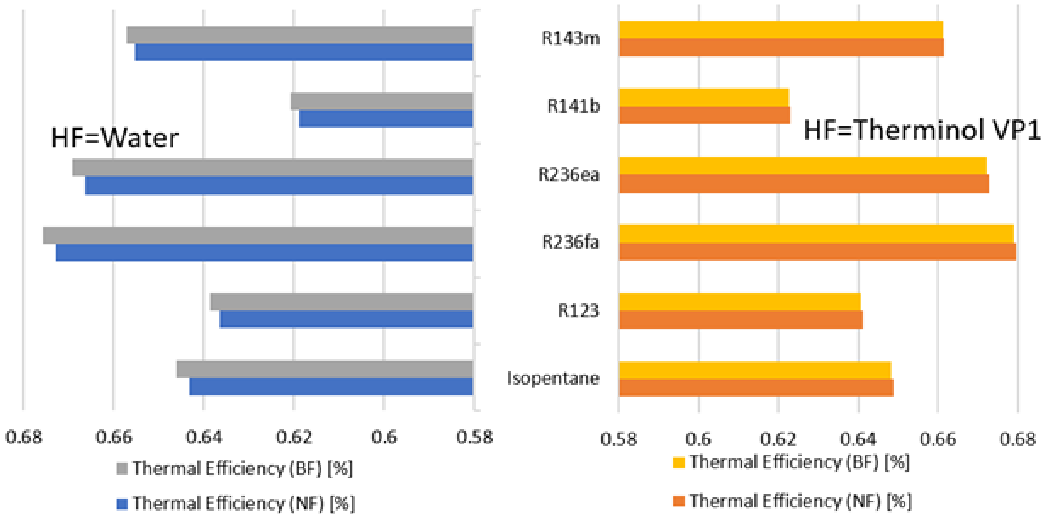

Figure 5.15. Thermal efficiency comparison with different hot fluids and working fluids

Water-based nanofluid shows a greater working fluid mass flow rate than therminol-based nanofluid due to their higher specific heat. Figure 5.16 demonstrates that the working fluids mass flow rate output for nanofluids is lower than their respective base fluids across all organic working fluids, as nanofluids lower the specific heat value, reducing heat transfer to the boiler under the same conditions. This results in a lower mass flow rate of organic working fluid and, consequently, lower turbine work. The PPTD value remains constant because the same heat exchanger is used. Generally, nanofluid applications necessitate heat exchanger redesign, reducing the PPTD for nanofluid-applied boilers.

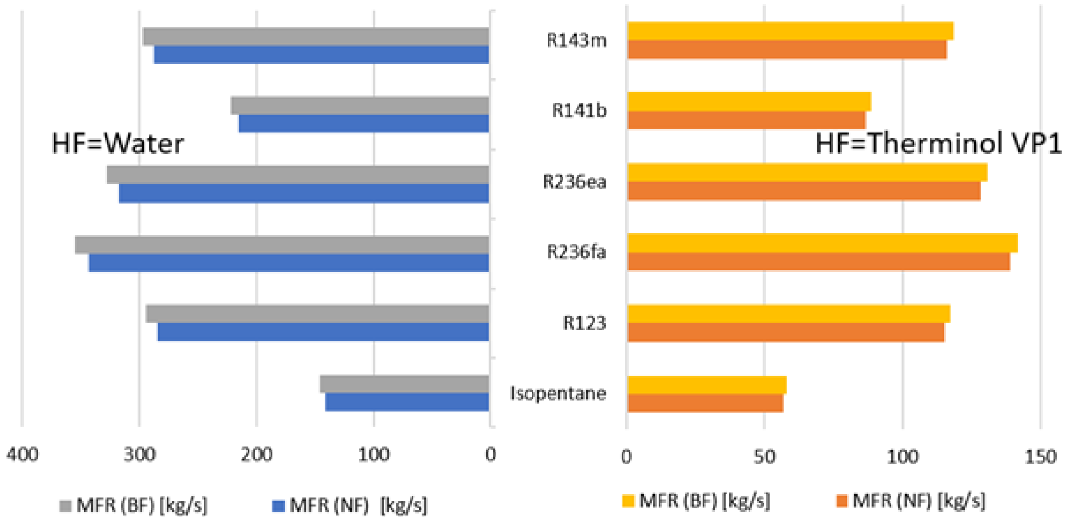

Figure 5.16. Working fluid mass flow rate comparison with different hot fluids and working fluids

Figure 5.17 compares the capital investment using base hot fluids and nanofluids with six different organic working fluids. Water-based nanofluid and water as base fluid yield higher capital investment due to greater specific heat of water which requires large size of heat exchangers (boiler and condenser) to handle large amount of heat. Figure 5.17 also shows, the capital investment value for nanofluids is lower than for their respective base fluids because nanofluids have lower specific heat and higher thermal conductivity than base fluid which affects the higher heat transport coefficient for nanofluid-based heat exchangers. Among the organic working fluids, R143m requires the largest capital investment, and R141b requires the lowest for both base hot fluid and its nanofluid combination.

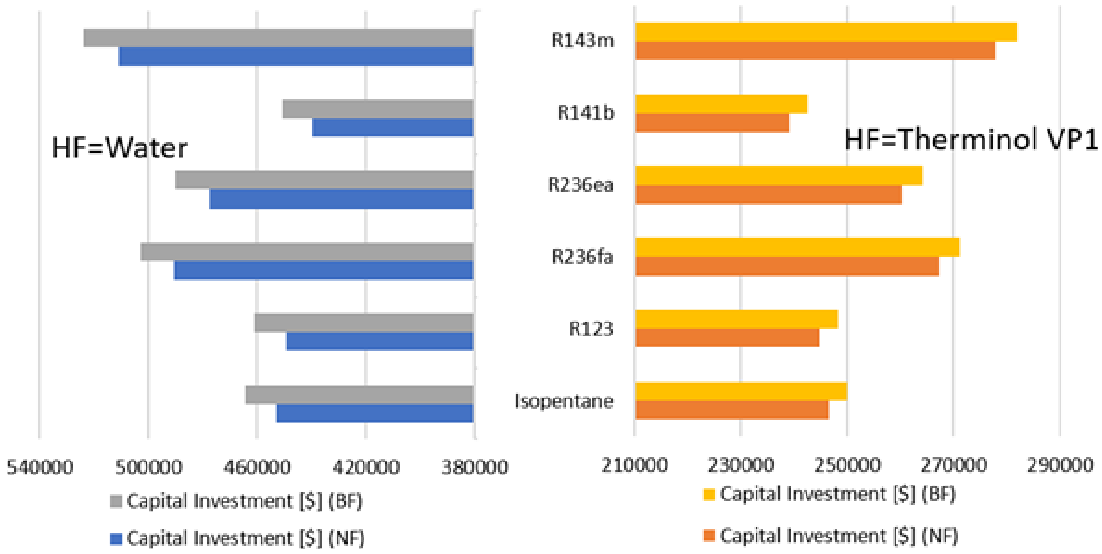

Figure 5.17. Capital investment comparison with different hot fluids and working fluids

Figure 5.18 illustrates the earned profit ($) comparison of ORC when water and therminol-VP1 hot fluid and its nanofluid are used with six different organic working fluids. The findings indicate that water-based nanofluid and water as a base fluid deliver higher profit (around 2.5 times higher) than therminol. This is due to the water's superior specific heat capacity (around 2.5 times higher) than therminol-VP1. However, the earned profit value for nanofluids is consistently lower than for their respective base fluids across all tested organic working fluids. This is attributed to the reduced specific heat capacity, higher thermal conductivity and increased heat transfer coefficient of nanofluids. Among the evaluated working fluids, R236fa and R236ea exhibited the highest performance and provided maximum profit, while R141b gives the lowest.

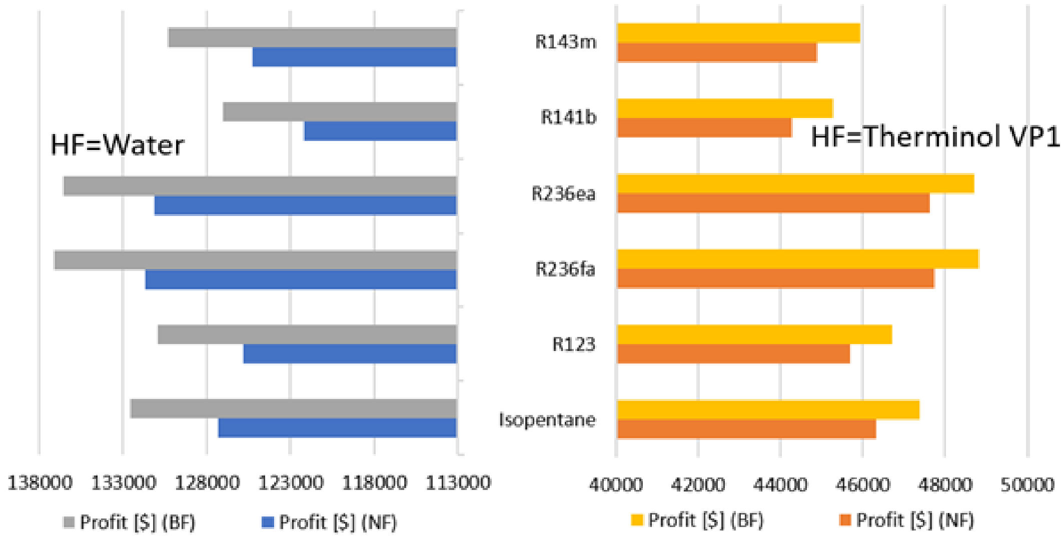

Figure 5.18. Earned profit comparison with different hot fluids and working fluids

5.4. Important Findings

It is important to find out the potential harnessing capacity of exhaust industrial heat to generate electricity with noticeable profit. In this regard, BRNS has provided the waste heat data of its nuclear power plant to generate electricity from its feasible location. The following outcomes are gathered from this study.

(i) Nine potential waste heat locations in nuclear power plants are identified and two feasible cases are proposed for bottoming ORC for waste heat recovery. Both cases are designed based on condenser temperature to optimize waste heat recovery.

(ii) Data from BRNS enabled the prediction of heat exchanger designs, component costs, overall power plant costs, net annual profit from surplus electricity, and environmental impact. This study proposed a capital cost correlation for alternators to accurately estimate system expenses.

(iii) Working fluid R245fa excelled in case-1, while R236ea was optimal for case-2, delivering higher yearly profits than R601a (Isopentane). Case 2, with a condenser temperature below 41°C, is recommended for NPP sites due to superior thermal and economic performance and faster capital returns. Both cases could generate 499.3 kW of electricity and an annual profit of 204.62 thousand USD, with a total capital investment of 764.96 thousand USD.

(iv) For a 40-year lifespan, the total NPV (capital and O&M investments) is 1.392 million USD, ensuring a minimum annual profit of 204.62 thousand USD.

(v) R236fa emerges as the most effective working fluid for ORC applications in terms of power output and profitability, followed by R236ea and isopentane (R601a). While R236fa offers the best thermodynamic performance, its high global warming potential (GWP: 9810) limits its usage in many countries, with R236ea (GWP: 1370) presenting a more environmentally viable option. However, both R236fa and R236ea require careful handling due to their low normal boiling points. So, if these drawbacks are considered, then isopentane is the best choice for the ORC application.

(vi) Experimental results showed significantly lower net power output and thermal efficiency compared to simulations due to constraints such as air-based cooling, evaporator under-sizing, mass flow rates and fixed hot fluid outlet temperatures but validated both cases to generate electricity.

(vii) Therminol-VP1-based nanofluids demonstrated efficiency improvements in waste heat recovery with required redesigned heat exchangers to fully utilize nanofluid advantages.