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# Analysis Energy and Exergy Residual Heat Utilization with additional Preheater in Organic Rankine Cycle

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***Abstract* - On this paper, ORC thermal efficiency increased 22.54%, ORC utilization increased 22.79%, and ORC Exergetic efficiency increased 22.78% from the HMB design. Author has analysis to change the specification of Feed Pump, and additional Preheater, result analysis, when increasing n-pentane flow rate and saturation temperature, the heat (Q) flowing into the reinjection well decreased from 52502.9 kW to 23488.17 kW, and exergy destruction decreased from 28536 kW to 20427 kW where this exergy injected into the reinjection well, means that some energy and exergy has been utilized before being flowed into the reinjection system. On the Turbine, increase in Gross Power ($W_{Turbine}$) 25.40% with gross power modification 17418 kW from Gross Power 13890 kW and increase net power 15102 kW and 12050 kW. In the ACHE, increase heat (Q) 27.10% from 76030 kW to 96633 kW which need to cool n-pentane, increase in heat (Q) followed by increasing in power Fan motor 14.66% where the air flow rate increases from 218798 ACFM to 294442 ACFM which need to cooled n-pentane. The power of the Feed pump increases 31.69% to 1600 kW from 1215 kW, this is because change in impeller diameter causes an increase in flowrate, pressure and motor power need to rotate the pump. On the Recuperator there is decrease in work (Q) 47.93%, this is because heating n-pentane to reach saturation temperature assisted by the presence of an additional preheater.**

***Keywords:*** ORC thermal efficiency, ORC utilization, ORC Exergetic efficiency, Additional Preheater.

## I. INTRODUCTION

Organic Rankine Cycle (ORC) technology is used for geothermal liquid (brine) utilization at low and medium temperatures and using a working fluid with a low boiling point. ORC cycle is adapted to all brine sources with low - medium temperatures 100-170°C [1]. The results of other studies reveal that the best working fluids are butane, neo-pentane and R245A with a temperature brine of100-150°C. At temperatures brine 100 - 150°C, other working fluids that are recommended and can be used in this ORC system are R245a, R11, R113, R114, R114b, R601 and R601a [2]. Other research[3], revealed that geothermal liquid sources with a temperature of 90°C, in this research used 27 working fluids (with boiling points between -47.69 to 47.59°C) which were divided into 4 groups, where group I, Tb < -35°C, group II, -35°C<Tb<-15°C, group III, 15°C<Tb<0°C and group IV, Tb > 0°C,where the results obtained in each group with the potential for the greatest electrical power that can be generated (Pnet) using working fluids R218 and R115 in group I, R227ea and R1234yf in group II, R318 and R236fa in group III, R236ea in group IV. In another study [4], the "Preheat – Parallel" arrangement of a brine source with a temperature 130°C and flow rate of 193 kg/s, using R236ea as the working fluid in the ORC system and. combined with a heating system. Meanwhile, [5], in this research, used geothermal liquid with a temperature 393 K (119°C), and the ORC working fluids used R21, R114 and R245fa.

In other research [6], simulation using water where the temperature used is 75°C to 95°C and using working fluid R245fa. Others research [7], conduct experiments using TY-1 as working fluid, with a heat source from water with a temperature of 65-95°C. Others research [8], using water with source temperature of 20-150°C, and using 16 working fluids which were simulated in single ORC, dual ORC and tripartite ORC.

In others research [9] explains that geothermal liquid/brine with temperatures below 80°C originating from geothermal sources is not economical due to low efficiency. Meanwhile [10], explains in this research, sufficient temperature is needed to prevent oversaturation which can cause silica scaling and damage to the heat exchanger. At low enthalpy geothermal, with a temperature between 110°C - 160°C, where the reinjection temperature is not considered to be injected below 70-80°C [11].

PLTP XYZ is a geothermal power plant that currently uses Organic Rankine Cycle technology. PLTP XYZ with total capacity 3 x 110 MW. PLTP XYZ uses two geothermal fluid phases, steam and geothermal liquid/brine. The two geothermal fluid phases are separated in the Separator, where the steam enters to the back pressure turbine at high pressure and the exhaust steam from the back pressure turbine is used

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to heat n-pentane in the ORC bottoming unit. Meanwhile, the brine/geothermal liquid from the accumulator enters to the ORC binary unit, then the brine that comes out of the ORC binary and the ORC bottoming condensate injected into the reinjection well.

Brine is the result of separation in the separator and accumulator, with a temperature of 209 – 211°C. With this temperature, brine is used as a heat source in small-scale ORC plants. PLTP XYZ uses this brine to generate electricity 13.8 MW. The binary cycle itself is an electricity generation cycle where in the process it uses a secondary fluid or working fluid with a low boiling point which produces vapor on the shell side of the heat exchanger by exchanging heat with brine on the tube side. The working fluid used at PLTP XYZ in this research n-pentane.

Temperature brine coming out from ORC_X 139.7°C - 141.8°C with a brine flow rate of 1050 t/h - 1113 t/h then brine injected into the reinjection well. The brine coming out of the ORC_X still contains wasted thermal energy which analyzed by the author to reused and improves the performance, efficiency of the ORC_*X*. In this study, author proposes additional preheater component as an alternative to utilizing brine that still has potential thermal energy. The brine coming out from the existing Preheater flowed into an additional preheater, heating the working fluid n-pentane and brine coming out from the outlet additional preheater where the temperature has dropped then injected into the reinjection well. While in another work, the flow of working fluid n-pentane enters to the existing preheater in the ORC_X then goes to the vaporizer and heated and create vapor then vapor reach saturation temperature enters the turbine inlet and rotates the Turbine-Generator. By adding a preheater to the existing ORC cycle, the author analyzes energy and exergy of each ORC component.

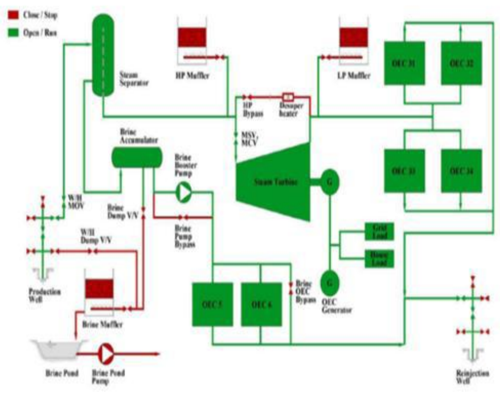

Figure 1: Mode Normal Operation

In this study, the author created a research framework by conducting a literature study first, then continued by determining the research parameters and model assumptions. In the validation stage, the author verified the ORC Heat Mass Balance model or design using EES software, then the author used assumptions and field data as input in calculating the energy and exergy analysis of each component.

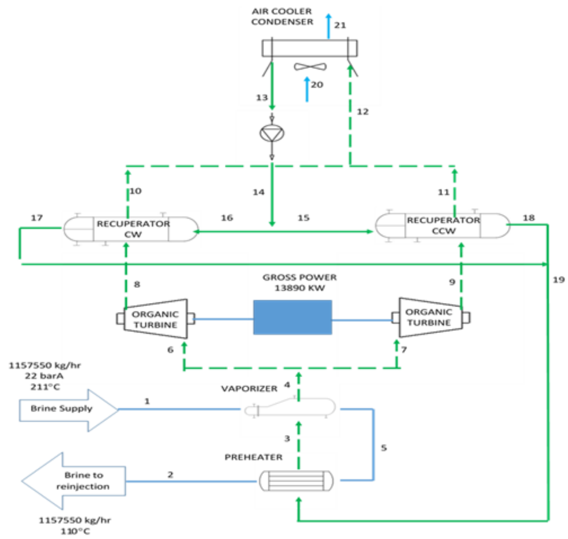


Figure 2: Heat Mass Balance ORC

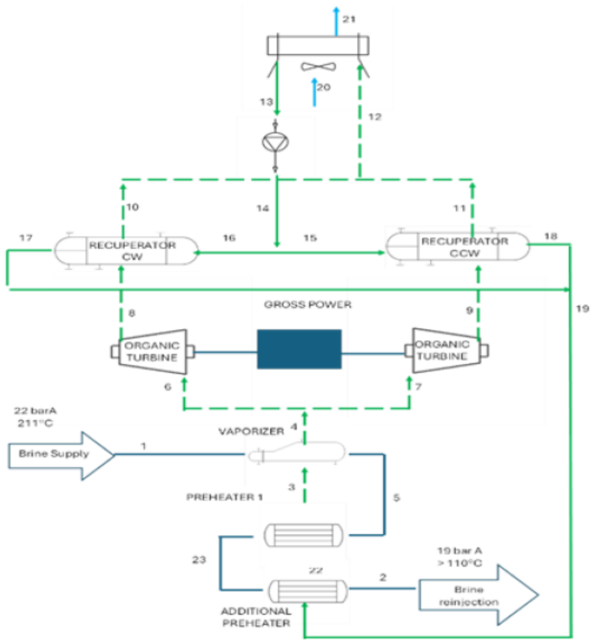


Figure 3: ORC modification

## II. RESEARCH METHOD

In this study, the author created a research framework by conducting a study literature, then continued by determining the research parameters and model assumptions. In the validation stage, the author verified the ORC Heat Mass Balance model or design using EES software, then the author used assumptions and field data as input in calculating the energy and exergy analysis for each component.

 

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## III. RESULTS AND DISCUSSIONS

In this research, the author carried out an energy and exergy analysis for each component, including energy input and output, exergy destruction and exergy efficiency which are presented in the table below. The author was calculation used EES software.

**Table 1: Energy & Exergy Analysis Formula**

| **Analisis ORC penambahan Preheater** | |
|---|---|
| **Vaporizer dan Preheater additional** | |
| Energi Balance at Vaporizer and Preheater additional | $\dot{m}_1 (h_1 - h_2) = \dot{m}_4 (h_4 - h_{19})$ |
| **Vaporizer** | |
| Energi balance | $m_1\ cp_1\ [T_1 - T_5] = m_4\ (h_4 - h_3)$ |
| $Q_{VAPORIZER}$ | $\tilde{U}A_{VAPORIZER}$ LMTD |
| Exergy balance | $\dot{m}_1\ ex_1 + \dot{m}_3\ ex_3 = \dot{m}_5\ ex_5 + \dot{m}_4\ ex_4 + Ex_{d\ VAPORIZER}$ |
| **Preheater 1** | |
| Energi balance | $m_5\ cp_5\ [T_5 - T_{23}] = m_{22}\ (h_3 - h_{22})$ |
| $Q_{PREHEATER\ 1}$ | $\tilde{U}A_{PREHEATER\ 1}$ LMTD |
| Exergy balance | $\dot{m}_5\ ex_5 + \dot{m}_{22}\ ex_{22} = \dot{m}_{23}\ ex_{23} + \dot{m}_3\ ex_3 + Ex_{d\ PREHEATER\ 1}$ |
| **Preheater additional (Preheater_2)** | |
| Energi balance | $m_{23}\ cp_{23}\ [T_{23} - T_2] = m_{19}\ (h_{22} - h_{19})$ |
| $Q_{PREHEATER\ ADD}$ | $\tilde{U}A_{PREHEATER\ ADD}$ LMTD |
| Exergy balance di Preheater_2 | $\dot{m}_{23}\ ex_{23} + \dot{m}_{19}\ ex_{19} = \dot{m}_2\ ex_2 + \dot{m}_{22}\ ex_{22} + Ex_{D,PHADD}$ |
| **Turbin CW & CCW** | |
| Energi balance | $\dot{m}_6 h_6 = \dot{m}_8 h_8 + \dot{W}_{ORC,TCW}$ atau $\dot{m}_7 h_7 = \dot{m}_9 h_9 + \dot{W}_{ORC,TCW}$<br>$\dot{W}_{ORC,TCW} = \dot{m}_6\ (h_6 - h_8)$ atau $\dot{m}_6\ \eta_T\ (h_6 - h_{8S})$ atau<br>$\dot{W}_{ORC,TCW} = \dot{m}_7\ (h_7 - h_9)$ atau $\dot{m}_7\ \eta_T\ (h_7 - h_{9S})$ |
| Exergy balance | $\dot{m}_6\ ex_6 = \dot{m}_8\ ex_8 + \dot{W}_{ORC,TCW} + Ex_{D,ORCTCW}$ atau $\dot{m}_7\ ex_7 = \dot{m}_9\ ex_9 + \dot{W}_{ORC,TCCW} + Ex_{D,ORCTCCW}$ |
| Efisiensi Turbine & Exergi Turbine | $\eta_T = \frac{(\dot{h}_6 - \dot{h}_8)}{(\dot{h}_6 - \dot{h}_{8S})}$ ; $\psi_T = \frac{\dot{m}_8 \dot{ex}_8 + \dot{W}_T}{\dot{m}_6 ex_6}$; $\eta_T = \frac{(\dot{h}_7 - \dot{h}_9)}{(\dot{h}_7 - \dot{h}_{9S})}$ ; $\psi_T = \frac{\dot{m}_9 \dot{ex}_9 + \dot{W}_T}{\dot{m}_7 ex_7}$ |
| **Recuperator CW & CCW** | |
|  | $\dot{m}_8 h_8 + \dot{m}_{16} h_{16} = \dot{m}_{10} h_{10} + \dot{m}_{17} h_{17}$ atau $\dot{m}_9 h_9 + \dot{m}_{15} h_{15} = \dot{m}_{11} h_{11} + \dot{m}_{18} h_{18}$ |
|  | $m_8\ cp_8\ [T_8 - T_{10}] = m_{16}\ (h_{16} - h_{17})$ atau $m_9\ cp_9\ [T_9 - T_{11}] = m_{15}\ (h_{15} - h_{18})$<br>$m_8\ (h_8 - h_{10}) = m_{16}\ (h_{16} - h_{17})$ atau $m_9\ (h_9 - h_{11}) = m_{15}\ (h_{15} - h_{18})$ |
| $Q_{RECUPERATOR}$ | $\tilde{U}A_{RECUPERATORCW}$ LMTD |
| Exergy balance | $\dot{m}_8\ ex_8 + \dot{m}_{16}\ ex_{16} = \dot{m}_{10}\ ex_{10} + \dot{m}_{17}\ ex_{17} + Ex_{D,RCPCW}$ |
| **ACHE** | |
| Energi balance | $\dot{m}_{12} h_{12} + \dot{m}_{20} h_{20} = \dot{m}_{13} h_{13} + \dot{m}_{21} h_{21}$ |
|  | $m_{20}\ cp_{20}\ [T_{21} - T_{20}] = m_{12}(h_{12} - h_{13})$ |
| $Q_{ACHE}$ | $\tilde{U}A_{ACC}$ LMTD |
| Exergy balance | $\dot{m}_{12}\ ex_{12} + \dot{m}_{20}\ ex_{20} = \dot{m}_{13}\ ex_{13} + \dot{m}_{21}\ ex_{21} + Ex_{dACC}$ |
| $W_{FAN}$ & $W_{gross}$ | $\frac{P_{diff} \text{ x Volume Fan}}{6342 \text{ x effisiensi Fan}}$ ; $\frac{W_{FAN}}{effisiensimotor}$ |
| **Pompa** | |
| Energi balance di pompa | $\dot{m}_{14} h_{14} = \dot{m}_{13} h_{13} + \dot{W}_{Pump}$<br>$\dot{W}_P = \dot{m}_{13}\ (h_{14} - h_{13}) = \dot{m}_{13}\ (h_{14S} - h_{13}) / \eta_p$ |
| Exergy balance di pompa | $\dot{m}_{14}\ ex_{14} + Ex_{d,PUMP} = \dot{m}_{13}\ ex_{13} + \dot{W}_P$ |
| Efisiensi Energi pompa & Exergi pompa | $\eta_p = \frac{\dot{m}_{14} h_{14} - \dot{m}_{13} h_{13}}{W_P}$ ; $\psi_T = \frac{\dot{m}_{14} ex_{14} - \dot{m}_{13} ex_{13}}{W_P}$ |

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**Table 2: Energy and Exergy Analysis Design HMB**

| SUMMARY Calculation | | | | | | | |
|---|---|---|---|---|---|---|---|
| WORK (Q) | Design HMB Work (KW) | Work EES Simulation (KW) | Different | Exergy Input (KW) | Exergy Output (kW) | Exergy Destruction (KW) | Utilisasi (%) |
| ACHE | 76030 | 75245 | 1.03% | 7,206 | 1,167 | 6,039 | 16 |
| PREHEATER | 55220 | 56140 | 1.67% | 48,877 | 45,150 | 3,727 | 92 |
| RECUPERATOR CW | 5560 | 5504 | 1.01% | 5,088 | 4,716 | 372 | 93 |
| RECUPERATOR CCW | 5560 | 5504 | 1.01% | 5,088 | 4,716 | 329 | 93 |
| VAPORIZER | 34000 | 34422 | 1.24% | 75,881 | 73,959 | 1,922 | 97 |
| PUMP | 1215 | 1247 | 2.63% | 1,526 | 1,073 | 395 | 70 |
| FAN | 625 | 638.4 | 2.14% | | | | |
| TURBINE gross CW elc | | 7097 | | 13,647 | 11,626 | 2,021 | 85 |
| TURBINE gross CCW elc | | 7097 | | 13,647 | 11,626 | 2,021 | 85 |
| Turbine Gross total ORC | 13890 | 14194 | 2.19% | | | | |
| Wnet ORC | 12050 | 12308.6 | 2.15% | | | | |
| Reinjection Well | | 52502.9 | | | | 28,536 | |
| thermal eff | | 13.79 | % | | | | |
| utilisasi ORC | | 20.74 | % | | | | |
| eksergetic eff ORC | | 40.01 | % | | | | |

From data above, the deviation between the Design Heat Mass Balance and the EES simulation is below 5%, so the thermodynamic equations in the EES have been validated.

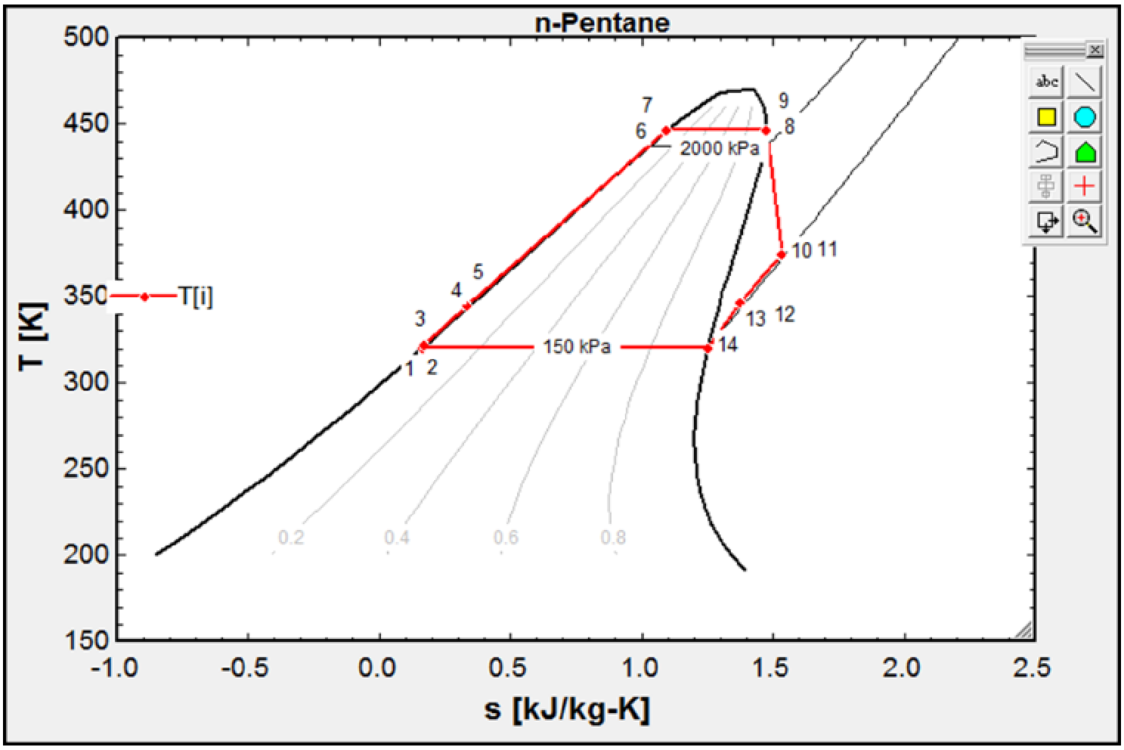


**Figure 4: T-S Diagram HMB design EES**

In below figure the Sankey diagram, the greatest exergy damage occurs in brine that injected into the reinjection well 28,535 kW. From the exergy value, the author conducted an analysis by utilizing the heat brine which still has the potential thermal energy by lowering the brine temperature from $140^0$ C to $110^0$ C -$125^0$C.

 

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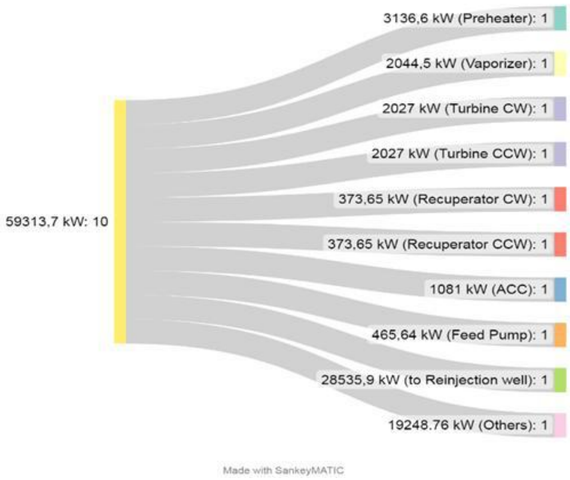


**Figure 5: Sankey Diagram**

Effect of increasing n-pentane saturation temperature, pressure saturation and n-pentane flow rate.

**Table 3: Effect of increasing n-pentane saturation temperature and n-pentane flow rate**

| SUMMARY ANALYSIS | | | | EXERGY (EES) | | | |
|---|---|---|---|---|---|---|---|
| WORK (Q) | HMB (KW) | EES (KW) | diff | EXERGY INPUT (KW) | EXERGY DESIRED OUTPUT (KW) | EXERGY DESTRUCTION (KW) | PERCENT |
| ACHE | 76030 | 96633 | 27.10%↑ | 9,720 | 1,628 | 8,092 | 16.7 |
| PREHEATER | 55220 | 51744 | 6.29%↓ | 53,004 | 49,627 | 3,377 | 93.6 |
| ADDITIONAL PREHEATER | 0 | 26370 | 100.00%↑ | 30,577 | 26,785 | 3,792 | 87.6 |
| RECUPERATOR CW | 5560 | 2895 | 47.93%↓ | 6,193 | 5,886 | 307 | 95.0 |
| RECUPERATOR CCW | 5560 | 2895 | 47.93%↓ | 6,193 | 5,886 | 329 | 95.0 |
| VAPORIZER | 34000 | 35469 | 4.32%↑ | 80,395 | 79,708 | 687 | 99.1 |
| PUMP | 1215 | 1600 | 31.69%↑ | 1,972 | 1,538 | 395 | 78.0 |
| FAN | 625 | 716.6 | 14.66%↑ | | | | |
| TURBINE gross CW | | 8709 | | 16,531 | 14,134 | 2,397 | 85.5 |
| TURBINE gross CCW | | 8709 | | 16,531 | 14,134 | 2,397 | 85.5 |
| Turbine Gross total | 13890 | 17418 | 25.40%↑ | | | | |
| Wnet | 12050 | 15101.4 | 25.32%↑ | | | | |
| To Reinjection Well | | 23488 | | | | 20427 | |
| Thermal effisiensi | 13.80 | 16.91 | 22.54%↑ | | | | |
| Utilisasi ORC | 20.74 | 25.47 | 22.79%↑ | | | | |
| Eksergetic Effisiensi ORC | 39.97 | 49.08 | 22.78%↑ | | | | |

From table 3, it can be concluded that from the analysis of increasing n-pentane flow rate and saturation temperature, the heat (Q) flowing into the reinjection well decreased from 52502.9 kW to 23488.17 kW, and exergy destruction decreased from 28536 kW to 20427 kW where this exergy injected into the reinjection well, means that some exergy has been utilized before being flowed into the reinjection system. On the Turbine, increase in Gross Power ($W_{Turbine}$) of 25.40% (17418 kW gross and 15102 kW net) from Gross Power 13890 kW and (Wnet) 12050 kW. In the ACHE, increase heat (Q) 27.10% from 76030 kW to 96633 kW which need to cooln-pentane, increase in heat (Q) followed by an increase in power Fan motor 14.66% where the air flow rate increases from 218798 ACFM to 294442 ACFM which need to cooled n-pentane. The power of the Feed pump increases 31.69% to 1600 kW from 1215 kW, this is because change in impeller diameter causes an increase in flowrate, pressure and motor power need to rotate the pump. On the Recuperator there is decrease in work (Q) 47.93%, this is because heating n-pentane reaches saturation temperature assisted by the presence of an additional preheater. ORC thermal efficiency increased 22.54% from the initial efficiency, ORC system utilization increased 22.79% from the existing HMB design, and ORC exegetic efficiency increased 22.78% from the HMB design.

By using parametric table in EES software, increase saturation temperature T[4] coming out of Vaporizer and entering Turbine will increase ORC exegetic efficiency, ORC system utilization efficiency and ORC thermal efficiency. On the other hand, Turbine work increases following the increase of temperature saturation and increase flowrate of n-pentane.

 

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**Table 4: Analysis of the increase in n-pentane saturation temperature on ORC efficiency at a temperature of 169$^0$ C – 178$^0$C with a flow rate of 226 kg/s**

| 1..10 | $T_4$ [K] | $h_6$ [kJ/kg] | $h_8$ [kJ/kg] | $m_{14}$ [kg/s] | $\eta_{eksergetic,plant,ORC}$ [%] | $\eta_{Thermal,effisiensi,ORC}$ [%] | $\eta_{utilisasi,effisiensi,ORC}$ [%] | $W_{netModifORC}$ [kW] | $W_{Turbine,gross,ModifORC}$ [kW] |
|---|---|---|---|---|---|---|---|---|---|
| Run 1 | 451.2 | 584.4 | 503.2 | 226 | 44.51 | 15.06 | 25.49 | 15113 | 17423 |
| Run 2 | 450.2 | 584 | 503.2 | 226 | 44.25 | 14.98 | 25.34 | 15026 | 17335 |
| Run 3 | 449.2 | 583.5 | 503.2 | 226 | 43.96 | 14.88 | 25.17 | 14925 | 17234 |
| Run 4 | 448.2 | 583 | 503.2 | 226 | 43.62 | 14.76 | 24.98 | 14811 | 17120 |
| Run 5 | 447.2 | 582.4 | 503.2 | 226 | 43.25 | 14.64 | 24.77 | 14685 | 16994 |
| Run 6 | 446.2 | 581.8 | 503.2 | 226 | 42.85 | 14.5 | 24.53 | 14547 | 16857 |
| Run 7 | 445.2 | 581.1 | 503.2 | 226 | 42.41 | 14.35 | 24.29 | 14400 | 16709 |
| Run 8 | 444.2 | 580.3 | 503.2 | 226 | 41.95 | 14.2 | 24.02 | 14242 | 16551 |
| Run 9 | 443.2 | 579.6 | 503.2 | 226 | 41.45 | 14.03 | 23.74 | 14075 | 16384 |
| Run 10 | 442.2 | 578.7 | 503.2 | 226 | 40.94 | 13.85 | 23.44 | 13899 | 16208 |

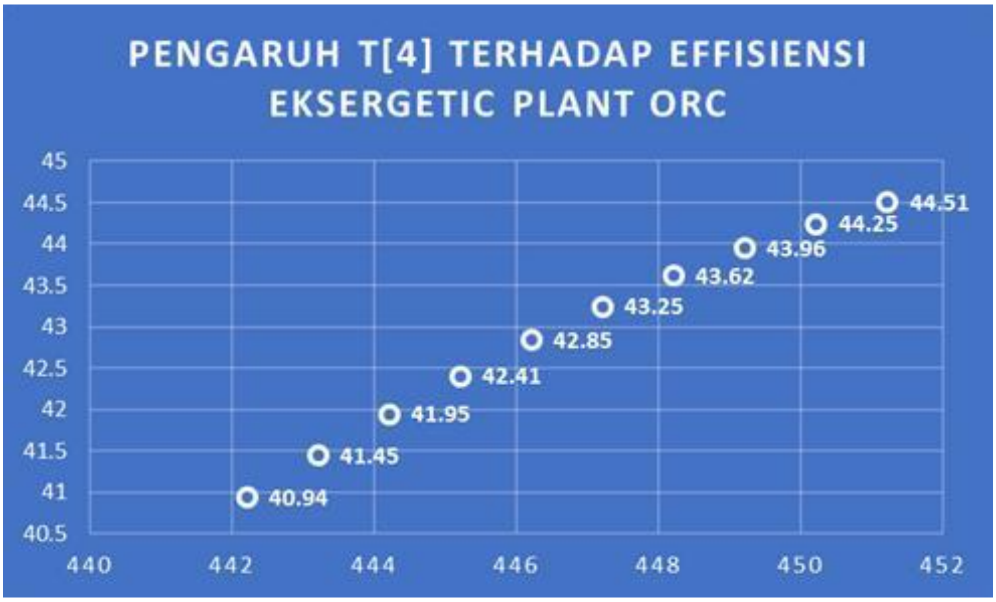


**Figure 6: Effect of increasing saturation temperature T[4] and ORC exergetic efficiency**

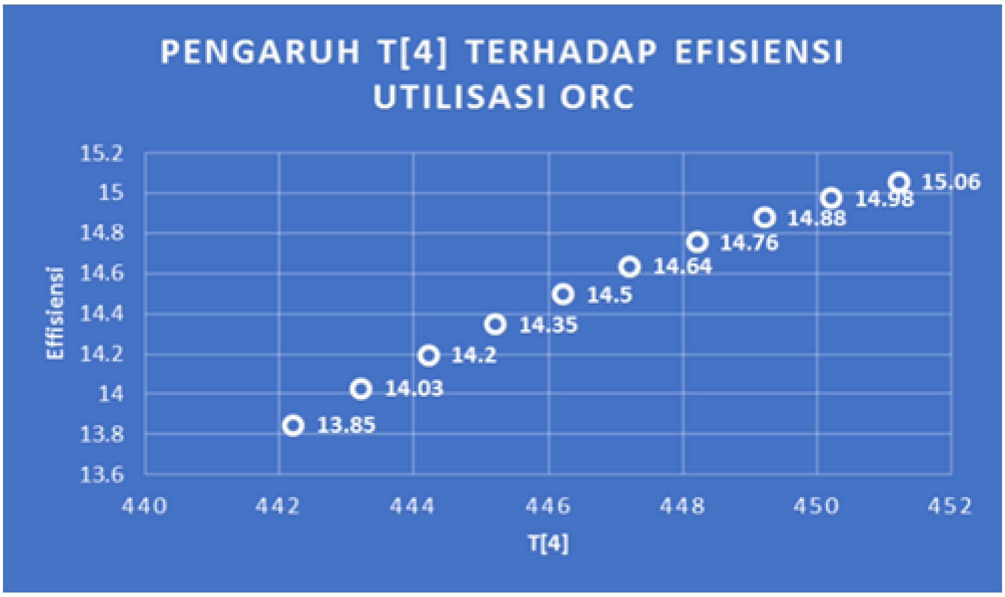


**Figure 7: Effect of increasing saturation temperature T[4] and ORC Utilization Efficiency**

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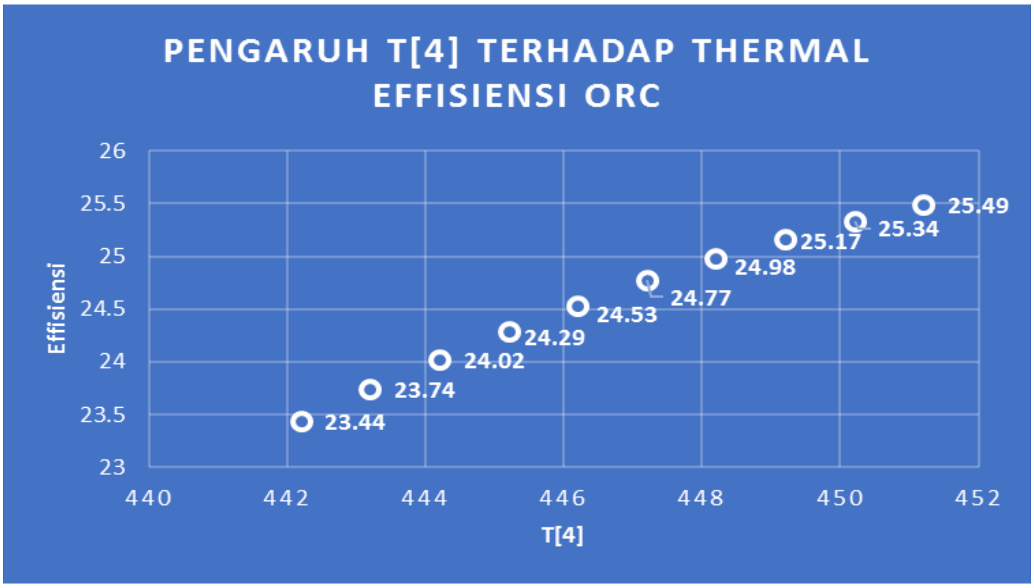


**Figure 8: Effect of increasing saturation temperature T[4] And Thermal efficiency ORC**

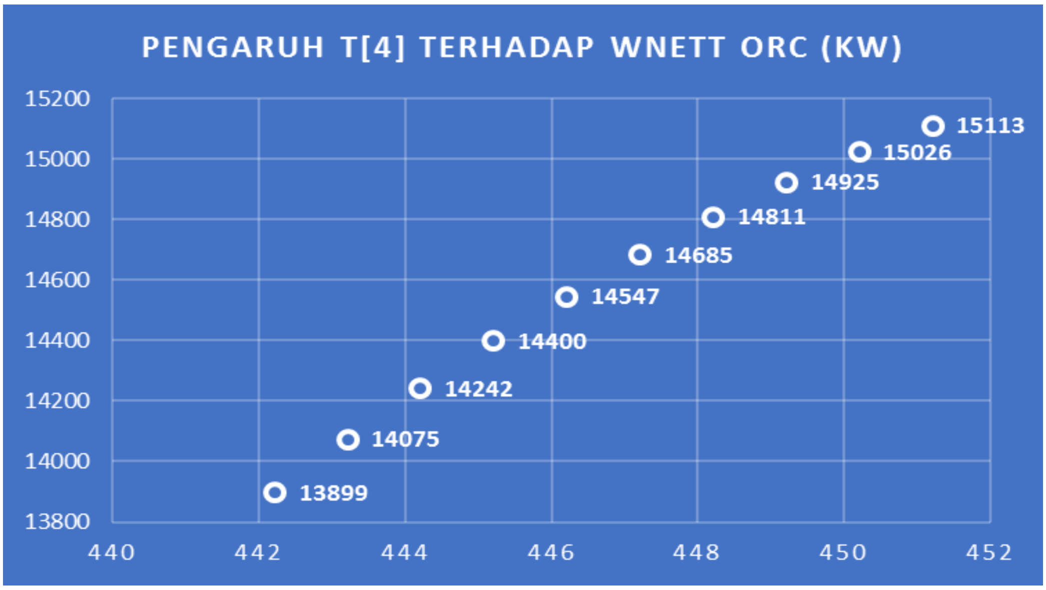


**Figure 9: Effect of increasing saturation temperature T[4] and Wnett ORC**

Changes in specifications on the Feed Pump, where the pump pressure changes to total head (TDH) 1657.3 feet where previously 1502 feet, n-pentane flowrate 195 kg/s becomes 226 kg/s with 2 Feed Pumps running in parallel, turbine inlet enthalpy increases than resulting Wnett increase, including ORC Exergy efficiency, Utilization efficiency, Thermal efficiency.

**New Feed Pump specifications:**

By using similarity formula on the pump, changes in the Feed pump impeller design are obtained as follows:

$\frac{H}{Hv} = \frac{N}{Nv} x\, (\frac{D}{Dv})^2$, where the speed N is the same at 1500 rpm, where the existing impeller diameter D is 10.55 inches and the diameter Dv is 11.08 inches.

The flow rate of the Feed pump with changes impeller can be calculated using formula:

$\frac{Q}{Qv} = \frac{N}{Nv} x\, (\frac{D}{Dv})^3$, with the existing design of the pentane flow rate of 195 kg/s to 226 kg/s.

 

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**Pump Specification:**

1. Head Type : VTP
2. Pump Head : 1657.2 FT
3. Capacity : 406.6 m3/h = 1791 GPM
4. Impeller Type : Enclose

Motor Specification:

5. HP : 450 kW
6. RPM : 1490
7. Phase/F/V : 3/50/400
8. Frame No. : 355J/H

**Pipe pressure in the existing ORC system**

Changes Feed pump specifications, author analyzed in relation to the existing pipe pressure installed. In the existing pipe installed, the pipe used the following specifications:

a. ½” – 1 1/2” seamless, schedule XS, ASTM A106 Gr B/ A.53 Gr. B
b. 2” – 6” seamless, schedule XS, ASTM A106 Gr B/ A.53 Gr. B
c. 8” – 24” ERW/ SAW, schedule standard, API 5LB/ A.53 Gr. B

$P = \frac{2tSEW}{D+2.c+2t(1-Y)}$ or $P = \frac{2tSEW}{D-2tY}$

| $D_n$ Nominal Dia, in | $D_o$ Outside Dia, in | Sch No | Wall Thk, in | $D_i$ Inside Dia, in | t-nominal Thickness, in | CA Corrosion Allowance in | ASME B31.3 2008 Allowable Internal Pressure $P_i$ =2tS,pipeEW/(D+2c+2t(1-Y)) In-Plant Piping | ASME B31.3 2008 Allowable Internal Pressure $P_i$ =2tS,pipeEW/(D+2c+2t(1-Y)) In-Plant Piping |
|---|---|---|---|---|---|---|---|---|
| | | | | | Wall Thk | User Input | psig | bar |
| 10 (10.750) | 10.75 | 40 | 0.365 | 10.020 | 0.365 | 0.118 | 634 | 43.7 |
| 18 | 18 | 40 | 0.562 | 16.876 | 0.562 | 0.118 | 711 | 49.0 |
| 12 (12.75) | 12.75 | 40 | 0.406 | 11.938 | 0.406 | 0.118 | 632 | 43.6 |
| 1 (1.305) | 1.305 | | 0.133 | 1.039 | 0.133 | 0.000 | 3,378 | 232.9 |
| 24 | 24 | 40 | 0.687 | 22.626 | 0.687 | 0.118 | 691 | 47.6 |

**New specifications Heat Exchangers**

Changes in Feed Pump specifications cause design changes to the heat exchanger. This can be seen from changes in the overall heat transfer area (A) on each heat exchanger; this can be seen in the table below:

| Equipment Detail | Overall Heat Transfer Area (m2) | | |
|---|---|---|---|
| | DESIGN | EES Modification | Percentage diff |
| ACHE | 4757.3 | 6040 | 26.96%↑ |
| PREHEATER | 1218.8 | 1196 | 1.87%↓ |
| ADDITIONAL PREHEATER | 0 | 512.2 | 100.00%↑ |
| RECUPERATOR CW | 1503.7 | 789.2 | 47.52%↓ |
| RECUPERATOR CCW | 1503.7 | 789.2 | 47.52%↓ |
| VAPORIZER | 1059 | 1128 | 6.52%↑ |

**New Turbine ORC Specification**

Turbine operating conditions, based on changes in Feed pump are determined in advance. Turbine operating conditions are adjusted to the following operating parameters:

 

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**Parameter Value**
Suhu Inlet Turbine : 1780C
Suhu outlet Turbine : 101.70C
Tekanan Inlet Turbine : 25.5 bar
Tekanan Outlet Turbine : 1.72 Kpa
Mass flow : 226 kg/s

**New Turbine Specifications:**
1. Rated speed : 1500 rpm
2. Rated Power : 9175 kW (2 units each ORC)
3. Turbine Type: Horizontal, Model 1500
4. Casing Split : Vertical
5. Number of Stages : 3
6. End Seal : Double Mechanical Seal
7. Temperature: 185$^0$C (critical point of n-pentane 193.9$^0$C)

**New Specification Fan ACHE**
Fan Description:
Number of Fans : 42 fans
Required Volume each : 294442 ACFM
Fan Diameter : 4877 mm
Rotor Shaft Power each : 17 kW

**Economic Analysis**

In economic analysis sectio, author assumes that changes in the Feed Pump specifications and the addition of a preheater, both thermodynamic analysis and economic analysis need to be analyzed first. The total investment cost consists of production costs influenced by the cost of equipment components: Heat exchanger, Turbine and Pump, other capital costs, operating and maintenance costs based on assumptions and suggestions (Bejan and Tsatsaronis 1996).

Investment cost using Chemical Engineering Plant Index (CEPCI2001 = 397; CEPCI2013 = 564; CEPCI2018 = 648.7)

Investment cost Preheater, it is calculated using the formula: $= 1397 \text{ x } A^{0.89}_{Add_{Pre\ heater}}$

Investment Cost Feed Pump $= 1120 \text{ x } W^{0.8}_{Feed\ \ Pump}$

Investment Cost Fan $= 1.31 \text{ x } 10^4 \text{ x } \left(\frac{W}{50}\right)^{0.76}_{Fan}$

Investment Cost ACHE $=1.67 \text{ x } 10^5 \text{ x } \left(\frac{A}{50}\right)^{0.89}_{ACHE}$

Investment Cost Turbine $=(-1.66 \text{ x } 10^4) + (716 \text{ x } (W/50)^{0.89}_{Turbine}\ )$

**Total Investment Cost equipment : USD 5,091,528.17**

Operation and Maintenance Costs

Operation and maintenance costs use an assumption 1.5% [11]

selling price per kWh USD 0.07 ; running hours in 1 year 7200 hours;

Payback Periode (PBP) : $\frac{\text{Ctot}}{(Wnet\ \ x\ 7200\ x\ 1000)-Com} =$ 4.0 years.

 

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## IV. CONCLUSION

Regarding change specification Feed Pump, and additional Preheater, result analysis, when increasing n-pentane flow rate and saturation temperature, the heat (Q) flowing into the reinjection well decreased from 52502.9 kW to 23488.17 kW, and exergy destruction decreased from 28536 kW to 20427 kW where this exergy injected into the reinjection well, means that some energy and exergy has been utilized before being flowed into the reinjection system. On the Turbine, increase in Gross Power ($W_{Turbine}$) 25.40% with gross power modification 17418 kW from Gross Power 13890 kW and increase net power 15102 kW and 12050 kW. In the ACHE, increase heat (Q) 27.10% from 76030 kW to 96633 kW which need to cool n-pentane, increase in heat (Q) followed by increasing in power Fan motor 14.66% where the air flow rate increases from 218798 ACFM to 294442 ACFM which need to cooled n-pentane. The power of the Feed pump increases 31.69% to 1600 kW from 1215 kW, this is because change in impeller diameter causes an increase in flowrate, pressure and motor power need to rotate the pump. On the Recuperator there is decrease in work (Q) 47.93%, this is because heating n-pentane to reach saturation temperature assisted by the presence of an additional preheater. ORC thermal efficiency increased 22.54% from the initial efficiency, ORC system utilization increased 22.79% from the existing HMB design, and ORC exegetic efficiency increased 22.78% from the HMB design.

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