MME-5177 -Energy Conversion

Modeling and Optimization of ORC Heat Recovery System for Heavy Duty Truck Exhaust

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Abstract

This paper aims to investigate the optimal thermodynamic Organic Rankine Cycle (ORC) model for a heat recovery system implemented in heavy duty trucks. Three different working fluids, R245fa, Cyclopentene, and Ethanol, were studied through parametric studies that varied the evaporator pressure. The selected optimal thermodynamic ORC model was used to design the heat exchangers of the system. The negative impact of ORC system installation was considered when selecting the final ORC model. The main selection criteria for the optimal ORC model is maximizing network output. It is concluded that using R245fa as the working fluid, without adding a recuperator and superheater, and with the given constraints and inputs, is the optimal choice for this application. This conclusion was reached based on a comprehensive evaluation of the performance and feasibility of different ORC models, considering the specific requirements and limitations of a heavy-duty truck.

Introduction

Heavy-duty trucks are known to consume a significant amount of fuel, contributing to environmental pollution and high operational costs. However, the development of heat recovery systems that can recover waste heat from the engine and exhaust gases can improve the efficiency of heavy-duty trucks and reduce their environmental impact. Among the various heat recovery technologies, the Organic Rankine Cycle (ORC) is considered as a promising option due to its ability to convert low-grade heat into useful power. The ORC system works by using a working fluid, such as water or refrigerants, to absorb heat from the waste heat source and convert it into mechanical or electrical energy. The efficiency of the ORC system depends on the selection of the working fluid, the thermodynamic cycle configuration, and the heat exchanger design. Therefore, choosing the optimal ORC model is crucial for maximizing the efficiency and feasibility of the heat recovery system in heavy-duty trucks.

This paper presents an investigation of the optimal thermodynamic ORC model for a heat recovery system implemented in heavy-duty trucks. Three different working fluids, R245fa, Ethanol, and Cyclopentane, were studied through parametric studies that varied the evaporator pressure. The selected optimal thermodynamic ORC model was used to design the heat exchangers of the system. The negative impacts, such as the installation effect, were considered when selecting the final ORC model. Finally, the paper concludes with the optimal ORC model for this application that accounts for all constraints.

1) Targeted Application for the Heat Source

When it comes to implementing an Organic Rankin Cycle system, ORC, as a waste heat recovery system, selecting the right application is necessary to ensure the effectiveness and efficiency of the system. ORC technology is created to generate electricity by utilizing low-grade heat wasted through an industrial process or power generation, and that cannot be recovered using the normal water Rankin cycle which usually used in high temperature applications. Nevertheless, the choice of the application depends on various factors such as temperature and mass flow rate of the waste heat source, the size and capacity of the system, and availability of suitable cooling system. Failure to choose the right application would result in increasing costs.

The exhaust of Heavy-Duty Diesel engines, HDD, that found in heavy duty trucks relatively has a significant potential for implanting ORC heat recovery system due to the high temperature of the exhaust gases generated during the engine operation. It’s reported that some heavy-duty truck exhaust gases can reach up to 700 C at some high speed and loads [1], making it an ideal source of waste heat that can be recovered and utilized to generate useful work in form electricity. Another reason that makes Heavy duty trucks a good choice is that they often operate over long distances and for extended periods, thus the potential energy savings from utilizing ORC recovery unit can be substantial. For this paper, the choice of the HHD engine is mainly dependent on the availability of information regarding its steady state operational points and the correspondent mass flow rate and temperature of the exhaust. Luckily, [2] conducted an experiment and simulations to obtain, at trainset operational points, the exhaust data of a turbocharged 12.8 L Volvo Diesel engine which is widely used in many heavy-duty Truck such as Volvo VNL series. Table [1] shows the important features of this engine. Also, figure (1) shows the different conditions of the exhaust exit during a driving cycle whereas figure (2) demonstrates 16 steady state operational points at which the thermal analysis of the ORC will be carried out.

Table [1]: Specifications of the engine selected [Reference 2]

Type	Volvo D13 US 2010
Peak Power (W_dot_E)	373 kW
Peak torque	2373 Nm
Compression Ratio	16.0:1
Bore x Stroke	131x 158 mm
Displacement	122.8 L
Aspiration	Turbocharged
Truck installed in	Volvo VNL

Figure [1]: Exhaust exit conditions along a driving cycle [Reference 2]

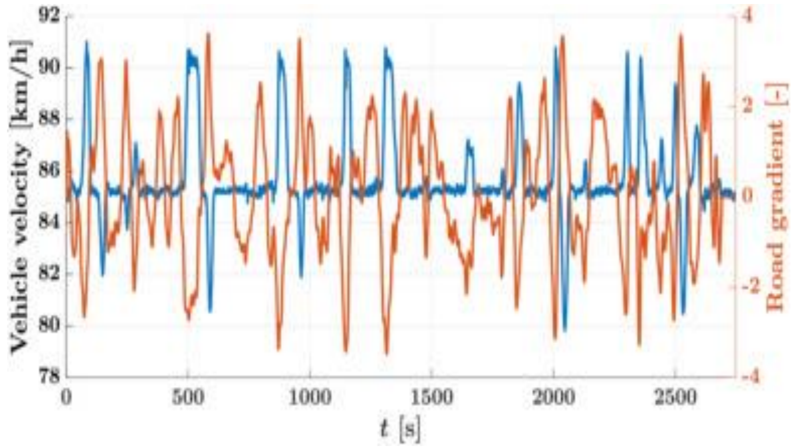
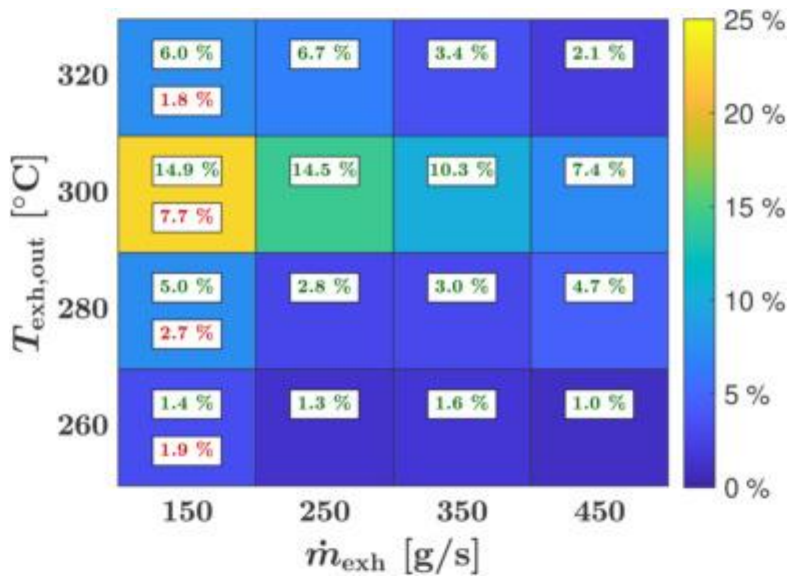


Figure [2]: Steady State operational points of the engine [Reference 2]



The candidate working fluids for the ORC will be compared in the 16 operational points shown in figure (2) but the final optimal thermal model will be chosen based on the point (T_exh_o = 300(C) and mass flow rate of 0.15 kg/s which is the dominant operational conditions among the 16 points.

2) Candidate Working Fluids

There are hundreds of working fluids that can be used in ORC cycle. Therefore, having a set of reasonable selection criteria is a very practical practice to eliminate the low potential working fluids, saving analysis time and cost. Many published papers, such as [3], investigated variety of high promising ORC working fluids based on three main criteria which are the thermodynamic criteria, environmental criteria and Global safety criteria. It is important to select a working fluid that is thermodynamically desirable. For example, the molecular complexity of the working fluid is profoundly reflected in the T-S diagram shape as well as the critical temperature and pressure. The molecular complexity of the working fluid is represented through the heat capacity for which it can be classified into three categories. These categories are dry fluid (positive slope $ds/dT > 1$), Isentropic fluid ($ds/dT = 0$) and wet fluid (negative slope $ds/dT < 0$). Dry and Isentropic fluids are considered to be thermodynamically desired for ORC applications due to their capacity to avoid moisture at the final stage of the expander. Regarding the environmental criteria, a desirable ORC would be the one that has low to zero Ozon Depletion Potential (ODP) and the same for Global Warming Potential (GWP). From which, a good number of working fluids can be eliminated such as R-11 and R-12. Furthermore, the safety of the fluid is a big concern when it comes to automobile applications. The chosen ORC working fluid needs to be not dangerous, especially in the context of flammability and toxicity because the risk of accidents and leakages are very common in automobiles. The paper in [3] came up with a short list of the candidate fluids and then narrowed down to three which all meet the criteria stated above. Those fluids are Cyclopentane, Ethanol and R245fa. Even though the paper suggested the use of Pentane instead of Cyclopentane because of safety reasons, the Cyclopentane has better thermodynamics properties (higher P_{crt} and T_{crt})

than pentane with an only trade-off which is that cyclopentane is mildly flammable. Table [2] summarizes the selected working fluid for the ORC modeling and their correspondent properties.

Table [2]: Selected ORC working fluid and their characteristics

ORC Working fluid	Main characteristics
Ethanol	Slightly wet. Zero ODP & GWP. Available, low cost.
Cyclopentane	Dry. mild flammability Zero ODP & GWP.
R245fa	Isentropic. Zero OPD & GWP. High cost

3) Description of the Overall System and Modeling Plan

Figure [3]: Overall general illustration of the heat recovery system

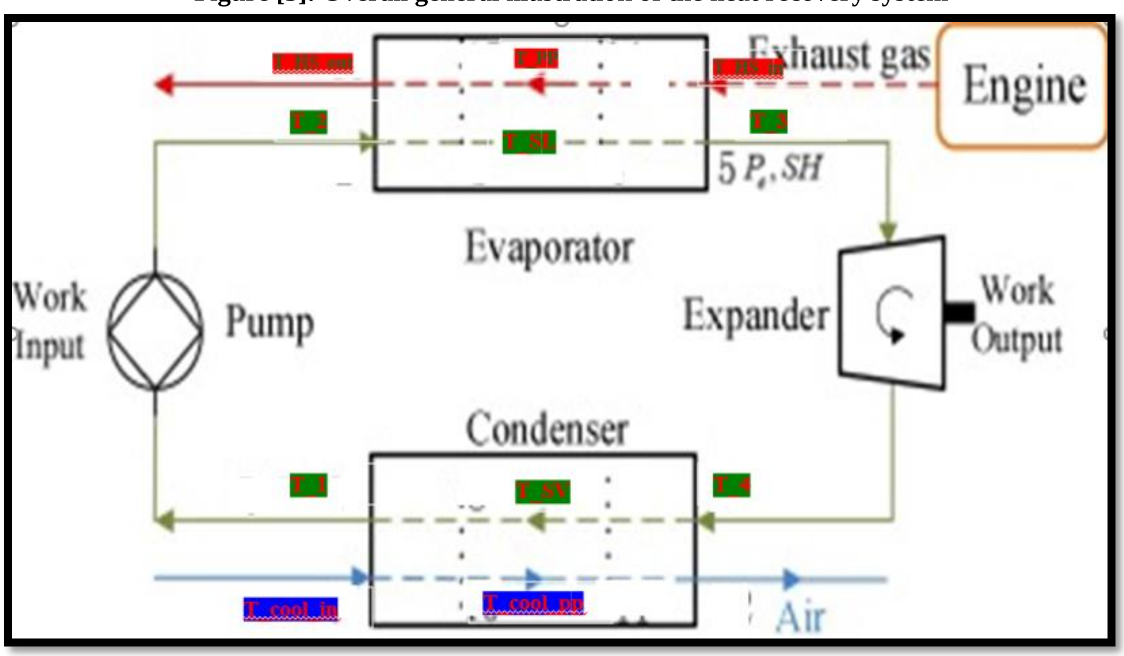
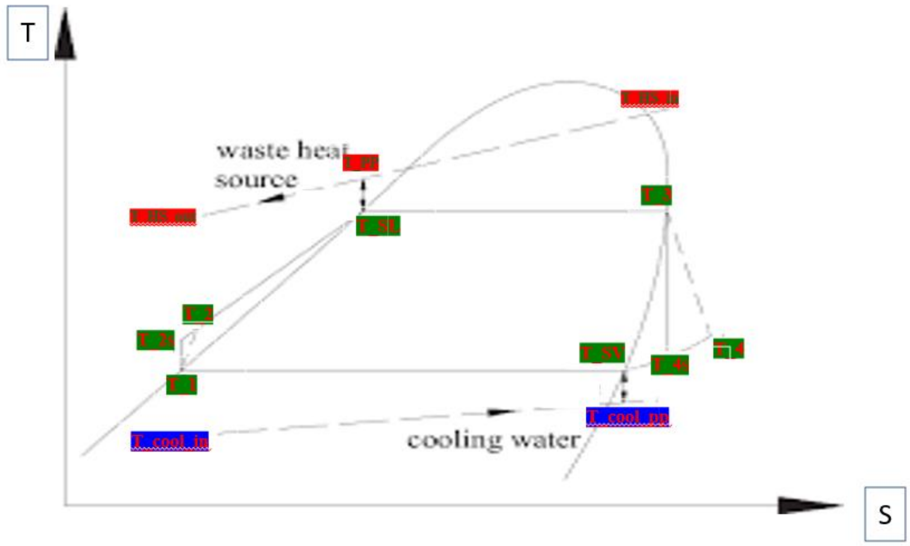


Figure [4]: Overall general T-S diagram of the ORC



Figure[3] illustares the main componets of the heat recovery system , and figure(3) shows the refernence T-S digram of ORC that will be used for modeling. It's worth mentioing that this T-S digram changes for a given working fluid and a given a set cycle modification, e.g., adding a recuprator or superheater. The heat recovery system consists of four main components which are expander, pump , condenser and evaporator. The exhasut exit temperature is donated as T_HS_in and T_HS_out is the tempearure at which the exhasut gases is cooled ddown to after leaving WHR system. T_pp is the pinch point that is associate with the saturated liquid temperatue point in the evaporator. The differnce between the the two points is constrained for a given work fluid at Temperature difference of 5 Celsius as long as the differnce temprature between the temperature of the pump outlet, T_2 and temperature of the heat source is kept at or above 5 Celsius.The later is illustarted in inputs and constrains section. T_3 is the temperature of the expander inlet, whereas T_4 is the temperature of its outlet. T_SV is temperature at the condenser staurate vapor state. T_1 is the temperature of the pump inlet . T_cool_in is the starting temperature of the cooling fan, and T_PP_cool is the pinch point which is associated with T_SV. The latter is constrained for a given working fluids.

4) Inputs and Constrains

Table [3] shows the main inputs and constrains that will be carried during thermodynamic modeling of the ORC

Table [3]: Inputs and Constrains

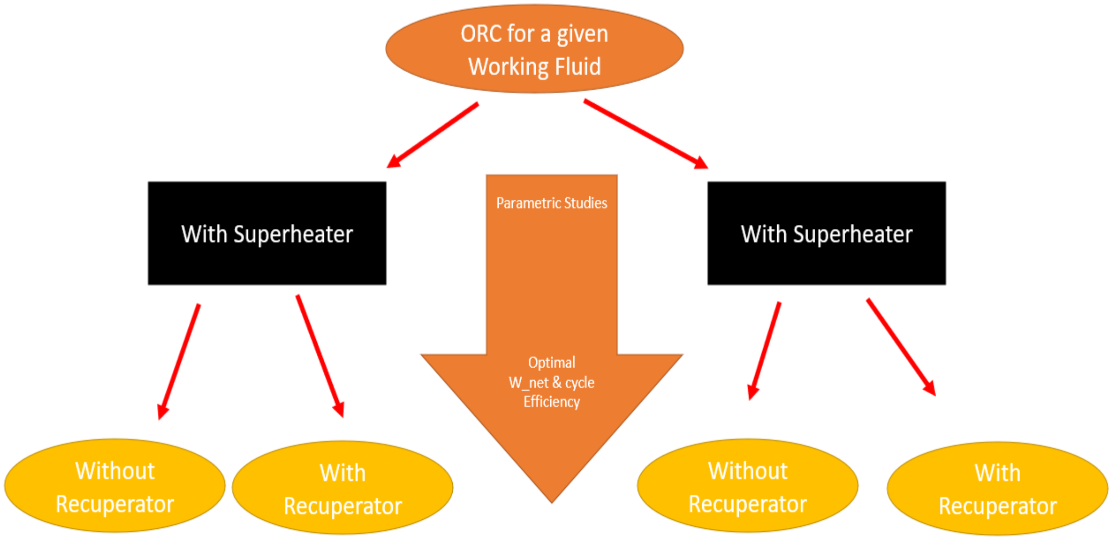
Inputs / Constrain	Notes
Exhaust gases is assumed to be an ideal air	-----
The coolant of the Condenser is air	-----
Assume no pressure drops in the heat exchangers of the WHR system	The energy loss due to the pressure drop in the evaporator and condenser is negligible compared the energy loss due the irreversibility in the expander and the pump
The work required by the cooling fan for the condenser is negligible	It's usually small
Ambient Temperature, T_amb=25 C	-----
Ambient Pressure = 105.25 kPa	-----
Maximum pressure of the evaporator cannot go beyond the critical pressure point of the working fluid	Only the subcritical ORC is considered for this analysis
Maximum temperature of the expander cannot go beyond 260 C	For practical reason such that this is the typical temperature constrain that a commercial expander can handle
Expander's maximum pressure range (1000 kPa to 3090 kPa)	For Practical reasons as well
The minimum pressure of the pump's outlet ranges from 1000 kPa to P Critical	-----
The condenser pressure is fixed at 110 kPa	It's above the ambient pressure and it's fixed for simplicity
The allowable minimum temperature between outlet temperature of the heat sources and the inlet temperature of the preheater is 5 C	-----
The difference temperature between T_pp and T_SL is fixed at 5 C. This is not applied	This constraint holds if the previous constraint is not violated. If does violates,

for R245fa and rather the difference between heat source outlet and inlet temperature of the preheater is fixed at 5 [C]	then a parametric study is required to find the Typical difference between T_pp and T_SL that satisfies this constrain
Minimum temperature between the condenser and cooling fan is 5 C	-----
Pump efficiency = 0.7	-----
Expander Efficiency =0.75	-----
Efficiency of the expander shaft work converted into electricity = 0.9	-----

5) Thermal Modeling and Analysis of the Organic Rankin Cycle

Considering the selected working fluids besides water as a reference, the ORC modeling and analysis were carried out based on the model shown in figure (5). This model gives a comprehensive demonstration of the ORC performance with all possible cycle modifications, considering the constraints stated in the previous section. The perfect performance of the cycle for this application is evaluated based on the model that has the maximum cycle work and cycle thermal efficiency. The work net of the cycle is the preferable criteria over efficiency since the main goal is to produce electricity. The performance of every working fluid is compared with and without adding a superheater. The latter is the main comparison branch. Also, the performance of the cycle is investigated when a recuperator is added and not added simultaneously.

Figure [5]: Evaluation ORC Performance model



The thermodynamics modeling starts off with a basic water Rankin cycle to use as a comparing reference for the examined ORC working fluids. All working fluids are investigated at the 16 operational points shown in figure (2), with condenser pressure fixed at 1000 kPa and without adding a superheater. The purpose of this investigation is to give an overall insight into the potential of every working fluid. Furthermore, the dominant operational point (T_exh= 300C and m_dot_exh = 0.15 kg/s) is then considered to carry out with optimal ORC analysis. The model shown figure (5) is investigated by varying the pressure of the evaporator at the constrained range from 1000 kPa to 3090 kPa, then the best evaporator pressure condition is chosen for the best cycle performance. Knowing that other constrains, such as the minimum allowable temperature between the heat source outlet and preheater inlet is 5 C, must be kept in mind when selecting the optimal evaporating temperature.

5-1) Water

A basic Rankin cycle is constructed using water as the working fluid and without preheater such that T_3 is equal to T_evap. Refer to Appendix-A for full solutions and Results.

- **Process (1-2) for Pump**

$$\dot{W}_{in} = \dot{m}_{steam} \cdot (h_2 - h_1) \qquad h_2 = \frac{h_{2s} - h_1}{\eta_{pump}} + h_1$$

- **Process (2-3) for evaporator (Heat added)**

$$\dot{Q}_{in} = \dot{m}_{steam} \cdot (h_3 - h_2)$$

- **Process (3-4) for Expander (Work Produced)**

$$\dot{W}_{out} = \eta_{mecha,elect} \cdot (h_3 - h_4) \cdot \dot{m}_{steam}$$

- **Process (4-1) for Condenser (Heat Rejected)**

$$\dot{Q}_{out} = \dot{m}_{steam} \cdot (h_4 - h_1)$$

- **Energy Balance between evaporator and the Heat source (HH_in a- T_pp) and (T_3-T_SL) to find the mass flow rate of the working fluid**

$$Q_{3,SL} = Q_{HS,in,HS,pp}$$

$$Q_{HS,in,HS,pp} = \dot{m}_{air} \cdot (h_{HS,in} - h_{HS,pp})$$

$$Q_{3,SL} = \dot{m}_{steam} \cdot (h_3 - h_{SL})$$

- **Energy balance to Find the temperature of the heat source outlet**

$$\dot{Q}_{in} = \dot{m}_{air} \cdot (h_{HS,in} - h_{HS,out})$$

$$h_{HS,out} = cp_{air} \cdot T_{HS,out}$$

- **Work net of the cycle**

$$\dot{W}_{net} = \dot{W}_{out} - \dot{W}_{in}$$

- **Efficiency of the cycle**

$$\eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

- **Maximum available thermal energy from the heat source**

$$\dot{Q}_{HS,max} = \dot{m}_{air} \cdot cp_{air} \cdot (T_{HS,in} - T_{amb})$$

- **Utilization Efficiency**

$$\eta_{utliz} = \frac{\dot{W}_{net}}{\dot{Q}_{HS,max}}$$

5-2) Cyclopentane, Ethanol and R245fa

Note that for full calculations, refer to the EES code that demonstrates the analysis process for Cyclopentane, Ethanol and R245fa which are listed in Appendices B, C and D respectively

- **Energy Balance established to evaluate the mass flow rate of the working fluid**

$$T_{HS,in} = T_{exh,in}$$

$$h_{HS,in} = cp_{air} \cdot T_{HS,in}$$

$$T_{HS,pp} = \delta_{T,pp} + T_{SL}$$

$$h_{HS,pp} = cp_{air} \cdot T_{HS,pp}$$

$$Q_{3,SL} = Q_{HS,in,HS,pp}$$

$$Q_{HS,in,HS,pp} = \dot{m}_{air} \cdot (h_{HS,in} - h_{HS,pp})$$

$$Q_{3,SL} = \dot{m}_{cy} \cdot (h_3 - h_{SL})$$

- **Energy Balance established for the Recuperator**

$$h_4 - h_{4r} = h_{2r} - h_2$$

$$\dot{Q}_{rec} = (h_4 - h_{4r}) \cdot \dot{m}_{cy}$$

- **Process (1-2) Pump**

$$\dot{W}_{in} = \dot{m}_{cy} \cdot (h_2 - h_1)$$

- **Process (2-3) Evaporator & Process (2r-3) Evaporator when recuperator is added**

$$\dot{Q}_{cut,non} = \dot{m}_{cy} \cdot (h_4 - h_1)$$

$$\dot{Q}_{cut,rec} = \dot{m}_{cy} \cdot (h_{4r} - h_1)$$

- **Process (3_4) Expander**

$$\dot{W}_{out} = \eta_{mecha,elect} \cdot (h_3 - h_4) \cdot \dot{m}_{cy}$$

- Net Work of the cycle

$$\dot{W}_{net} = \dot{W}_{out} - \dot{W}_{in}$$

- Thermal efficiency with and without recuperator

$$\eta_{cycle,non} = \frac{\dot{W}_{net}}{\dot{Q}_{in,non}}$$

$$\eta_{cycle,rec} = \frac{\dot{W}_{net}}{\dot{Q}_{in,rec}}$$

6) Thermal modeling Results and Discussion

6-1) Water (the reference cycle)

Appendix [A] shows different parameters at the 16 points operational conditions as well as parametric study conducted to investigate the effect of varying evaporating pressure from 1000 kPa to 3090 kPa. The plot in figure (6) shows how thermal efficiency enhances as the evaporating pressure increases. This is due to the increase in the area enclosed in the T-S diagram. Figure (7) shows the effect of the varying pressure on the W_dot_net which results in a decaying trend unlike the efficiency because the enthalpy of the expander inlet decreases as the T-S diagram shifts toward the lift, and not to mention how the moisture content would dramatically increase at the final stages of the expander, which is another big concern. Also, the utilization factor decreases as well since it’s correlating with the net output power. This parametric study demonstrates the limitations of using the basic Rankin cycle for low grade heat sources. Figure (8) shows the plot of T-S diagram at the optimal evaporating pressure.

Table [4]: Optimal Configuration for water

Optimal configuration	No Superheater, No Recuperator, P_evp = 1422 kPa . W_net_cycle = 2.298kW Eta_cycle= 12.4 %

Figure [6]: Effect of varying P_evp in efficiency (Water)

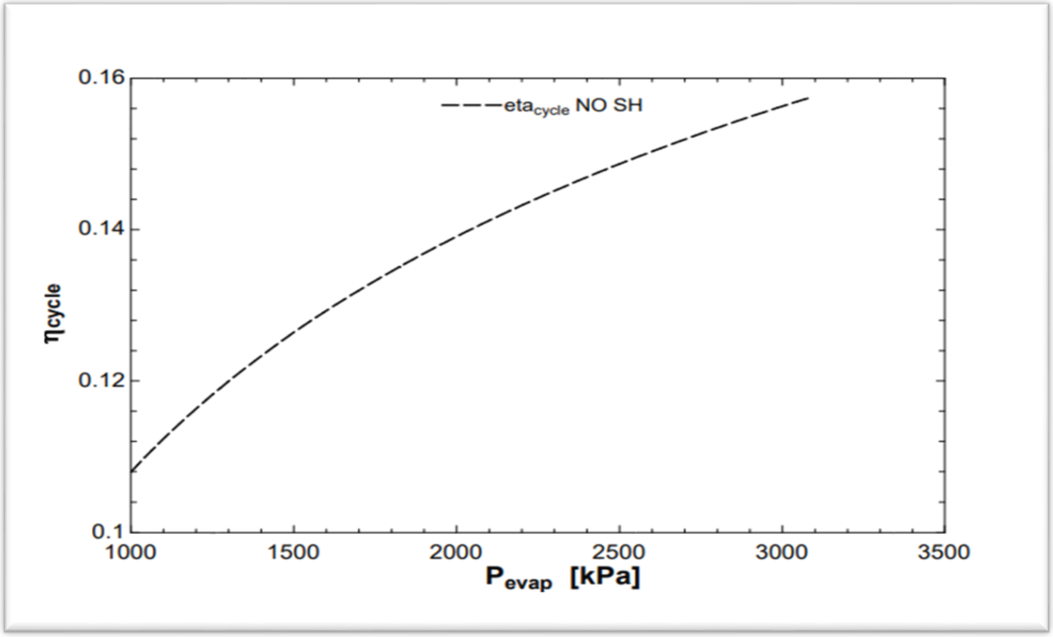


Figure [7]: Effect of varying P_evp on W_net_cycle (Water)

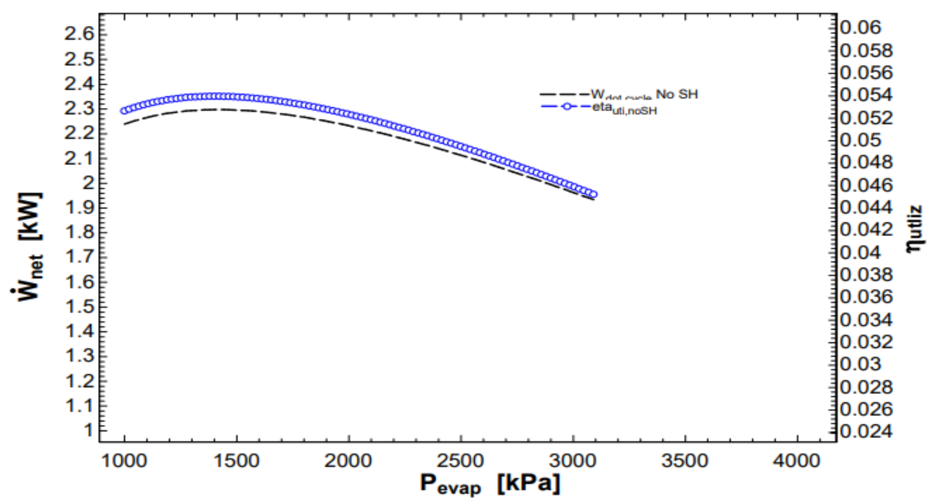
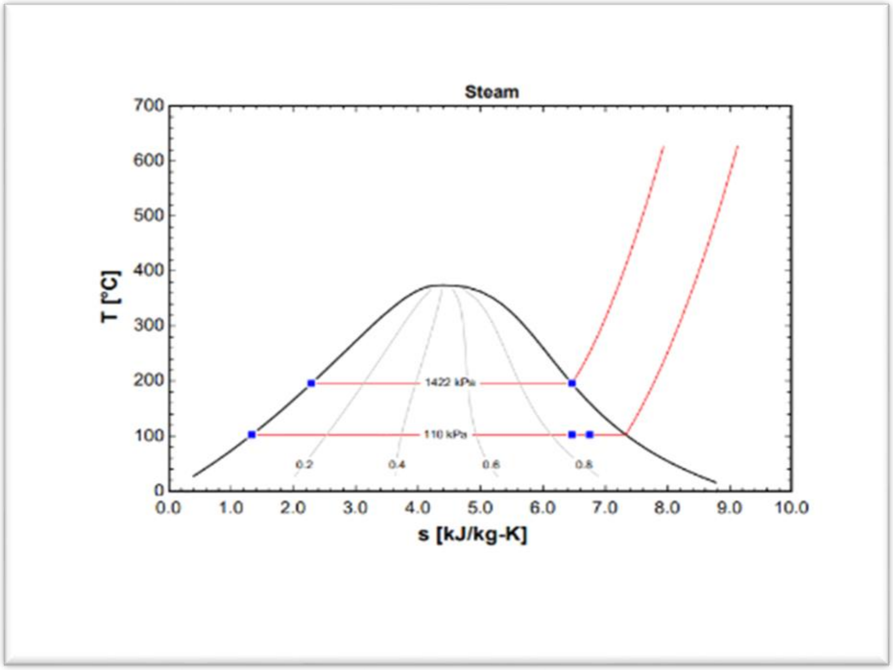


Figure [8]: T-S diagram at optimal P_evp (Water)



6-2) Cyclopentane

Appendix [B] shows different parameters at the 16 points operational conditions and shows a parametric study conducted to investigate the effect of varying evaporating pressure from 1000 kPa to 3090 kPa without and with superheater respectively.

Figure (9) shows the effect of the varying evaporating pressure on the thermal efficiencies based on the model described in figure (5). Adding a recuperator, in general, increases thermal efficiency because it reduces the required heat added in the evaporator even though it does not affect the work net produced by the cycle. Cyclopentane is a suitable working fluid to add a recuperator to since it has a positive slope ($dT/ds > 1$). Adding both a recuperator and superheater would even increase the efficiency further, but it almost does not affect the efficiency of the non-recuperator configuration.

Figure [9]: Effect of varying P_evp in efficiencies (Cyclopentane)

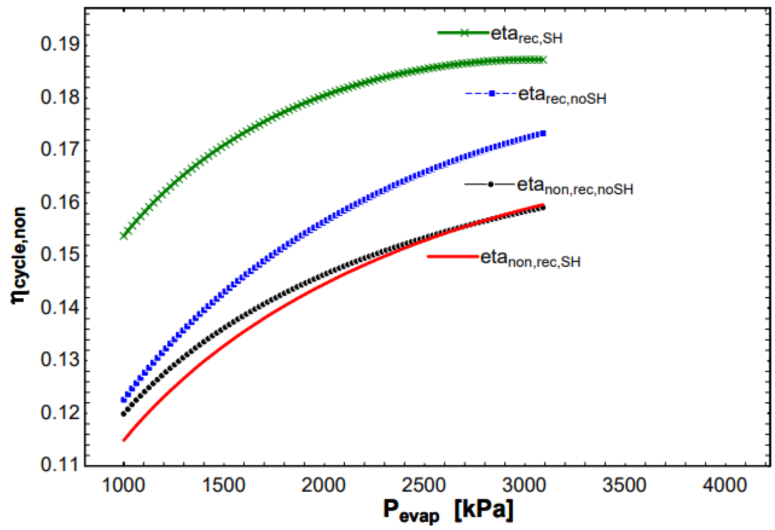
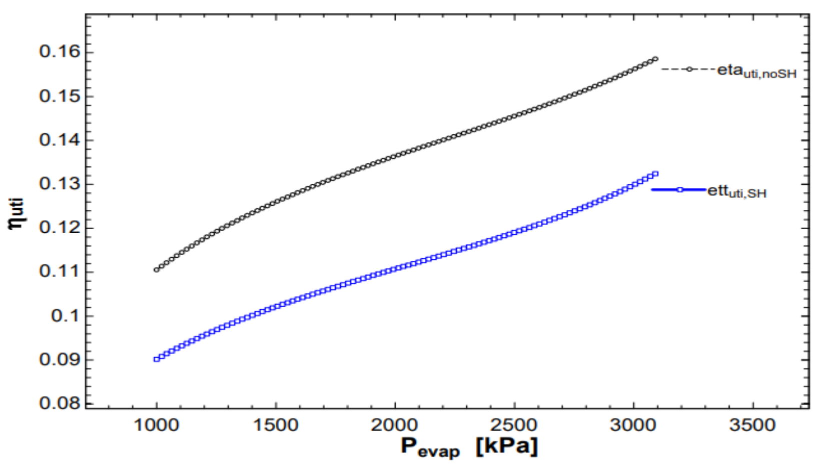


Figure [10]: Effect of varying P_evp on utilization factor (Cyclopentane)



However, adding a superheater in general would result in having a less amount of work produced compared to the configuration without a superheater. Figure (10) shows a comparison between the utilization factor when a superheater is and not added. The latter is correlated with work net produced by the cycle. The effect of increasing the evaporating pressure would enhance the overall performance of the cycle in terms of the work produced and thermal efficiency; however, a careful look was taken to the possibility of violating the constrain regarding the allowable minimum Temperature between heat source outlet and preheater inlet. It was found that despite (P_evp=3090 kPa) would result in having the maximum work possible, it violates the constraint. A modification must be implemented in the pinch difference temperature in the evaporator side. Another parametric study was conducted to address this concern which is shown in appendix B. The next valid pinch difference point was then obtained at 15 C. The analysis was redone at this new pinch difference temperature. Two optimal thermodynamics configurations were obtained which are elaborated in table [5].

Table [5]: Optimal Configuration for Cyclopentane

Optimal configuration (1)	No Superheater , No Recuperator, P_evp = 3090 kPa . W_net_cycle = 5.04 kW Eta_cycle= 11.69 %
Optimal configuration (2)	No Superheater , with Recuperator, P_evp = 3090 kPa . W_net_cycle = Eta_cycle= 16.77%

Despite the better efficiency with recuperator configuration, it may result in having higher negative installation impact which will be addressed in the upcoming sections. Figure (11) and (12) show the T-S diagram at the optimal evaporating pressure.

Figure [11]: T-S diagram at optimal P_evp -without Superheater (Cyclopentane)

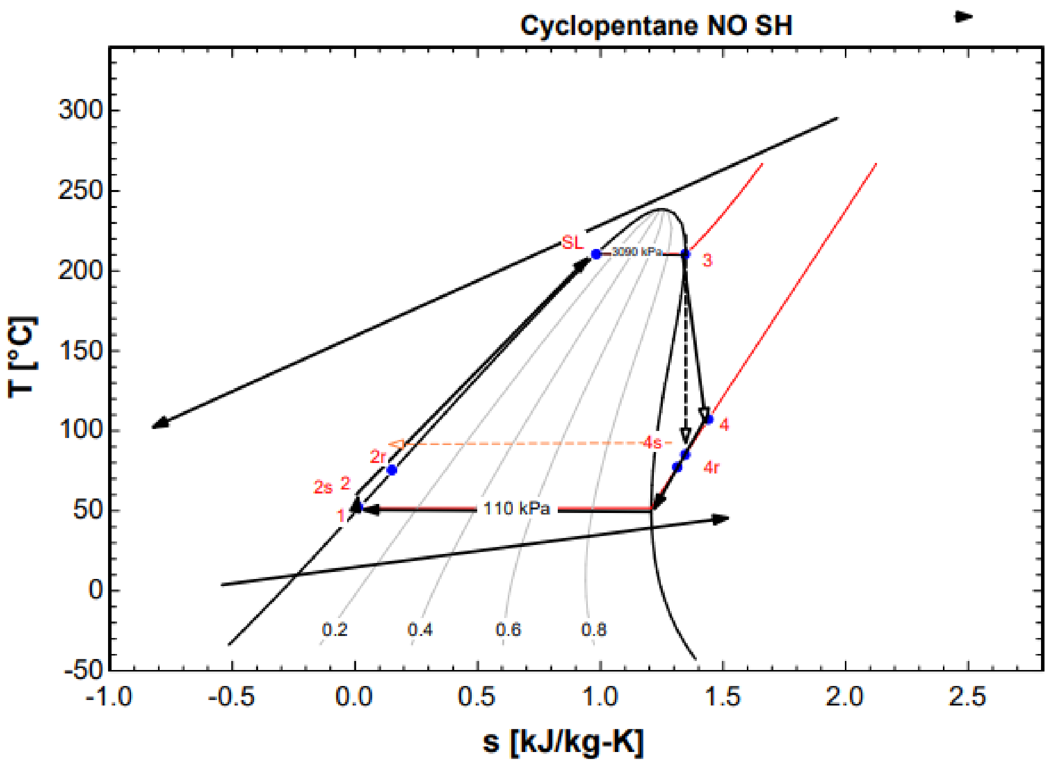
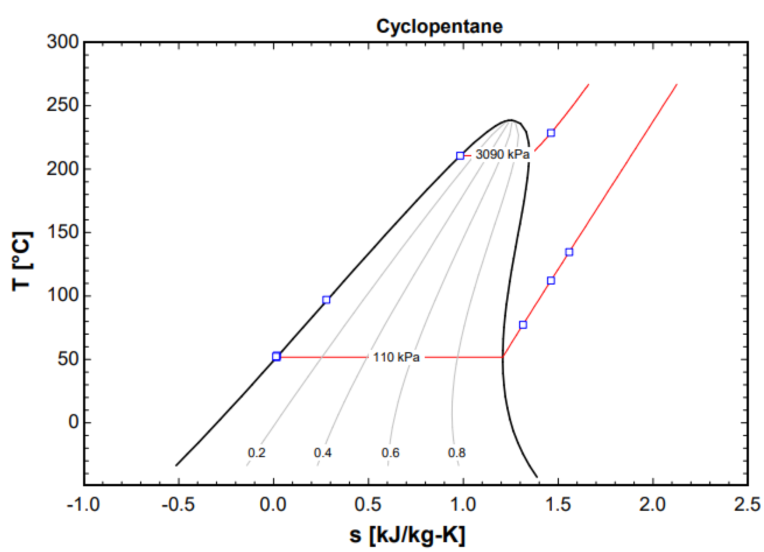


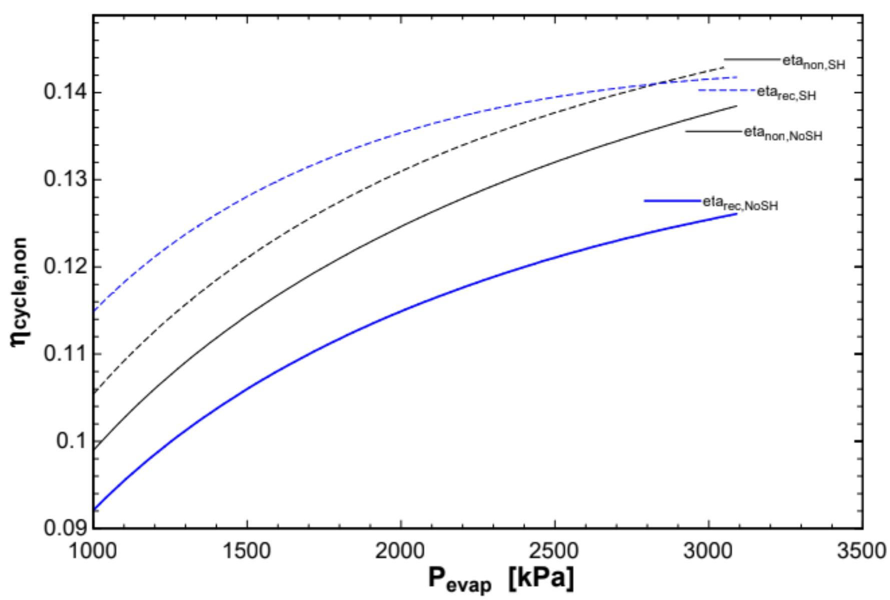
Figure [12]: T-S diagram at optimal P_ev -with Superheater (Cyclopentane)



6-3) Ethanol

Appendix [C] shows different parameters at the 16 points operational conditions and shows a parametric study conducted to investigate the effect of varying evaporating pressure from 1000 kPa to 3090 kPa without and with superheater respectively

Figure [13]: Effect of varying P_evp in efficiencies (Ethanol)



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Figure[13] shows the effect of varying evaporating pressure on the thermal efficincies based on the modifications shown in figure(5). It's noticed that adding superheater would improve the efficiency. On the other hand , adding a recuperator is not practical as it would decrease the efficiency, especially the model without superheater. This is a result of the thermodynamic properties of ethanol as it is a semi wet working fluid which does not have that much of supereating content at the condenser's inelt, unlike the dry working fluid cyclopentane. Figure[14] shows the effect of the varying P_evp on the net work cycle for the model with and without the superheater. It shows that adding a superheater is not practical for this application even though it would increase the overall efficiency since the priorirty is to produce the highest possible work net of the cycle. The opitaml thermodynamics configuration is summirized in table[6]:

Table [6]: Optimal Configuration for Ethanol

Optimal configuration	No Superheater , No Recuperator, P_evp = 3090 kPa . W_net_cycle = 3.898 kW Eta_cycle= 13.89 %

Figure(15) shows the T-S digram of ethanol at the optimal configration conditions. It's noticed how adding a recuprator is not promising as there is a minimal amount of superheating working fluid content at the condenser's inelt.

Figure [14]: Effect of varying P_evp in net output power (Ethanol)

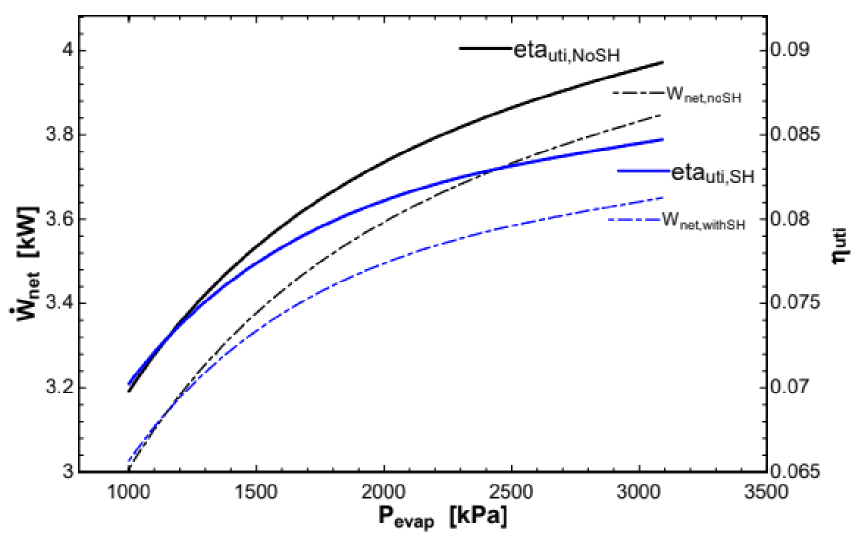


Figure [15]: Effect of varying P_evp in efficiencies (Ethanol)

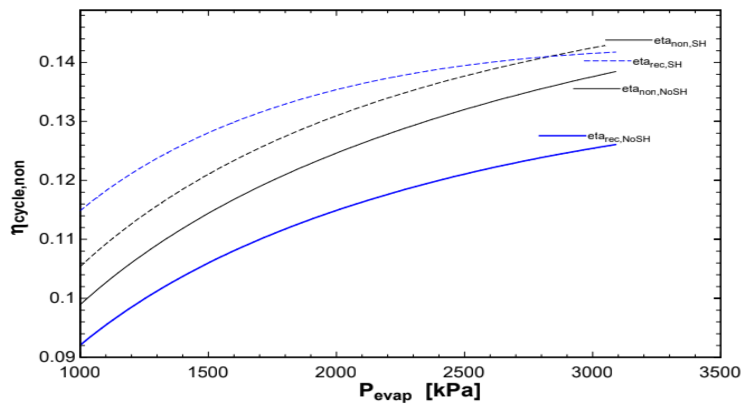
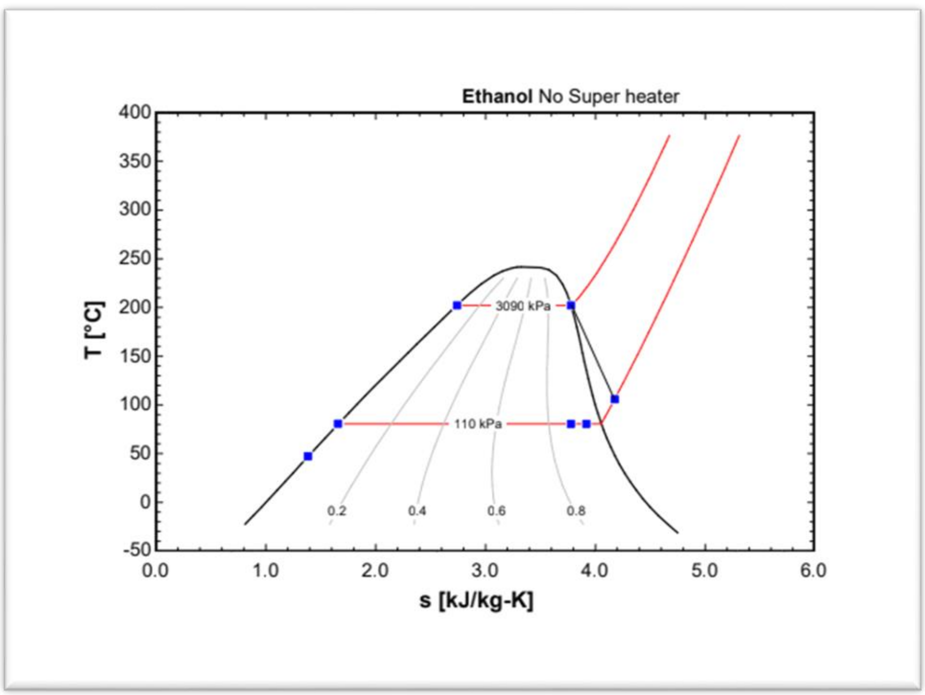


Figure [16]: T-S diagram of ethanol at optima P_evp (Ethanol)



6-4) R245fa

Appendix [D] shows different parameters at the 16 points operational conditions and shows a parametric study conducted to investigate the effect of varying evaporating pressure from 1000 kPa to 3090 kPa without and with superheater respectively. Figure (17) shows the effect of increasing the evaporating pressure on the efficiencies. Adding a recuperator to the configuration that does not involve a superheater would increase efficiency slightly. Adding a superheater to the recuperating model would increase the efficiency dramatically to a certain pressure value which is 3006 kPa but then a sudden drop would be experienced. The same thing is noticed for the works net of the cycle as shown in figure (18). Thus, adding a superheater is not the right path to go since the low critical temperature of the refrigerant limits the room for playing with the expander's nlet value. Figure (19) shows the plot of T-S diagram for R245fa at optimal selected conditions which elaborated in table [7]

Table [7]: Optimal Configuration for Ethanol

Optimal configuration	No Superheater , No Recuperator, P_evp = 3090 kPa . W_net_cycle = 6.265 kW Eta_cycle= 14.56 %

Figure [17]: Effect of varying P_evp in efficiencies (R-245fa)

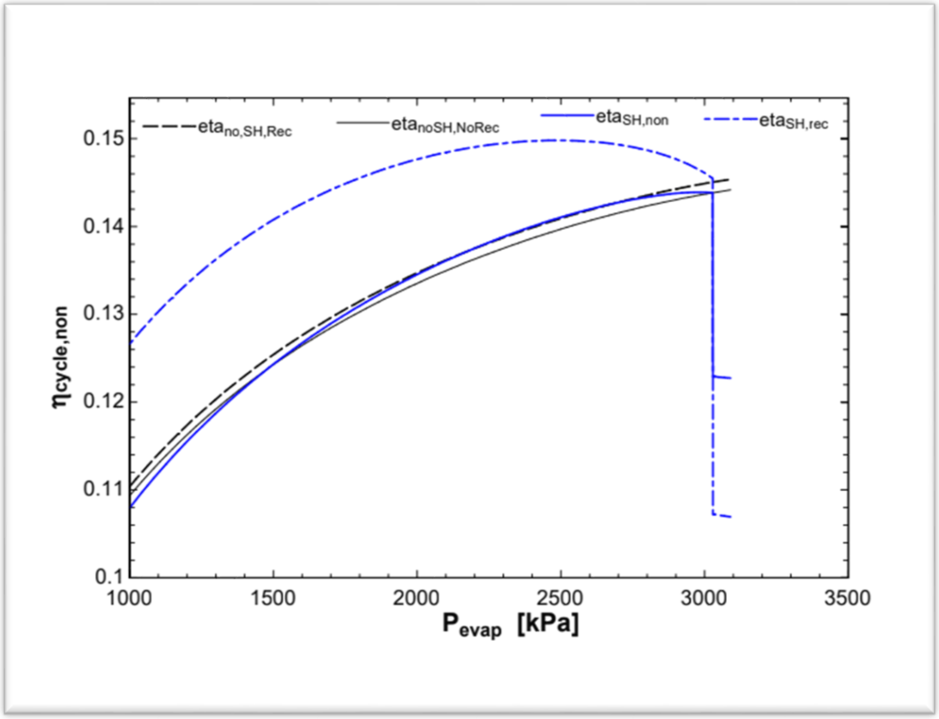


Figure [18]: Effect of varying P_evp in net output power (R-245fa)

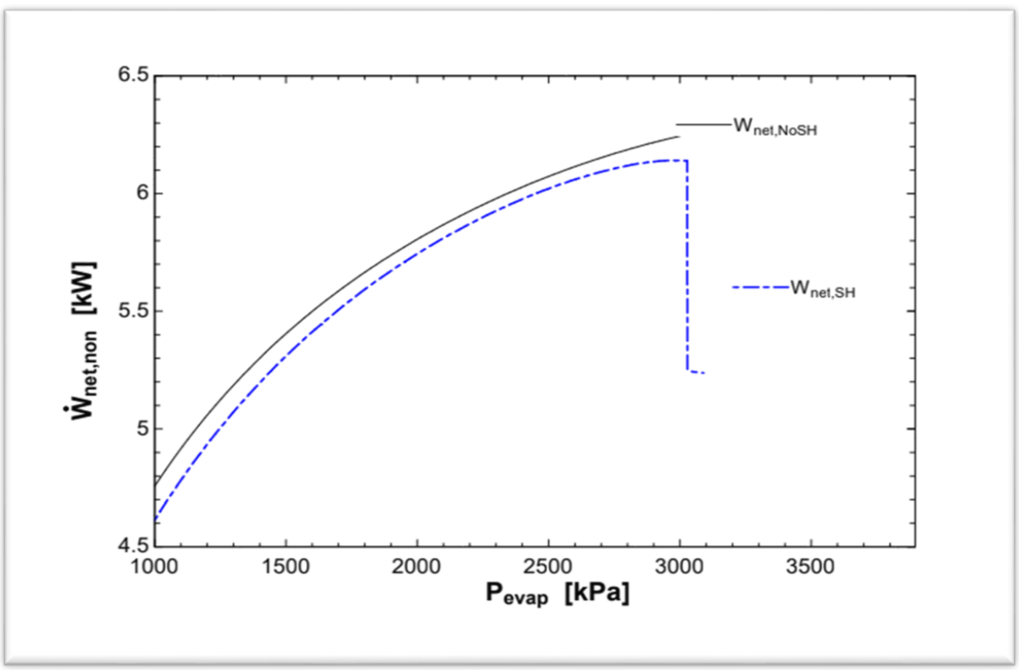
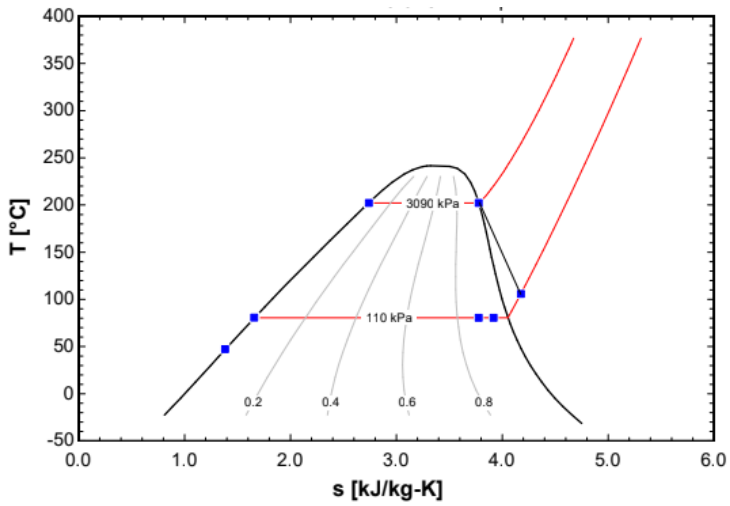


Figure [19]: T-S diagram of R245fa at optimal P_evp (Ethanol)



6-5) Summary of overall results

Table [8] shows the overall performance of ORC using different working fluids at the 16 engine operation points. The results give a sense that cyclopentane and R245fa has the most promising organic fluids for the specific given constrains

Table [8]: summery of the results at 16engine operational conditions

Working Fluid	W_net_cycle (kW)	Eta_cycle
Water	1.345 to 8.573	3.7 to 6.2
Cyclopentane	3.74 to 18.91	13.65
Ethanol	2.156 to 10.32	9.89
R245fa	4.107 to 15.63	10.83

Furthermore, Table [9] elaborates the optimal thermodynamics configuration for each working fluid.

Table [9]: Optimal Configurations for each working fluid

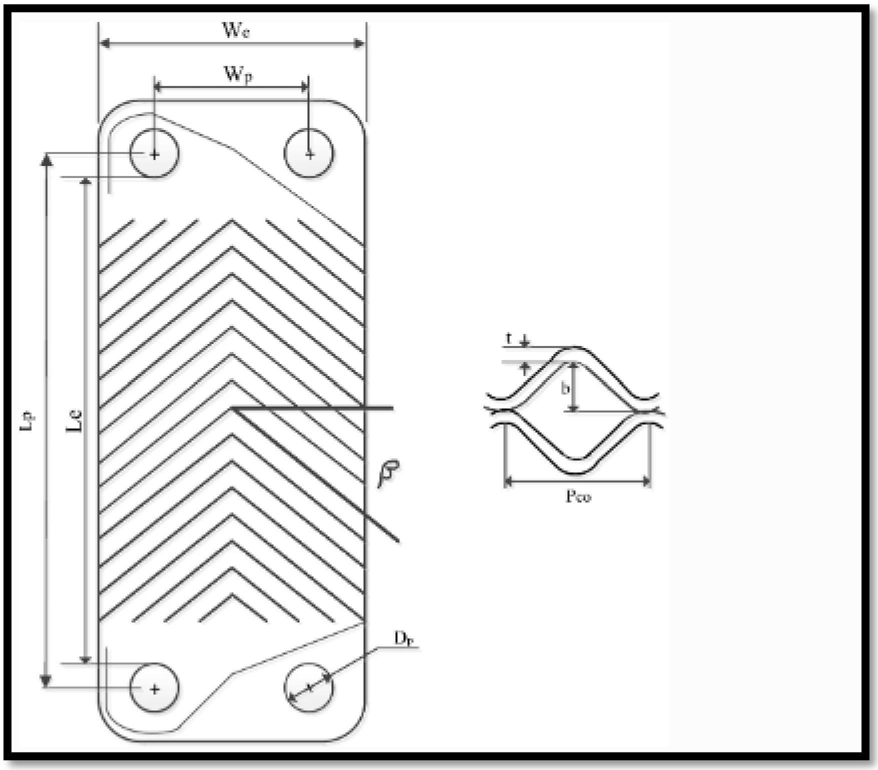
Working Fluid	Description of the thermodynamic model based on Figure (5)	P_evp[kPa]	W_net[kW]	Thermal Efficiency
Water (Reference)	No Preheater & No recuperator	1422	2.298	12.4
Cyclopentane	No Preheater & No recuperator	3090	5.04	11.64
	No Preheater & With recuperator	3090	5.04	16.77
Ethanol	No Preheater & No recuperator	3090	3.898	13.89
R245fa	No Preheater & No recuperator	3090	6.205	14.56

As stated previously, the criteria selection for the working fluid is based on producing the maximum work net of the cycle, and this criterion will be considered over the efficiency. R245fa is the primary selected working fluid. Cyclopentane with its two configurations will be considered since its work net cycle is slightly smaller than that in R245fa. Another reason is that R245fa may yield to be the best option after considering the negative installation impact.

7) Heat Exchangers Design for the Selected working fluid

The ORC heat recovery system that will be implemented consists mainly of the evaporator and condenser. The evaporator will be designed in two zones which are the superheating zone (zone a) and the phase change zone (zone b). For condenser, the phase change zone is the only focus. Also, a design for the recuperator will be given to address its installation impact. The type of heat exchanger that is going to be selected is a Chevron type heat exchanger which has a V-shape pattern known as herringbone. This is a very practical type of heat exchanger due to its compact size, which is needed for automobile applications. Figure (20) shows the scheme of this heat exchanger [4].

Figure [20]: scheme of selected heat exchanger type



The primary geometry specifications, e.g., W_e, W_p , L_p, are chosen such that there are commercially available (check out [5]). The design process is carried out using LMTD method to find the total number of plates required for all heat exchangers. The total number of plates required is obtained for the two optimal thermodynamics ORC for cyclopentane and for the optimal thermodynamics cycle for R245fa. The design process is illustrated in Appendix[E] for Cyclopentane and Appendix [F] for R245fa. The Nusselt number correlations used are obtained from [7].

7-1) Overall Results of the Heat exchangers design

Table [10]: Heat exchanger results

Working Fluid	Total surface area without recuperator[m^2]	Total number of Plates without Recuperator	Total surface area with recuperator[m^2]	Total Number of plates with Recuperator
Cyclopentane	**28.41**	**248**	**33.96**	**296**
R245fa	**11.33**	**100**	**30.81**	**269**

As shown in the table, adding a recuperator will increase the size of the heat exchangers and this increase is correlated with an increase of total cost and the negative installation impact which will be addressed in the next section. Moreover, it's noticed that R245fa requires a smaller size heat exchanger

7-2) Installation Effect of the WHR system

It's extremely necessary to address the impact of the added weight from the ORC heat recovery system that is going to be installed. Even though the thermodynamics analysis of the cycle may show a good overall performance, its installation impact would result in decreasing the net power produced by the cycle. The net output of the of the heat recovery system when considering the negative impact is given as follows:

W_dot_net = W_ dot_net (before considering the negative impact) minus W_dot_orc, where W_dot_orc is the increased engine load caused by the increased weight due to the installation of ORC system which is given as follows:

$$\dot{W}_{orc,w,non} = \frac{0.04 \cdot \dot{W}_{engine} \cdot M_{orc,non}}{0.1 \cdot M_{vehicle}}$$

Where W_dot engine is the engine power which is equal to 373 kW, and M_vehicle is the total mass of the vehicle which is 36278 kg (about 79979.2 lb). M_org is the total mass of the ORC system which includes the expander, pump and the heat exchangers. [6] provides mass correlations to give an estimate of those components' masses.

7-3) Calculation of installation's negative impact for Cyclopentane

$$\rho_{pp} = 7500 \quad [kg/m^3]$$

$$v_{pp} = 0.00027071 \quad [m^3]$$

$$M_{pp} = \rho_{pp} \cdot v_{pp}$$

$$N_{p,total,non} = 248$$

$$N_{p,total,rec} = 296$$

$$M_{hx,non} = N_{p,total,non} \cdot M_{pp}$$

$$M_{hx,rec} = N_{p,total,rec} \cdot M_{pp}$$

$$M_p = 1.0764 \cdot \dot{W}_{in} + 1.8022$$

$$M_{exp} = 0.3448 \cdot \dot{W}_{out} + 6.4655$$

$$M_{orc,non} = M_{hx,non} + M_p + M_{exp}$$

$$M_{orc,rec} = M_{hx,rec} + M_p + M_{exp}$$

$$\dot{W}_{engine} = 373 \quad [kW]$$

$$M_{vehicle} = 36278 \quad [kg]$$

$$\dot{W}_{orc,w,non} = \frac{0.04 \cdot \dot{W}_{engine} \cdot M_{orc,non}}{0.1 \cdot M_{vehicle}}$$

$$\dot{W}_{orc,w,rec} = \frac{0.04 \cdot \dot{W}_{engine} \cdot M_{orc,rec}}{0.1 \cdot M_{vehicle}}$$

$$W_{cycle,non,NE} = \dot{W}_{net} - \dot{W}_{orc,w,non}$$

$$W_{cycle,rec,NE} = \dot{W}_{net} - \dot{W}_{orc,w,rec}$$

7-34 Calculation of installation's negative impact for R245fa

$\rho_{pp} = 7500$ [kg/m³]

$v_{pp} = 0.00027071$ [m³]

$M_{pp} = \rho_{pp} \cdot v_{pp}$

$N_{p,total,non} = 100$

$N_{p,total,rec} = 269$

$M_{hx,non} = N_{p,total,non} \cdot M_{pp}$

$M_{hx,rec} = N_{p,total,rec} \cdot M_{pp}$

$M_p = 1.0764 \cdot \dot{W}_{in,non} + 1.8022$

$M_{exp} = 0.3448 \cdot \dot{W}_{out,non} + 6.4655$

$M_{orc,non} = M_{hx,non} + M_p + M_{exp}$

$M_{orc,rec} = M_{hx,rec} + M_p + M_{exp}$

$\dot{W}_{engine} = 373$ [kW]

$M_{vehicle} = 36278$ [kg]

$$\dot{W}_{orc,w,non} = \frac{0.04 \cdot \dot{W}_{engine} \cdot M_{orc,non}}{0.1 \cdot M_{vehicle}}$$

$$\dot{W}_{orc,w,rec} = \frac{0.04 \cdot \dot{W}_{engine} \cdot M_{orc,rec}}{0.1 \cdot M_{vehicle}}$$

$$W_{cycle,non,NE} = \dot{W}_{net,non} - \dot{W}_{orc,w,non}$$

$$W_{cycle,rec,NE} = \dot{W}_{net,non} - \dot{W}_{orc,w,rec}$$

7-5) Results and conclusion from installation impact analysis

Table [11]: Installation effect

	Before Considering Installation Impact		After considering the Negative Impact	
Working Fluid	Output power without Recuperator [kW]	Output power With Recuperator kW]	Output power Without Recuperator [kW]	Output power With Recuperator [kW]
Cyclopentane	5.822	5.822	3.707	3.306
R245fa	6.265	6.265	5.385	3.974

It's noticed that adding a recuperator would cut down a significant amount of the power output for both working fluids. Also, it can be concluded that R245fa without the Recuperator would result in the least negative impact of the system installation. Therefore, the optimal design for this ORC system is shown in table [12]

Table [12]: Specification of the optimal selected ORC model

Working Fluid	R245fa
Description of the thermodynamic model based on Figure (5)	No superheater, No recuperator
Evaporating pressure	3090 kPa
Total Surface Area of the heat Exchangers	11.33 m^2
Total Number of Plates required	100
The Net Output power	5.385 kW
Thermal Efficiency of the cycle	14.56%

Statement of Uncertainty

There are many sources of errors that might have altered results if they were considered. One of them may be the low accuracy of the Nusselt number correlations used to evaluate the heat transfer coefficients. Another source of error that would have impacted on the final net output power is not considering the negative impact of the backpressure of the engine, which would increase in the presence of ORC system.

Conclusion

The main purpose of this paper is to investigate the optimal thermodynamics ORC model to be a working principle for a heat recovery system implemented in a heavy-duty truck. Three different working fluids are investigated through many parametric studies, such as varying the evaporator pressure. The selected optimal thermodynamic ORC model was used to design the heat exchangers of the system. Also, the negative impact was taken into account when selecting the final ORC model. It's concluded that using R245fa as the working fluid without adding a recuperator and superheater, and with the given constraints and inputs, is the optimal choice for this application.

References

[1]:https://www.sciencedirect.com/topics/engineering/exhaust-gas#:~:text=High%20loads%20and%20high%20speeds,C%20(788%C2%B0F).

[2]: https://www.sciencedirect.com/science/article/pii/S0360544221019460#fd35

[3]: https://www.sciencedirect.com/science/article/pii/S1359431115003774#tbl5

[4]: https://link.springer.com/article/10.1007/s00231-018-2446-8

[5]: https://www.dudadiesel.com/choose_item.php?id=HX115060

[6]: https://www.sciencedirect.com/science/article/pii/S1359431120331276#b0195

[7]: https://link.springer.com/article/10.1007/s00231-018-2446-8

Appendix A: Water Rankin Cycle Modeling

```
//Basic Rankin Cycle_ Steam

"Inputs"
P_evap = 1422[kPa]//  solution is initionlized at 10 bar, then the optimal value P_evap choosen out of the range (1000 to 3090 kPa)
P_cond=   110 [  kPa] // condensing pressure is kept constant at 1.1 bar for all cases
T_exh_in= 300[C] // solution is initilized as this temp which repesents the dominant ehasut temp, parametric study for P_evap is done at this point
P_exh_in = 103 [kPa] //solution is initilized as this temp which repesents the  dominant ehasut pressure , P_evap parametric study is done at this point
T_amb =  25 [C]
P_amb = 105.25 [kPa]
Delta_T_pp = 5 [C] // delta T between SL temp and pintch point HS
Delta_T_pp_cond = 5 [C]  // delta T betwen SV point in condenser and coolant pinch point
m_dot_air=  0.15 [kg/s]      // solution is initilized as this mass flow rate which represents  the  dominant exhaust pressure , P_evap parametric study is done at this point
bulk_HS_T =( T_exh_in +T_HS_out )/2    //c_p is taken at mean tempearure of the heat source
Delta_T_min_r = 25 [C] // mimimum temprature difference in the recuporator

cp_air =Cp(Air,T=bulk_HS_T) //varying input  depends on Delta_HS_T
eta_pump= 0.7
eta_expander= 0.75
eta_mecha_elect= 0.9
"Working Fluid: Water"
"state1"
 P_1 = P_cond
x_1 = 0
s_1=entropy(Steam,P=P_1,x=x_1)
h_1=enthalpy(Steam,P=P_1,x=x_1)
T_1= temperature(Steam,P=P_1,x=x_1)



"state _2s"
P_2s= P_evap
s_2s= s_1
h_2s= enthalpy(Steam,P=P_2s,s=s_2s)
T_2s = temperature(Steam,P=P_2s,s=s_2s)
"State 2"
P_2 = P_evap
 h_2 = ( ( h_2s-h_1)/eta_pump )+h_1
T_2 = temperature(Steam,P=P_2,h=h_2)
s_2=entropy(Steam,T=T_2,P=P_2)

"State SL"
 P_SL= P_evap
x_SL= 0
T_SL = temperature(Steam,x=x_SL,P=P_SL)
s_SL=entropy(Steam,P=P_SL,x=x_SL)
 h_SL = enthalpy(Steam, P=P_SL,x=x_SL)

"State 3"
P_3= P_evap
T_3= temperature(Steam,P=P_3, x= 1 )
s_3=entropy(Steam,P=P_3, x=1)
h_3 =enthalpy (Steam,P=P_3, x=1)

//"State 3" for ORC  analysis
//P_3 = P_evap
//T_3 = 160[C]
//s_3 = entropy( Steam, P= P_3, T= T_3)
//h_3=  enthalpy (Steam,P=P_3, T = T_3)


"State 4s"
 P_4s= P_cond
s_4s= s_3
 x_4s=quality(Steam,P=P_4s,s=s_4s)
```

```
" State 4"
 h_4= - (  (eta_expander*( h_3-h_4s) ) - h_3 )
P_4= P_cond
T_4 = temperature(Steam,P=P_4,h=h_4)
x_4 = quality(Steam,P=P_4,h=h_4)
s_4 = entropy(Steam,P=P_4,x=x_4)
/////////////////////////////////////////////////////////////////////////////////////////////////
// if Recuperator added, only for Oragnic working fluids
"State 4r"
//P_4r = P_cond
//T_4r = Delta_T_min_r + T_2
//h_4r = enthalpy ( staem, P=P_4r, T= T_4r)
//" State 2r"
//h_4 - h _4r = h_2r- h _2
//P_2r= P_evap
//T_2r = temperature ( staem, P= P_2r, h= h_2r)

/////////////////////////////////////////////////////////////////////////////////////////////////

"HS_in "
T_HS_in = T_exh_in
 h_HS_in = cp_air*T_HS_in

"HS_pp"
 T_HS_pp=Delta_T_pp+T_SL
h_HS_pp= cp_air*T_HS_pp

"Enegy Balance Evaporator & Heat Source"
Q_3_SL= Q_HS_in_HS_pp
Q_HS_in_HS_pp= m_dot_air*(h_HS_in-h_HS_pp)
Q_3_SL= m_dot_steam * (h_3-h_SL)

"heat added to cycle"
Q_dot_in=m_dot_steam *(h_3-h_2)

"heat rejected through condenser"
Q_dot_out= m_dot_steam*(h_4-h_1)

"Work requred by the pump"
W_dot_in= m_dot_steam*(h_2-h_1)

"Useful work produced by the expander"
W_dot_out=eta_mecha_elect *(h_3-h_4)*m_dot_steam
"back work ratio"
bwr = W_dot_in/W_dot_out

"Net Work output of the cycle"
W_dot_net= W_dot_out - W_dot_in

"Thermal Efficiency of the cycle"
eta_cycle = W_dot_net / Q_dot_in

"HS_out"
Q_dot_in = m_dot_air *(h_HS_in-h_HS_out)
h_HS_out= cp_air *T_HS_out

" Maximum availblethermal energy from the heat source (heat source energy content)"
Q_dot_HS_max = m_dot_air *cp_air*(T_HS_in-T_amb)
" Utilization Fcator "
eta_utliz = W_dot_net / Q_dot_HS_max
// Data for array tables :

//T[1]=  T_1
 //s[1]=s_1
```

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$bulk_{HS,T}$ = 240.2 [C]
cp_{air} = 1.032 [KJ/kg-C]
δT_{pp} = 5 [C]
η_{cycle} = 0.1236
$\eta_{mecha,elect}$ = 0.9
η_{utliz} = 0.05379
h_2 = 430.8 [kJ/kg]
h_3 = 2789 [kJ/kg]
h_{4s} = 2354 [kJ/kg]
$h_{HS,out}$ = 186.1 [kJ/kg]
h_{SL} = 833.2 [kJ/kg]
$\dot{m}_{steam}$ = 0.007853 [kg/s]
P_2 = 1422 [kPa]
P_3 = 1422 [kPa]
P_{4s} = 110 [kPa]
P_{cond} = 110 [-kPa]
$P_{exh,in}$ = 103 [kPa]
$Q_{3,SL}$ = 15.36 [kW]
$\dot{Q}_{in}$ = 18.52 [kW]
$Q_{HS,in,HS,pp}$ = 15.36 [kW]
s_2 = 1.335 [kJ/kg-K]
s_3 = 6.462 [kJ/kg-K]
s_{4s} = 6.462 [kJ/kg-K]
T_1 = 102.3 [C]
T_{2s} = 102.4 [C]
T_4 = 102.3 [C]
T_{amb} = 25 [C]
$T_{HS,in}$ = 300 [C]
$T_{HS,pp}$ = 200.8 [C]
$\dot{W}_{in}$ = 0.01538 [kW]
$\dot{W}_{out}$ = 2.305 [kW]
x_4 = 0.904
x_{SL} = 0

bwr = 0.006672
$\delta T_{min,r}$ = 25 [C]
$\delta T_{pp,cond}$ = 5 [C]
$\eta_{expander}$ = 0.75
η_{pump} = 0.7
h_1 = 428.8 [kJ/kg]
h_{2s} = 430.2 [kJ/kg]
h_4 = 2463 [kJ/kg]
$h_{HS,in}$ = 309.6 [kJ/kg]
$h_{HS,pp}$ = 207.2 [kJ/kg]
$\dot{m}_{air}$ = 0.15 [kg/s]
P_1 = 110 [kPa]
P_{2s} = 1422 [kPa]
P_4 = 110 [kPa]
P_{amb} = 105.3 [kPa]
P_{evap} = 1422 [kPa]
P_{SL} = 1422 [kPa]
$\dot{Q}_{HS,max}$ = 42.57 [kW]
$\dot{Q}_{out}$ = 15.98 [kW]
s_1 = 1.333 [KJ/kg-K]
s_{2s} = 1.333 [kJ/kg-K]
s_4 = 6.752 [kJ/kg-K]
s_{SL} = 2.29 [kJ/kg-K]
T_2 = 102.5 [C]
T_3 = 195.8 [C]
T_{4s} = 102.3 [C]
$T_{exh,in}$ = 300 [C]
$T_{HS,out}$ = 180.4 [C]
T_{SL} = 195.8 [C]
$\dot{W}_{net}$ = 2.29 [kW]
x_1 = 0
x_{4s} = 0.8557

No unit problems were detected.

Parametric Table: Varying Engine Conditions water

	$T_{exh,in}$ [C]	$\dot{m}_{air}$ [kg/s]	$\dot{W}_{net}$ [kW]	$\dot{Q}_{in}$ [kW]	η_{cycle}	$\dot{Q}_{HS,max}$ [kW]	η_{utiliz}
Run 1	260	0.15	1.345	10.45	0.1287	36.28	0.03707
Run 2	260	0.25	2.241	17.42	0.1287	60.47	0.03707
Run 3	260	0.35	3.138	24.39	0.1287	84.66	0.03707
Run 4	260	0.45	4.035	31.35	0.1287	108.8	0.03707
Run 5	280	0.15	1.836	14.13	0.13	39.43	0.04657
Run 6	280	0.25	3.06	23.54	0.13	65.72	0.04657
Run 7	280	0.35	4.285	32.96	0.13	92	0.04657
Run 8	280	0.45	5.509	42.38	0.13	118.3	0.04657
Run 9	300	0.15	2.34	17.79	0.1315	42.59	0.05494
Run 10	300	0.25	3.9	29.65	0.1315	70.98	0.05494
Run 11	300	0.35	5.46	41.52	0.1315	99.38	0.05494
Run 12	300	0.45	7.019	53.38	0.1315	127.8	0.05494
Run 13	320	0.15	2.858	21.45	0.1332	45.76	0.06244
Run 14	320	0.25	4.763	35.75	0.1332	76.27	0.06244
Run 15	320	0.35	6.668	50.05	0.1332	106.8	0.06244
Run 16	320	0.45	8.573	64.35	0.1332	137.3	0.06244

Parametric Table: Varying maximum cycle pressure_water

	P_{evap}	T_3	$\dot{m}_{steam}$	$\dot{W}_{net}$	$\dot{Q}_{in}$	η_{cycle}	$\dot{Q}_{HS,max}$	η_{utiliz}
	[kPa]	[C]	[kg/s]	[kW]	[kW]		[kW]	
Run 1	1000	179.9	0.008834	2.239	20.74	0.108	42.51	0.05267
Run 2	1021	180.8	0.008778	2.246	20.62	0.1089	42.51	0.05282
Run 3	1042	181.7	0.008724	2.252	20.49	0.1099	42.52	0.05296
Run 4	1063	182.6	0.008671	2.257	20.37	0.1108	42.52	0.05309
Run 5	1084	183.4	0.008618	2.263	20.26	0.1117	42.52	0.05321
Run 6	1106	184.3	0.008566	2.267	20.14	0.1126	42.53	0.05332
Run 7	1127	185.1	0.008514	2.272	20.02	0.1134	42.53	0.05341
Run 8	1148	186	0.008463	2.276	19.91	0.1143	42.53	0.0535
Run 9	1169	186.8	0.008413	2.279	19.8	0.1151	42.54	0.05358
Run 10	1190	187.6	0.008363	2.282	19.69	0.1159	42.54	0.05366
Run 11	1211	188.4	0.008314	2.285	19.58	0.1167	42.54	0.05372
Run 12	1232	189.2	0.008266	2.288	19.47	0.1175	42.54	0.05377
Run 13	1253	189.9	0.008218	2.29	19.36	0.1183	42.55	0.05382
Run 14	1274	190.7	0.008171	2.292	19.25	0.1191	42.55	0.05386
Run 15	1296	191.4	0.008124	2.294	19.14	0.1198	42.55	0.0539
Run 16	1317	192.2	0.008077	2.295	19.04	0.1205	42.56	0.05393
Run 17	1338	192.9	0.008031	2.296	18.94	0.1213	42.56	0.05395
Run 18	1359	193.7	0.007986	2.297	18.83	0.122	42.56	0.05396
Run 19	1380	194.4	0.007941	2.297	18.73	0.1227	42.56	0.05397
Run 20	1401	195.1	0.007896	2.298	18.63	0.1233	42.57	0.05397
Run 21	1422	195.8	0.007852	2.298	18.53	0.124	42.57	0.05397
Run 22	1443	196.5	0.007809	2.297	18.43	0.1247	42.57	0.05397
Run 23	1464	197.2	0.007765	2.297	18.33	0.1253	42.57	0.05395
Run 24	1486	197.8	0.007722	2.296	18.23	0.126	42.58	0.05394
Run 25	1507	198.5	0.00768	2.296	18.13	0.1266	42.58	0.05391
Run 26	1528	199.2	0.007638	2.295	18.04	0.1272	42.58	0.05389
Run 27	1549	199.8	0.007596	2.293	17.94	0.1278	42.59	0.05386
Run 28	1570	200.5	0.007555	2.292	17.85	0.1284	42.59	0.05382
Run 29	1591	201.1	0.007514	2.291	17.75	0.129	42.59	0.05378
Run 30	1612	201.7	0.007473	2.289	17.66	0.1296	42.59	0.05374
Run 31	1633	202.4	0.007433	2.287	17.57	0.1302	42.6	0.05369
Run 32	1654	203	0.007393	2.285	17.47	0.1308	42.6	0.05364
Run 33	1676	203.6	0.007353	2.283	17.38	0.1313	42.6	0.05359

Run 90	2879	231.6	0.005457	2.001	12.95	0.1546	42.72	0.04685
Run 91	2900	232	0.005429	1.995	12.88	0.1549	42.72	0.04669
Run 92	2921	232.4	0.0054	1.988	12.81	0.1552	42.73	0.04653
Run 93	2942	232.8	0.005371	1.981	12.74	0.1555	42.73	0.04637
Run 94	2963	233.2	0.005343	1.975	12.67	0.1558	42.73	0.04621
Run 95	2984	233.6	0.005315	1.968	12.61	0.1561	42.73	0.04605
Run 96	3006	234	0.005287	1.961	12.54	0.1564	42.73	0.04589
Run 97	3027	234.3	0.005258	1.954	12.47	0.1567	42.74	0.04573
Run 98	3048	234.7	0.005231	1.948	12.41	0.157	42.74	0.04557
Run 99	3069	235.1	0.005203	1.941	12.34	0.1572	42.74	0.04541
Run 100	3090	235.5	0.005175	1.934	12.28	0.1575	42.74	0.04524

Appendix B: Ethanol ORC modeling

```
//ORC-Cyclopentane

"Inputs"
P_evap = 3090[kPa]//  solution is initionlized at 15.1 bar, then the optimal value P_evap choosen out of the range (15.1 to 30.9 bar )
P_cond=   110 [  kPa] // condensing pressure is kept constant at 1.1 bar for all cases
T_exh_in= 300[C] // solution is initilized as this temp which repesents the dominant exhaust temp, parametric study for P_evap is done at this point
P_exh_in = 103 [kPa] //solution is initilized as this temp which repesents the  dominant ehasut pressure , P_evap parametric study is done at this point
T_amb =  25 [C]
P_amb = 105.25 [kPa]
Delta_T_pp = 15[C] // delta T between SL temp and pintch point HS initiated at 25 C
Delta_TC_T_SH= 10 [C] // temperature difference between the superheater outlet temp and crtical temp of the working fluid
Delta_T_pp_cond = 5 [C]  // delta T betwen SV point in condenser and coolant pinch point
m_dot_air=  0.15 [kg/s]       // solution is initilized as this mass flow rate which represents  the  dominant exhaust pressure , P_evap parametric study is done at this point
bulk_HS_T =( T_exh_in +T_HS_out_non )/2     //c_p is taken at mean tempearure of the heat source, a neglegible deviation in this value (by 0.008% when a recuprator is added)
Delta_T_min_r = 25 [C] // mimimum temprature difference in the recuporator

cp_air =Cp(Air,T=T_exh_in) //varying input  depends on Delta_HS_T
eta_pump= 0.7
eta_expander= 0.75
eta_mecha_elect= 0.9

"Working Fluid: Cyclopentane"
TC=t_crit(Cyclopentane)
PC=p_crit(Cyclopentane)

"state1"
 P_1 = P_cond
x_1 = 0
s_1=entropy(Cyclopentane,P=P_1,x=x_1)
h_1=enthalpy(Cyclopentane,P=P_1,x=x_1)
T_1= temperature(Cyclopentane,P=P_1,x=x_1)



"state _2s"
P_2s= P_evap
s_2s= s_1
h_2s= enthalpy(Cyclopentane,P=P_2s,s=s_2s)
T_2s = temperature(Cyclopentane,P=P_2s,s=s_2s)

"State 2"
P_2 = P_evap
 h_2 = ( ( h_2s-h_1)/eta_pump )+h_1
T_2 = temperature(Cyclopentane,P=P_2,h=h_2)
s_2=entropy(Cyclopentane,T=T_2,P=P_2)

"State SL"
 P_SL= P_evap
x_SL= 0
T_SL = temperature(Cyclopentane,x=x_SL,P=P_SL)
s_SL=entropy(Cyclopentane,P=P_SL,x=x_SL)
 h_SL = enthalpy(Cyclopentane, P=P_SL,x=x_SL)

//"State 3"
P_3= P_evap
T_3= temperature(Cyclopentane,P=P_3, x= 1 )
s_3=entropy(Cyclopentane,P=P_3, x=1)
h_3=enthalpy (Cyclopentane,P=P_3, x=1)

//if superheater added
"State 3"
//P_3 = P_evap
```

```
//T_3 =  TC- Delta_TC_T_SH // If T3> 260 then set T_3 = 260
//s_3 = entropy( Cyclopentane, P= P_3, T= T_3)
//h_3=  enthalpy (Cyclopentane,P=P_3, T = T_3)


"State 4s"
 P_4s= P_cond
s_4s= s_3
T_4s= temperature(Cyclopentane,P=P_4s,s=s_4s)
h_4s = enthalpy(Cyclopentane, P=P_4s, s=s_4s)

" State 4"
 h_4= - (  (eta_expander*( h_3-h_4s) ) - h_3 )
P_4= P_cond
T_4 = temperature(Cyclopentane,P=P_4,h=h_4)
s_4 = entropy(Cyclopentane,P=P_4,h= h_4)

// if Recuperator added, only for Oragnic working fluids
"State 4r"
P_4r = P_cond
T_4r = Delta_T_min_r + T_2
h_4r = enthalpy ( Cyclopentane, P=P_4r, T= T_4r)
s_4r= entropy (Cyclopentane, P=P_4r,h= h_4r)

//State 2r
h_4 - h_4r = h_2r- h_2
Q_dot_rec =( h_4 - h_4r) *m_dot_cy
P_2r= P_evap
T_2r = temperature ( Cyclopentane, P= P_2r, h= h_2r)
s_2r = entropy (Cyclopentane, P= P_2r, h= h_2r)

//state Cond SV
P_cond_SV = P_cond
T_cond_SV = temperature ( Cyclopentane, P= P_cond_SV, x = 1)
h_cond_SV = enthalpy   ( Cyclopentane, P= P_cond_SV, x = 1)
//For condenser design:
Q_SV_1 = m_dot_cy*(h_cond_SV-h_1)
Q_SV_1=Q_cooling
T_cool_out= T_cond_SV - Delta_T_pp_cond
T_bulk_cool=( T_amb+T_cool_out)/2
cp_air_fan = cp(Air,T = T_bulk_cool)
Q_cooling = m_dot_cool *cp_air_fan *(T_cool_out- T_amb)

// HS_in
T_HS_in = T_exh_in
 h_HS_in = cp_air*T_HS_in

// HS_pp
 T_HS_pp=Delta_T_pp+T_SL
h_HS_pp= cp_air*T_HS_pp

// Enegy Balance Evaporator & Heat Source
Q_3_SL= Q_HS_in_HS_pp
Q_HS_in_HS_pp= m_dot_air*(h_HS_in-h_HS_pp)
Q_3_SL= m_dot_cy * (h_3-h_SL)

 // heat added to cycle without Recuprator
Q_dot_in_non=m_dot_cy *(h_3-h_2)

// heat added to cycle without Recuprator
Q_dot_in_rec=m_dot_cy *(h_3-h_2r)

// heat rejected through condenser without Recuprator
Q_dot_out_non= m_dot_cy*(h_4-h_1)

// heat rejected through condenser with Recuprator
Q_dot_out_rec= m_dot_cy*(h_4r-h_1)
```

```
//Work requred by the pump
W_dot_in= m_dot_cy*(h_2-h_1)


// Useful work produced by the expander
W_dot_out=eta_mecha_elect *(h_3-h_4)*m_dot_cy
// back work ratio
bwr = W_dot_in/W_dot_out

// Net Work output of the cycle
W_dot_net= W_dot_out - W_dot_in

// Thermal Efficiency of the cycle without Recuprator
eta_cycle_non = W_dot_net / Q_dot_in_non

// Thermal Efficiency of the cycle with Recuprator
eta_cycle_rec = W_dot_net / Q_dot_in_rec

// HS_out_non
Q_dot_in_non = m_dot_air *(h_HS_in-h_HS_out_non)
h_HS_out_non= cp_air *T_HS_out_non

// HS_out_rec
Q_dot_in_rec = m_dot_air *(h_HS_in-h_HS_out_rec)
 h_HS_out_rec= cp_air *T_HS_out_rec

// Maximum availble thermal energy from the heat source (heat source energy content)

 Q_dot_HS_max = m_dot_air *cp_air*(T_HS_in-T_amb)
eta_uti = W_dot_net/ Q_dot_HS_max

T_diff_PH_HS_out = T_HS_out_non - T_2


// Mass Correlations:
rho_pp = 7500 [kg/m^3]
v_pp = 2.7071e-4 [m^3]
M_pp = rho_pp*v_pp
N_p_total_non = 248
N_p_total_rec = 296
M_hx_non = N_p_total_non * M_pp
M_hx_rec =  N_p_total_rec * M_pp
M_p =  (1.0764* W_dot_in ) + 1.8022
M_exp= ( 0.3448 *W_dot_out) + 6.4655

M_orc_non = M_hx_non+ M_p+M_exp // Neglect of the mass of the working fluid and piping
M_orc_rec = M_hx_rec+ M_p+M_exp   // Neglect of the mass of the working fluid and piping
W_dot_engine =  373[kW]
M_vehicle = 36278[kg]  //when fully loaded
 // The increased engine load casued by teh ORC installation
W_dot_orc_w_non =  ( 0.04 * W_dot_engine *M_orc_non ) /(0.1*M_vehicle)
W_dot_orc_w_rec = ( 0.04 * W_dot_engine *M_orc_rec ) /(0.1*M_vehicle)

// Now accounting the negative effect of installing the ORC system
W_cycle_non_NE =  W_dot_net - W_dot_orc_w_non
W_cycle_rec_NE =  W_dot_net - W_dot_orc_w_rec

// T-S plot points
//T[1]=  T_1
//s[1]=s_1
//T[2]= T_2s
//s[2]=s_2s
//T[3]= T_2
//s[3]= s_2
//T[4]= T_3
//s[4]= s_3
//T[5] = T_4s
//s[5]= s_4s
```

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$bulk_{HS,T}$ = 179.9 [C]	bwr = 0.06357
cp_{air} = 1.045 [kJ/kg-C]	$cp_{air,fan}$ = 1.005 [kJ/kg-C]
$\delta_{TC,T,SH}$ = 10 [C]	$\delta_{T,min,r}$ = 25 [C]
$\delta_{T,pp}$ = 15 [C]	$\delta_{T,pp,cond}$ = 5 [C]
$\eta_{cycle,non}$ = 0.1547	$\eta_{cycle,rec}$ = 0.1677
$\eta_{expander}$ = 0.75	$\eta_{mecha,elect}$ = 0.9
η_{pump} = 0.7	η_{uti} = 0.1351
h_1 = 5.287 [kJ/kg]	h_2 = 11.25 [kJ/kg]
h_{2r} = 55.38 [kJ/kg]	h_{2s} = 9.459 [kJ/kg]
h_3 = 579 [kJ/kg]	h_4 = 474.8 [kJ/kg]
h_{4r} = 430.7 [kJ/kg]	h_{4s} = 440.1 [kJ/kg]
$h_{cond,SV}$ = 392.3 [kJ/kg]	$h_{HS,in}$ = 313.4 [kJ/kg]
$h_{HS,out,non}$ = 62.45 [kJ/kg]	$h_{HS,out,rec}$ = 81.96 [kJ/kg]
$h_{HS,pp}$ = 235.7 [kJ/kg]	h_{SL} = 403.2 [kJ/kg]
$\dot{m}_{air}$ = 0.15 [kg/s]	$\dot{m}_{cool}$ = 1.172 [kg/s]
$\dot{m}_{cy}$ = 0.06631 [kg/s]	M_{exp} = 8.609
$M_{hx,non}$ = 503.5	$M_{hx,rec}$ = 601
$M_{orc,non}$ = 514.4	$M_{orc,rec}$ = 611.8 [kg]
M_p = 2.228	M_{pp} = 2.03
$M_{vehicle}$ = 36278 [kg]	$N_{p,total,non}$ = 248
$N_{p,total,rec}$ = 296	PC = 4571 [kPa]
P_1 = 110 [kPa]	P_2 = 3090 [kPa]
P_{2r} = 3090 [kPa]	P_{2s} = 3090 [kPa]
P_3 = 3090 [kPa]	P_4 = 110 [kPa]
P_{4r} = 110 [kPa]	P_{4s} = 110 [kPa]
P_{amb} = 105.3 [kPa]	P_{cond} = 110 [·kPa]
$P_{cond,SV}$ = 110 [kPa]	P_{evap} = 3090 [kPa]
$P_{exh,in}$ = 103 [kPa]	P_{SL} = 3090 [kPa]
$Q_{3,SL}$ = 11.66 [kW]	$Q_{cooling}$ = 25.66 [kW]
$\dot{Q}_{HS,max}$ = 43.1 [kW]	$\dot{Q}_{in,non}$ = 37.65 [kW]
$\dot{Q}_{in,rec}$ = 34.72 [kW]	$\dot{Q}_{out,non}$ = 31.13 [kW]
$\dot{Q}_{out,rec}$ = 28.21 [kW]	$\dot{Q}_{rec}$ = 2.926 [kW]
$Q_{HS,in,HS,pp}$ = 11.66 [kW]	$Q_{SV,1}$ = 25.66 [kW]
ρ_{pp} = 7500 [kg/m^3]	s_1 = 0.01629 [kJ/kg-K]
s_2 = 0.02177 [kJ/kg-K]	s_{2r} = 0.1523 [kJ/kg-K]
s_{2s} = 0.01629 [kJ/kg-K]	s_3 = 1.347 [kJ/kg-K]
s_4 = 1.441 [kJ/kg-K]	s_{4r} = 1.321 [kJ/kg-K]
s_{4s} = 1.347 [kJ/kg-K]	s_{SL} = 0.9837 [kJ/kg-K]
TC = 238.6 [C]	T_1 = 51.79 [C]
T_2 = 53.9 [C]	T_{2r} = 75.84 [C]
T_{2s} = 52.98 [C]	T_3 = 210.6 [C]
T_4 = 107.4 [C]	T_{4r} = 78.9 [C]
T_{4s} = 85.18 [C]	T_{amb} = 25 [C]
$T_{bulk,cool}$ = 35.89 [C]	$T_{cond,SV}$ = 51.79 [C]
$T_{cool,out}$ = 46.79 [C]	$T_{diff,PH,HS,out}$ = 5.879 [C]

$T_{exh,in}$ = 300 [C]	$T_{HS,in}$ = 300 [C]
$T_{HS,out,non}$ = 59.78 [C]	$T_{HS,out,rec}$ = 78.45 [C]
$T_{HS,pp}$ = 225.6 [C]	T_{SL} = 210.6 [C]
V_{pp} = 0.0002707 [m^3]	$W_{cycle,non,NE}$ = 3.707 [kW]
$W_{cycle,rec,NE}$ = 3.306	$\dot{W}_{engine}$ = 373 [kW]
$\dot{W}_{in}$ = 0.3952 [kW]	$\dot{W}_{net}$ = 5.822 [kW]
$\dot{W}_{orc,w,non}$ = 2.115 [kW]	$\dot{W}_{orc,w,rec}$ = 2.516 [kW]
$\dot{W}_{out}$ = 6.218 [kW]	x_1 = 0
x_{SL} = 0	

Appendix C: Ethanol ORC modeling

```
//ORC-Ethanol


"Inputs"
P_evap = 3090 [kPa]//  solution is initionlized at 15.1 bar, then the optimal value P_evap choosen out of the range (15.1 to 30.9 bar )
P_cond=   110 [  kPa] // condensing pressure is kept constant at 1.1 bar for all cases
T_exh_in= 300[C] // solution is initilized as this temp which repesents the dominant exhaust temp, parametric study for P_evap is done at this point
P_exh_in = 103 [kPa] //solution is initilized as this temp which repesents the  dominant ehasut pressure , P_evap parametric study is done at this point
T_amb =  25 [C]
P_amb = 105.25 [kPa]
Delta_T_pp = 5[C] // delta T between SL temp and pintch point HS
Delta_TC_T_SH= 10 [C] // temperature difference between the superheater outlet temp and crtical temp of the working fluid
Delta_T_pp_cond = 5 [C]  // delta T betwen SV point in condenser and coolant pinch point
m_dot_air=  0.15 [kg/s]      // solution is initilized as this mass flow rate which represents  the  dominant exhaust pressure , P_evap parametric study is done at this point
bulk_HS_T =( T_exh_in +T_HS_out_non )/2    //c_p is taken at mean tempearure of the heat source, a neglegible deviation in this value (by 0.008% when a recuprator is added)
Delta_T_min_r = 25 [C] // mimimum temprature difference in the recuporator

cp_air =Cp(Air,T=T_exh_in) //varying input  depends on Delta_HS_T
eta_pump= 0.7
eta_expander= 0.75
eta_mecha_elect= 0.9

"Working Fluid: Ethanol"
TC=t_crit(Ethanol)
PC=p_crit(Ethanol)

"state1"
 P_1 = P_cond
x_1 = 0
s_1=entropy(Ethanol,P=P_1,x=x_1)
h_1=enthalpy(Ethanol,P=P_1,x=x_1)
T_1= temperature(Ethanol,P=P_1,x=x_1)



"state _2s"
P_2s= P_evap
s_2s= s_1
h_2s= enthalpy(Ethanol,P=P_2s,s=s_2s)
T_2s = temperature(Ethanol,P=P_2s,s=s_2s)

"State 2"
P_2 = P_evap
 h_2 = ( ( h_2s-h_1)*eta_pump )+h_1
T_2 = temperature(Ethanol,P=P_2,h=h_2)
s_2=entropy(Ethanol,T=T_2,P=P_2)

"State SL"
 P_SL= P_evap
x_SL= 0
T_SL = temperature(Ethanol,x=x_SL,P=P_SL)
s_SL=entropy(Ethanol,P=P_SL,x=x_SL)
 h_SL = enthalpy(Ethanol, P=P_SL,x=x_SL)

//"State 3"
P_3= P_evap
T_3= temperature(Ethanol,P=P_3, x= 1 )
s_3=entropy(Ethanol,P=P_3, x=1)
h_3=enthalpy (Ethanol,P=P_3, x=1)

//if superheater added
"State 3"
```

```
//P_3 = P_evap
//T_3 =  TC- Delta_TC_T_SH // If T3> 260 then set T_3 = 260
//s_3 = entropy( Ethanol, P= P_3, T= T_3)
//h_3=  enthalpy (Ethanol,P=P_3, T = T_3)


"State 4s"
 P_4s= P_cond
s_4s= s_3
T_4s= temperature(Ethanol,P=P_4s,s=s_4s)
h_4s = enthalpy(Ethanol, P=P_4s, s=s_4s)

" State 4"
 h_4= - (  (eta_expander*( h_3-h_4s) ) - h_3 )
P_4= P_cond
T_4 = temperature(Ethanol,P=P_4,h=h_4)
s_4 = entropy(Ethanol,P=P_4,h= h_4)

// if Recuperator added, only for Oragnic working fluids
"State 4r"
P_4r = P_cond
T_4r = Delta_T_min_r + T_2
h_4r = enthalpy (Ethanol , P=P_4r, T= T_4r)
s_4r= entropy (Ethanol, P=P_4r,h= h_4r)

//State 2r
h_4 - h_4r = h_2r- h_2
P_2r= P_evap
T_2r = temperature (Ethanol , P= P_2r, h= h_2r)
s_2r = entropy (Ethanol, P= P_2r, h= h_2r)



// HS_in
T_HS_in = T_exh_in
 h_HS_in = cp_air*T_HS_in

// HS_pp
 T_HS_pp=Delta_T_pp+T_SL
h_HS_pp= cp_air*T_HS_pp

// Enegy Balance Evaporator & Heat Source
Q_3_SL= Q_HS_in_HS_pp
Q_HS_in_HS_pp= m_dot_air*(h_HS_in-h_HS_pp)
Q_3_SL= m_dot_eth * (h_3-h_SL)
 // heat added to cycle without Recuprator
Q_dot_in_non=m_dot_eth *(h_3-h_2)

// heat added to cycle without Recuprator
Q_dot_in_rec=m_dot_eth *(h_3-h_2r)

// heat rejected through condenser without Recuprator
Q_dot_out_non= m_dot_eth*(h_4-h_1)

// heat rejected through condenser with Recuprator
Q_dot_out_rec= m_dot_eth*(h_4r-h_1)

//Work requred by the pump
W_dot_in= m_dot_eth*(h_2-h_1)


// Useful work produced by the expander
W_dot_out=eta_mecha_elect *(h_3-h_4)*m_dot_eth
// back work ratio
bwr = W_dot_in/W_dot_out

// Net Work output of the cycle
W_dot_net= W_dot_out - W_dot_in
```

```
// Thermal Efficiency of the cycle without Recuprator
eta_cycle_non = W_dot_net / Q_dot_in_non

// Thermal Efficiency of the cycle with Recuprator
eta_cycle_rec = W_dot_net / Q_dot_in_rec

// HS_out_non
Q_dot_in_non = m_dot_air *(h_HS_in-h_HS_out_non)
h_HS_out_non= cp_air *T_HS_out_non

// HS_out_rec
Q_dot_in_rec = m_dot_air *(h_HS_in-h_HS_out_rec)
 h_HS_out_rec= cp_air *T_HS_out_rec

// Maximum availble thermal energy from the heat source (heat source energy content)

 Q_dot_HS_max = m_dot_air *cp_air*(T_HS_in-T_amb)
eta_uti = W_dot_net/ Q_dot_HS_max

// T-S plot points
//T[1]=  T_1
//s[1]=s_1
//T[2]= T_2s
//s[2]=s_2s
//T[3]= T_2
//s[3]= s_2
//T[4]= T_3
//s[4]= s_3
//T[5] = T_4s
//s[5]= s_4s
//T[6] = T_4
//s[6] = s_4
//T[7] = T_SL
//s[7] = s_SL
//T[8] = T_2r
//s[8] = s_2r
//T[9] = T_4r
//s[9] = s_4r
```

Appendix D: R245fa ORC modeling

```
//ORC-R245fa


"Inputs"
P_evap = 3090[kPa]//  solution is initionlized at 15.1 bar, then the optimal value P_evap choosen out of the range (15.1 to 30.9 bar )
P_cond=   110 [  kPa] // condensing pressure is kept constant at 1.1 bar for all cases
T_exh_in= 300[C] // solution is initilized as this temp which repesents the dominant exhaust temp, parametric study for P_evap is done at this point
P_exh_in = 103 [kPa] //solution is initilized as this temp which repesents the  dominant ehasut pressure , P_evap parametric study is done at this point
T_amb =  25 [C]
P_amb = 105.25 [kPa]
//Delta_T_pp = 5[C] // delta T between SL temp and pintch point HS
Delta_TC_T_SH= 10 [C] // temperature difference between the superheater outlet temp and crtical temp of the working fluid
Delta_T_pp_cond = 5 [C]  // delta T betwen SV point in condenser and coolant pinch point
m_dot_air=  0.15 [kg/s]      // solution is initilized as this mass flow rate which represents  the  dominant exhaust pressure , P_evap parametric study is done at this point
bulk_HS_T =( T_exh_in +T_HS_out_non )/2    //c_p is taken at mean tempearure of the heat source, a neglegible deviation in this value (by 0.008% when a recuprator is added)
Delta_T_min_r = 25 [C] // mimimum temprature difference in the recuporator
 Delta_T_min_PH =  5[C]
cp_air =Cp(Air,T=T_exh_in) //varying input  depends on Delta_HS_T
eta_pump= 0.7
eta_expander= 0.75
eta_mecha_elect= 0.9

"Working Fluid: R245fa"
TC=t_crit(R245fa)
PC=p_crit(R245fa)

"state1"
 P_1 = P_cond
x_1 = 0
s_1=entropy(R245fa,P=P_1,x=x_1)
h_1=enthalpy(R245fa,P=P_1,x=x_1)
T_1= temperature(R245fa,P=P_1,x=x_1)



"state _2s"
P_2s= P_evap
s_2s= s_1
h_2s= enthalpy(R245fa,P=P_2s,s=s_2s)
T_2s = temperature(R245fa,P=P_2s,s=s_2s)

"State 2"
P_2 = P_evap
 h_2 = ( ( h_2s-h_1)*eta_pump )+h_1
T_2 = temperature(R245fa,P=P_2,h=h_2)
s_2=entropy(R245fa,T=T_2,P=P_2)

"State SL"
 P_SL= P_evap
x_SL= 0
T_SL = temperature(R245fa,x=x_SL,P=P_SL)
s_SL=entropy(R245fa,P=P_SL,x=x_SL)
 h_SL = enthalpy(R245fa, P=P_SL,x=x_SL)

//"State 3"
P_3= P_evap
T_3= temperature(R245fa,P=P_3, x= 1 )
s_3=entropy(R245fa,P=P_3, x=1)
h_3=enthalpy (R245fa,P=P_3, x=1)

//if superheater added
"State 3"
```

```
//P_3 = P_evap
//T_3 =  TC- Delta_TC_T_SH // If T3> 260 then set T_3 = 260
//s_3 = entropy(R245fa, P= P_3, T= T_3)
//h_3=  enthalpy (R245fa,P=P_3, T = T_3)


"State 4s"
 P_4s= P_cond
s_4s= s_3
T_4s= temperature(R245fa,P=P_4s,s=s_4s)
h_4s = enthalpy(R245fa, P=P_4s, s=s_4s)

" State 4"
 h_4= - (  (eta_expander*( h_3-h_4s) ) - h_3 )
P_4= P_cond
T_4 = temperature(R245fa,P=P_4,h=h_4)
s_4 = entropy(R245fa,P=P_4,h= h_4)

// if Recuperator added, only for Oragnic working fluids
"State 4r"
P_4r = P_cond
T_4r = Delta_T_min_r + T_2
h_4r = enthalpy (R245fa , P=P_4r, T= T_4r)
s_4r= entropy (R245fa, P=P_4r,h= h_4r)

//State 2r
h_4 - h_4r = h_2r- h_2
P_2r= P_evap
T_2r = temperature (R245fa , P= P_2r, h= h_2r)
s_2r = entropy (R245fa, P= P_2r, h= h_2r)


// HS_in
T_HS_in = T_exh_in
 h_HS_in = cp_air*T_HS_in


 // HS_out_non
T_HS_out_non = Delta_T_min_PH+ T_2
h_HS_out_non = T_HS_out_non *cp_air
//HS_out_rec
T_HS_out_rec = Delta_T_min_PH+ T_2r
h_HS_out_rec= T_HS_out_rec *cp_air

// Enegy Balance Evaporator & Heat Source to find mass flow rate without Recuprator +Heat added
Q_dot_in_non= Q_HS_in_HS_out_non
Q_HS_in_HS_out_non= m_dot_air*(h_HS_in-h_HS_out_non)
Q_dot_in_non= m_dot_R_non * (h_3-h_2)

 // Enegy Balance Evaporator & Heat Source to find mass flow rate with Recuprator +Heat added
Q_dot_in_rec= Q_HS_in_HS_out_rec
Q_HS_in_HS_out_rec= m_dot_air*(h_HS_in-h_HS_out_rec)
Q_dot_in_non= m_dot_R_rec * (h_3-h_2)


 // HS_pp_non
Q_in_pp_non = Q_3_SL_non
Q_3_SL_non = m_dot_R_non *(h_3-h_SL)
Q_in_pp_non = (h_HS_in - h_PP_non)*m_dot_R_non
h_PP_non = cp_air *T_PP_non

 // HS_pp_rec
Q_in_pp_rec = Q_3_SL_rec
Q_3_SL_rec = m_dot_R_rec *(h_3-h_SL)
Q_in_pp_rec = (h_HS_in - h_PP_rec)*m_dot_R_rec
h_PP_rec = cp_air *T_PP_rec
```

```
//state Cond SV
P_cond_SV = P_cond
T_cond_SV = temperature ( R245fa, P= P_cond_SV, x = 1)
h_cond_SV = enthalpy   ( R245fa, P= P_cond_SV, x = 1)
//For condenser design:
Q_SV_1 = m_dot_R_non*(h_cond_SV-h_1)
Q_SV_1=Q_cooling
T_cool_in= T_cond_SV - 8
T_cool_out = T_cond_SV-5
T_bulk_cool=( T_cool_out+T_cool_in)/2
cp_air_fan = cp(Air,T = T_bulk_cool)
Q_cooling = m_dot_cool *cp_air_fan *( T_cool_out-T_cool_in)


// heat rejected through condenser without Recuprator
Q_dot_out_non= m_dot_R_non*(h_4-h_1)

// heat rejected through condenser with Recuprator
Q_dot_out_rec= m_dot_R_rec*(h_4r-h_1)

//Work requred by the pump without rec
W_dot_in_non= m_dot_R_non*(h_2-h_1)

//Work requred by the pump with  rec
W_dot_in_rec= m_dot_R_rec*(h_2-h_1)


// Useful work produced by the expander without rec
W_dot_out_non=eta_mecha_elect *(h_3-h_4)*m_dot_R_non

// Useful work produced by the expander with rec
W_dot_out_rec=eta_mecha_elect *(h_3-h_4)*m_dot_R_rec

// back work ratio
bwr = W_dot_in_non/W_dot_out_non

// Net Work output of the cycle without rec
W_dot_net_non= W_dot_out_non - W_dot_in_non

// Net Work output of the cycle with rec
W_dot_net_rec= W_dot_out_rec - W_dot_in_rec

// Thermal Efficiency of the cycle without Recuprator
eta_cycle_non = W_dot_net_non / Q_dot_in_non

// Thermal Efficiency of the cycle with Recuprator
eta_cycle_rec = W_dot_net_rec / Q_dot_in_rec



// Maximum availble thermal energy from the heat source (heat source energy content)

 Q_dot_HS_max = m_dot_air *cp_air*(T_HS_in-T_amb)
eta_uti_non = W_dot_net_non/ Q_dot_HS_max
eta_uti_rec = W_dot_net_rec/ Q_dot_HS_max

// Mass Correlations:
rho_pp = 7500 [kg/m^3]
v_pp = 2.7071e-4 [m^3]
M_pp = rho_pp*v_pp
N_p_total_non = 100
N_p_total_rec = 269
M_hx_non = N_p_total_non * M_pp
M_hx_rec =  N_p_total_rec * M_pp
M_p =  (1.0764* W_dot_in_non ) + 1.8022
M_exp= ( 0.3448 *W_dot_out_non) + 6.4655

M_orc_non = M_hx_non+ M_p+M_exp // Neglect of the mass of the working fluid and piping
```

```
M_orc_rec = M_hx_rec+ M_p+M_exp   // Neglect of the mass of the working fluid and piping
W_dot_engine =  373[kW]
M_vehicle = 36278[kg]  //when fully loaded
 // The increased engine load casued by teh ORC installation
W_dot_orc_w_non =  ( 0.04 * W_dot_engine *M_orc_non ) /(0.1*M_vehicle)
W_dot_orc_w_rec = ( 0.04 * W_dot_engine *M_orc_rec ) /(0.1*M_vehicle)

// Now accounting the negative effect of installing the ORC system
W_cycle_non_NE =  W_dot_net_non - W_dot_orc_w_non
W_cycle_rec_NE =  W_dot_net_non - W_dot_orc_w_rec


// T-S plot points
//T[1]=  T_1
//s[1]=s_1
//T[2]= T_2s
//s[2]=s_2s
//T[3]= T_2
//s[3]= s_2
//T[4]= T_3
//s[4]= s_3
//T[5] = T_4s
//s[5]= s_4s
//T[6] = T_4
//s[6] = s_4
//T[7] = T_SL
//s[7] = s_SL
//T[8] = T_2r
//s[8] = s_2r
//T[9] = T_4r
//s[9] = s_4r
```

Appendix E: Heat Exchangers Design for Cyclopentane

```
// fixed HX Geometry
W_e = 0.253[m] // width of the plate
L_p = 0.456 [m]
t_p = 0.002 [m] // Plate thickness
beta_chovron = 0.785  // Chevron angle
b = 0.002[m]
pco = 0.004[m] //. corrugation pitch
X_x= (b*pi#)/ pco // Wavenumber
 phi = (1/6) * (   1+  (1+(X_x^2)    )^(0.5)+ 4*(1+( (X_x^2)/2) )^(0.5) ) // Arae Increase Factor
D_h = (2*b)/ phi
d_eq = 2*b // eqivalint diameter
k_wall = 15 [W/m-K]  //conductivity of stainless steel

// Preheater , zone a( single Phase)
// Hot Flow (Heat Source)
P_h_a = 103[kPa]
T_h_in_a= 225.6[C]

T_h_out_a= 59.78 [C] //without Recuprator
//T_h_out_a= 78.45[C]  // withrecuprator

T_f_h_a = (T_h_in_a+T_h_out_a)/2
cp_h_a = cp(Air,T=T_f_h_a)
k_h_a = conductivity(Air,T=T_f_h_a)
pr_h_a = prandtl(Air,T=T_f_h_a)
mu_h_a = viscosity(Air,T=T_f_h_a)
rho_h_a = density(Air,T=T_f_h_a,P=P_h_a)
m_dot_h_a = 0.15 [kg/s]
G_h_a = m_dot_h_a/(b*W_e) //Mass flux
Re_h_a = (G_h_a*d_eq)/ ( mu_h_a)  // Re_h_a = 58799

//Convective heat transfer coeff (hot flow)
f_h_a=   (   (1.82*ln(Re_h_a) )    -   1.64)^(-2)
Nusselt_h_a  =      (     (f_h_a/ 8)* (Re_h_a-1)   *pr_h_a    )          / (    ( 12.7*(f_h_a/8)^(0.5)   *(pr_h_a^(2/3) -1)   )    +1.07
    )
 h_h_a= (Nusselt_h_a*k_h_a)/(d_eq)

// cold Flow (ORC working flow)
P_c_a = 3090[kPa]
T_c_out_a = 210.6 [C]


T_c_in_a= 53.9[C]  //without recuprator
//T_c_in_a= 75.84[C]        //with recuprator

T_f_c_a = (T_c_in_a+T_c_out_a)/2
cp_c_a = cp(Cyclopentane,T=T_f_c_a, P=P_c_a)
k_c_a = conductivity(Cyclopentane,T=T_f_c_a,P=P_c_a)
pr_c_a = prandtl(Cyclopentane,T=T_f_c_a,P=P_c_a)
mu_c_a = viscosity(Cyclopentane,T=T_f_c_a,P=P_c_a)
rho_c_a = density(Cyclopentane,T=T_f_c_a,P=P_c_a)
m_dot_c_a = 0.06631 [kg/s]
G_c_a = m_dot_c_a/(b*W_e)
Re_c_a = (G_c_a*d_eq)/ ( mu_c_a)
// Find wall viscousity mu_w using the sutherland equation
T_f_c_a_k =405.4[K]
mu_w = mu_ref *( T_f_c_a_k/T_ref)^(3/2)*(T_ref+S)/( T_f_c_a_k+S)
mu_ref = 1.74e-5[Pa-s] //Reference viscosity for stainless steel
T_ref = 273 [K]    // Refernce temp for stainless steel
S= 80 [K]   // Reference Sutherland constant for stainless steel

// Convection Heat trasnfer coefficient (cold side)
Nusselt_c_a =  (  (0.0154*beta_chovron)+ 0.1298) * Re_c_a^( (0.1892*beta_chovron)+0.6398     ) * pr_c_a^(0.35)*(
mu_c_a/mu_w)^(0.14)
h_c_a = (Nusselt_c_a*k_c_a)/(d_eq)
```

```
// heat tranfer area of zone a (preheater)

Q_a =  25990 [W]  //without recuprator
//Q_a =  23061 [W]  //with recuprator

delta1_a = T_h_in_a- T_c_out_a
delta2_a = T_h_out_a-T_c_in_a
LMTD_a = (delta1_a-delta2_a)/ln(delta1_a/delta2_a)
(1/U_a) = (1/h_h_a)+(1/h_c_a)+ (t_p/k_wall)
A_a = Q_a/(LMTD_a*U_a)
// Number of palte required for zone a
A_a = (N_p_a-2)*L_p*W_e
A_a_rec = 29.76 [m^2]
// ........................................................................................................................

//Design of evaporator, zone b
//Hot fluid (Heat source)
P_h_b = 103[kPa]
T_h_in_b= 300[C]
T_h_out_b= 225.6 [C]
T_f_h_b = (T_h_in_b+T_h_out_b)/2
cp_h_b = cp(Air,T=T_f_h_b)
k_h_b = conductivity(Air,T=T_f_h_b)
pr_h_b = prandtl(Air,T=T_f_h_b)

mu_h_b = viscosity(Air,T=T_f_h_b)
rho_h_b = density(Air,T=T_f_h_b,P=P_h_b)
m_dot_h_b = 0.15 [kg/s]
G_h_b = m_dot_h_a/(b*W_e)
Re_h_b = (G_h_b *d_eq)/ ( mu_h_b)  // Re_h_b = 49378
//Convective heat transfer coeff (hot flow)
f_h_b=    (   (1.82*ln(Re_h_b) )    -   1.64)^(-2)
Nusselt_h_b  =      (     (f_h_b/ 8)* (Re_h_b-1)   *pr_h_b    )          / (    ( 12.7*(f_h_b/8)^(0.5)   *(pr_h_b^(2/3) -1)   )    +1.07
    )
 h_h_b= (Nusselt_h_b*k_h_b)/(d_eq)

// Cold side ( Phase change)
P_c_b = 3090 [kPa]
 T_c_b=210.6[C]
 cp_c_b =cp(Cyclopentane,T=T_c_b,P=P_c_b)
k_c_b = conductivity(Cyclopentane,T=T_c_b,P=P_c_b)
pr_c_b = prandtl(Cyclopentane,T=T_c_b,P=P_c_b)
mu_c_b = viscosity(Cyclopentane,T=T_c_b,P=P_c_a)
rho_c_b_L = density(Cyclopentane,P=P_c_b, x=0)
rho_c_b_V = density(Cyclopentane,P=P_c_b, x=1)
m_dot_c_b = 0.06631 [kg/s]
G_c_b= m_dot_c_b/(b*W_e)
x_c_b  =0.75//vapor quality
G_c_b_eq = G_c_b*(   (1-x_c_b)+(    x_c_b *(rho_c_b_L/rho_c_b_V)^(0.5)     ) )
Re_c_b_eq = (G_c_b_eq*D_h)/(mu_c_b)
// Evaporating Heat Transfer Coeff:
h_c_b = 5.323*(k_c_b/D_h)*Re_c_b_eq^(0.42)*pr_c_b^(1/3)

// heat tranfer area of zone b (Evaporator)
Q_b =  11660 [W]
delta1_b = T_h_in_b- T_c_b
delta2_b = T_h_out_b - T_c_b
LMTD_b = (delta1_b-delta2_b)/ln(delta1_b/delta2_b)
(1/U_b) = (1/h_h_b)+(1/h_c_b)+ (t_p/k_wall)
A_b = Q_b/(LMTD_b*U_b)
// Number of palte required for zone a
A_b = (N_p_b-2)*L_p*W_e

//Design of condenser,
//Hot fluid (Heat source)
P_c_cond= 103.5[kPa]
T_c_in_cond= 25[C]
T_c_out_cond= 46.79 [C]
```

```
T_f_c_cond = (T_c_in_cond+T_c_out_cond)/2
cp_c_cond = cp(Air,T=T_f_c_cond)
k_c_cond = conductivity(Air,T=T_f_c_cond)
pr_c_cond = prandtl(Air,T=T_f_c_cond)
mu_c_cond = viscosity(Air,T=T_f_c_cond)
rho_c_cond = density(Air,T=T_f_c_cond,P=P_c_cond)
m_dot_c_cond = 1.172 [kg/s]
G_c_cond = m_dot_c_cond/(b*W_e)
Re_c_cond = (G_c_cond *d_eq)/ ( mu_c_cond)  // Re_h_b = 49378
//Convective heat transfer coeff (hot flow)
f_c_cond=   (   (1.82*ln(Re_c_cond) )    -    1.64)^(-2)
Nusselt_c_cond  =       (     (f_c_cond/ 8)* (Re_c_cond-1)   *pr_c_cond     )          / (    ( 12.7*(f_c_cond/8)^(0.5)   *(
pr_c_cond^(2/3) -1)   )    +1.07       )
h_c_cond= (Nusselt_c_cond*k_c_cond)/(d_eq)

// hot side ( Phase change)-condensing
P_h_cond = 110 [kPa]
 T_h_cond= 51.79[C]
 cp_h_cond =cp(Cyclopentane,T=T_h_cond,P=P_h_cond)
k_h_cond = conductivity(Cyclopentane,T=T_h_cond,P=P_h_cond)
pr_h_cond = prandtl(Cyclopentane,T=T_h_cond,P=P_h_cond)
mu_h_cond = viscosity(Cyclopentane,T=T_h_cond,P=P_h_cond)
rho_h_cond_L = density(Cyclopentane,P=P_h_cond, x=0)
rho_h_cond_V = density(Cyclopentane,P=P_h_cond, x=1)
m_dot_h_cond = 0.06631 [kg/s]
G_h_cond= m_dot_h_cond/(b*W_e)
x_h_cond  =0.75//vapor quality
G_h_cond_eq = G_h_cond*(   (1-x_h_cond)+(    x_h_cond *(rho_h_cond_L/rho_h_cond_V)^(0.5)     ) )
Re_h_cond_eq = (G_c_b_eq*D_h)/(mu_h_cond)
// Condensing Heat Transfer Coeff:
h_h_cond = 4.118*(k_h_cond/D_h)*Re_h_cond_eq^(0.4)*pr_h_cond^(1/3)

// heat tranfer area of zone b (Condenser)
Q_cond =  25660 [W]
delta1_cond = T_h_cond - T_c_out_cond
delta2_cond = T_h_cond- T_c_in_cond
LMTD_cond = (delta1_b-delta2_b)/ln(delta1_b/delta2_b)
(1/U_cond) = (1/h_h_cond)+(1/h_c_cond)+ (t_p/k_wall)
A_cond = Q_cond/(LMTD_cond*U_cond)
// Number of palte required for zone a
A_cond = (N_p_cond-2)*L_p*W_e

// Recuprator
//cold side - recuprator
P_c_rec = 3090[kPa]
T_c_in_rec= 53.9[C]
T_c_out_rec= 75.84 [C]
T_f_c_rec = (T_c_in_rec+T_c_out_rec)/2
cp_c_rec = cp(Cyclopentane,T=T_f_c_rec, P=P_c_rec)
k_c_rec = conductivity(Cyclopentane,T=T_f_c_rec,P=P_c_rec)
pr_c_rec = prandtl(Cyclopentane,T=T_f_c_rec,P=P_c_rec)
mu_c_rec = viscosity(Cyclopentane,T=T_f_c_rec,P=P_c_rec )
rho_c_rec = density(Cyclopentane,T=T_f_c_rec ,P=P_c_rec)
m_dot_c_rec = 0.06631 [kg/s]
G_c_rec = m_dot_c_rec/(b*W_e)
Re_c_rec = (G_c_rec*d_eq)/ ( mu_c_rec)
// Find wall viscousity mu_w using the sutherland equation
T_f_c_rec_k =338.02[K]
mu_w_rec = mu_ref *( T_f_c_rec_k/T_ref)^(3/2)*(T_ref+S)/( T_f_c_rec_k+S)



// Convection Heat trasnfer coefficient (cold side)
Nusselt_c_rec =  (  (0.0154*beta_chovron)+ 0.1298) * Re_c_rec^( (0.1892*beta_chovron)+0.6398     ) * pr_c_rec^(0.35)*(
mu_c_rec/mu_w_rec)^(0.14)
h_c_rec = (Nusselt_c_rec*k_c_rec)/(d_eq)

// Hot side-recuprator:
```

```
P_h_rec = 110[kPa]
T_h_in_rec= 107.4[C]
T_h_out_rec= 78.9 [C]
T_f_h_rec = (T_h_in_rec+T_h_out_rec)/2
cp_h_rec = cp(Cyclopentane,T=T_f_h_rec, P=P_h_rec)
k_h_rec = conductivity(Cyclopentane,T=T_f_h_rec,P=P_h_rec)
pr_h_rec = prandtl(Cyclopentane,T=T_f_h_rec,P=P_h_rec)
mu_h_rec = viscosity(Cyclopentane,T=T_f_h_rec,P=P_h_rec )
rho_h_rec = density(Cyclopentane,T=T_f_h_rec ,P=P_h_rec)
m_dot_h_rec = 0.06631 [kg/s]
G_h_rec = m_dot_h_rec/(b*W_e)
Re_h_rec = (G_h_rec*d_eq)/ ( mu_h_rec)  // Re = 56901
// Find wall viscousity mu_w using the sutherland equation
T_f_h_rec_k =366.3[K]
mu_w_rec_h = mu_ref *( T_f_h_rec_k/T_ref)^(3/2)*(T_ref+S)/( T_f_h_rec_k+S)
// Convective heat transfer coefficent for hot side of recuprator :
Nusselt_h_rec= 0.122*(pr_h_rec^(1/3)) *(mu_h_rec/mu_w_rec_h)^(1/6)*(f_h_rec *(Re_h_rec^2) * sin(2*beta_chovron) )^(
0.374)
//(1/(f_h_rec^0.5) )  =   (   cos(beta_chovron) )/  (  (  0.18 *tan(beta_chovron)  )+  (0.36*sin(beta_chovron)  )+ ( f_0/cos(
beta_chovron)    )      )^(0.5)+(  (1-cos(beta_chovron) )/(3.8*f_1)^(0.5)    )
f_0=   (   (1.81*ln(Re_h_rec) )   -   1.5)^(-2)
f_1 = 39/(Re_h_rec^0.289)
f_h_rec= 0.7883
h_h_rec = (Nusselt_h_rec*k_h_rec)/(d_eq)
// heat tranfer area of zone b (Recuprator)
Q_rec =  2926 [W]
delta1_rec = T_h_in_rec- T_c_out_rec
delta2_rec = T_h_out_rec - T_c_in_rec
LMTD_rec = (delta1_rec-delta2_rec)/ln(delta1_rec/delta2_rec)
(1/U_rec) = (1/h_h_rec)+(1/h_c_rec)+ (t_p/k_wall)
A_rec = Q_rec/(LMTD_rec*U_rec)
// Number of palte required for recuprator
A_rec = (N_p_rec-2)*L_p*W_e
// total Plates

A_total_non = A_a+A_b+A_cond
A_total_non= (N_p_total-2)*L_p*W_e
A_total_rec= A_a_rec+A_b+A_cond+A_rec
A_total_rec = (N_p_total_rec-2)*L_p*W_e
```

SOLUTION

Unit Settings: SI C kPa kJ mass deg

A_a = 24.52 [m^2]	$A_{a,rec}$ = 29.76 [m^2]
A_b = 2.397 [m^2]	A_{cond} = 1.495 [m^2]
A_{rec} = 0.3045 [m^2]	$A_{total,non}$ = 28.41
$A_{total,rec}$ = 33.96	b = 0.002 [m]
$\beta_{chovron}$ = 0.785	$cp_{c,a}$ = 2.413 [kJ/kg-C]
$cp_{c,b}$ = 3.761 [kJ/kg-C]	$cp_{c,cond}$ = 1.005 [kJ/kg-C]
$cp_{c,rec}$ = 2.011 [kJ/kg-C]	$cp_{h,a}$ = 1.015 [kJ/kg-C]
$cp_{h,b}$ = 1.037 [kJ/kg-C]	$cp_{h,cond}$ = 1.356 [kJ/kg-C]
$cp_{h,rec}$ = 1.551 [kJ/kg-C]	$\delta 1_a$ = 15 [C]
$\delta 1_b$ = 89.4 [C]	$\delta 1_{cond}$ = 5 [C]
$\delta 1_{rec}$ = 31.56 [C]	$\delta 2_a$ = 5.88 [C]
$\delta 2_b$ = 15 [C]	$\delta 2_{cond}$ = 26.79 [C]
$\delta 2_{rec}$ = 25 [C]	d_{eq} = 0.004 [m]
D_h = 0.002715 [m]	f_0 = 0.00298
f_1 = 1.648	$f_{c,cond}$ = 0.002029
$f_{h,a}$ = 0.003065	$f_{h,b}$ = 0.003175
$f_{h,rec}$ = 0.7883	$G_{c,a}$ = 131 [kg/m^2-s]
$G_{c,b}$ = 131 [kg/m^2-s]	$G_{c,b,eq}$ = 248 [kg/m^2-s]
$G_{c,cond}$ = 2316 [kg/m^2-s]	$G_{c,rec}$ = 131 [kg/m^2-s]
$G_{h,a}$ = 296.4 [kg/m^2-s]	$G_{h,b}$ = 296.4 [kg/m^2-s]
$G_{h,cond}$ = 131 [kg/s-m^2]	$G_{h,cond,eq}$ = 1558 [kg/s-m^2]

$G_{h,rec}$ = 131 [kg/m^2-s]
$h_{c,b}$ = 9221 [W/m^2-K]
$h_{c,rec}$ = 4021 [W/m^2-K]
$h_{h,b}$ = 120.1 [W/m^2-K]
$h_{h,rec}$ = 392.5 [W/m^2-K]
$k_{c,b}$ = 0.06098 [W/m-K]
$k_{c,rec}$ = 0.1151 [W/m-K]
$k_{h,b}$ = 0.04185 [W/m-K]
$k_{h,rec}$ = 0.01895 [W/(m-K)]
$LMTD_a$ = 9.738 [C]
$LMTD_{cond}$ = 41.68 [C]
L_p = 0.456 [m]
$\mu_{c,b}$ = 0.00006556 [Pa-s]
$\mu_{c,rec}$ = 0.0002539 [pa-s]
$\mu_{h,b}$ = 0.00002805 [Pa-s]
$\mu_{h,rec}$ = 0.000009212 [Pa-s]
μ_w = 0.0000229 [Pa-s]
$\mu_{w,rec,h}$ = 0.00002139 [Pa-s]
$\dot{m}_{c,b}$ = 0.06631 [kg/s]
$\dot{m}_{c,rec}$ = 0.06631 [kg/s]
$\dot{m}_{h,b}$ = 0.15 [kg/s]
$\dot{m}_{h,rec}$ = 0.06631 [kg/s]
$Nusselt_{c,cond}$ = 87.07
$Nusselt_{h,a}$ = 13.35
$Nusselt_{h,rec}$ = 82.87
$N_{p,b}$ = 22.78
$N_{p,rec}$ = 4.639
$N_{p,total,rec}$ = 296.3
ϕ = 1.473
$pr_{c,b}$ = 4.043
$pr_{c,rec}$ = 4.436
$pr_{h,b}$ = 0.6948
$pr_{h,rec}$ = 0.7539
$P_{c,b}$ = 3090 [kPa]
$P_{c,rec}$ = 3090 [kPa]
$P_{h,b}$ = 103 [kPa]
$P_{h,rec}$ = 110 [kPa]
Q_b = 11660 [W]
Q_{rec} = 2926 [W]
$Re_{c,b,eq}$ = 10272
$Re_{c,rec}$ = 2065
$Re_{h,b}$ = 42272
$Re_{h,rec}$ = 56901
$\rho_{c,b,L}$ = 463.9 [kg/m^3]
$\rho_{c,cond}$ = 1.167 [kg/m^3]
$\rho_{h,a}$ = 0.8629 [kg/m^3]
$\rho_{h,cond,L}$ = 712.4 [kg/m^3]
$\rho_{h,rec}$ = 2.598 [kg/m^3]
$T_{c,b}$ = 210.6 [C]
$T_{c,in,cond}$ = 25 [C]
$T_{c,out,a}$ = 210.6 [C]
$T_{c,out,rec}$ = 75.84 [C]
$T_{f,c,a,k}$ = 405.4 [K]
$T_{f,c,rec}$ = 64.87 [C]
$T_{f,h,a}$ = 142.7 [C]
$T_{f,h,rec}$ = 93.15 [C]
$T_{h,cond}$ = 51.79 [C]
$T_{h,in,b}$ = 300 [C]
$T_{h,out,a}$ = 59.78 [C]
$T_{h,out,rec}$ = 78.9 [C]

$h_{c,a}$ = 4504 [W/K-m^2]
$h_{c,cond}$ = 572.9 [W/m^2-K]
$h_{h,a}$ = 113.2 [W/K-m^2]
$h_{h,cond}$ = 1818 [W/K-m^2]
$k_{c,a}$ = 0.09382 [W/m-K]
$k_{c,cond}$ = 0.02632 [W/m-K]
$k_{h,a}$ = 0.03393 [W/m-K]
$k_{h,cond}$ = 0.01403 [W/m-K]
k_{wall} = 15 [W/m-K]
$LMTD_b$ = 41.68 [C]
$LMTD_{rec}$ = 28.15 [C]
$\mu_{c,a}$ = 0.00013 [Pa-s]
$\mu_{c,cond}$ = 0.00001899 [Pa-s]
$\mu_{h,a}$ = 0.00002356 [Pa-s]
$\mu_{h,cond}$ = 0.000008323 [Pa-s]
μ_{ref} = 0.0000174 [Pa-s]
$\mu_{w,rec}$ = 0.00002024 [Pa-s]
$\dot{m}_{c,a}$ = 0.06631 [kg/s]
$\dot{m}_{c,cond}$ = 1.172 [kg/s]
$\dot{m}_{h,a}$ = 0.15 [kg/s]
$\dot{m}_{h,cond}$ = 0.06631 [kg/s]
$Nusselt_{c,a}$ = 192
$Nusselt_{c,rec}$ = 139.7
$Nusselt_{h,b}$ = 11.48
$N_{p,a}$ = 214.5
$N_{p,cond}$ = 14.96
$N_{p,total}$ = 248.3
pco = 0.004 [m]
$pr_{c,a}$ = 3.344
$pr_{c,cond}$ = 0.7255
$pr_{h,a}$ = 0.705
$pr_{h,cond}$ = 0.8046
$P_{c,a}$ = 3090 [kPa]
$P_{c,cond}$ = 103.5 [kPa]
$P_{h,a}$ = 103 [kPa]
$P_{h,cond}$ = 110 [kPa]
Q_a = 25990 [W]
Q_{cond} = 25660 [W]
$Re_{c,a}$ = 4033
$Re_{c,cond}$ = 487805
$Re_{h,a}$ = 50337
$Re_{h,cond,eq}$ = 80911
$\rho_{c,a}$ = 624.3 [kg/m^3]
$\rho_{c,b,V}$ = 96.69 [kg/m^3]
$\rho_{c,rec}$ = 703.1 [kg/m^3]
$\rho_{h,b}$ = 0.6695 [kg/m^3]
$\rho_{h,cond,V}$ = 2.958 [kg/m^3]
S = 80 [K]
$T_{c,in,a}$ = 53.9 [C]
$T_{c,in,rec}$ = 53.9 [C]
$T_{c,out,cond}$ = 46.79 [C]
$T_{f,c,a}$ = 132.3 [C]
$T_{f,c,cond}$ = 35.9 [C]
$T_{f,c,rec,k}$ = 338 [K]
$T_{f,h,b}$ = 262.8 [C]
$T_{f,h,rec,k}$ = 366.3 [K]
$T_{h,in,a}$ = 225.6 [C]
$T_{h,in,rec}$ = 107.4 [C]
$T_{h,out,b}$ = 225.6 [C]
t_p = 0.002 [m]

T_{ref} = 273 [K]
U_b = 116.7 [W/m^2-C]
U_{rec} = 341.4 [W/m^2-C]
$x_{c,b}$ = 0.75
X_x = 1.571

U_a = 108.8 [W/m^2-C]
U_{cond} = 411.7 [W/m^2-C]
W_e = 0.253 [m]
$x_{h,cond}$ = 0.75

Appendix F: Heat Exchangers Design for R_245fa

SOLUTION

Unit Settings: SI C kPa kJ mass deg

A_a = 10.37	$A_{a,rec}$ = 29.76 [m^2]
A_b = 0.717	A_{cond} = 0.2376
A_{rec} = 0.09231	$A_{total,non}$ = 11.33
$A_{total,rec}$ = 30.81	b = 0.002 [m]
$\beta_{chovron}$ = 0.785	$cp_{c,a}$ = 1.462
$cp_{c,b}$ = 2.951	$cp_{c,cond}$ = 1.004
$cp_{c,rec}$ = 1.301	$cp_{h,a}$ = 1.014
$cp_{h,b}$ = 1.038	$cp_{h,cond}$ = 0.9262
$cp_{h,rec}$ = 0.9371	$\delta 1_a$ = 90.6
$\delta 1_b$ = 154.9	$\delta 1_{cond}$ = 5
$\delta 1_{rec}$ = 26.04	$\delta 2_a$ = 5.05
$\delta 2_b$ = 90.6	$\delta 2_{cond}$ = 8.003
$\delta 2_{rec}$ = 25	d_{eq} = 0.004
D_h = 0.002715	f_0 = 0.003078
f_1 = 1.726	$f_{c,cond}$ = 0.001443
$f_{h,a}$ = 0.00305	$f_{h,b}$ = 0.003179
$f_{h,rec}$ = 0.7883	$G_{c,a}$ = 324.7
$G_{c,b}$ = 324.7	$G_{c,b,eq}$ = 526.8
$G_{c,cond}$ = 21008	$G_{c,rec}$ = 131
$G_{h,a}$ = 296.4	$G_{h,b}$ = 296.4
$G_{h,cond}$ = 324.7	$G_{h,cond,eq}$ = 3619
$G_{h,rec}$ = 131	$h_{c,a}$ = 5551

$h_{c,b}$ = 9833
$h_{c,rec}$ = 2366
$h_{h,b}$ = 120.4
$h_{h,rec}$ = 282.5
$k_{c,b}$ = 0.04736
$k_{c,rec}$ = 0.08295
$k_{h,b}$ = 0.04217
$k_{h,rec}$ = 0.01542
$LMTD_a$ = 29.63
$LMTD_{cond}$ = 119.9
L_p = 0.456 [m]
$\mu_{c,b}$ = 0.00006747
$\mu_{c,rec}$ = 0.0004416
$\mu_{h,b}$ = 0.00002823
$\mu_{h,rec}$ = 0.00001083
μ_w = 0.00001827
$\mu_{w,rec,h}$ = 0.00001939
$\dot{m}_{c,b}$ = 0.1643 [kg/s]
$\dot{m}_{c,rec}$ = 0.06631 [kg/s]
$\dot{m}_{h,b}$ = 0.15 [kg/s]
$\dot{m}_{h,rec}$ = 0.06631 [kg/s]
$Nusselt_{c,cond}$ = 599.7
$Nusselt_{h,a}$ = 13.63
$Nusselt_{h,rec}$ = 73.29
$N_{p,b}$ = 8.215
$N_{p,rec}$ = 2.8
$N_{p,total,rec}$ = 269
ϕ = 1.473
$pr_{c,b}$ = 4.204
$pr_{c,rec}$ = 6.926
$pr_{h,b}$ = 0.6946
$pr_{h,rec}$ = 0.6581
$P_{c,b}$ = 3090 [kPa]
$P_{c,rec}$ = 3090 [kPa]
$P_{h,b}$ = 103 [kPa]
$P_{h,rec}$ = 110 [kPa]
Q_b = 10065 [W]
Q_{rec} = 575.1 [W]
$Re_{c,b,eq}$ = 21198
$Re_{c,rec}$ = 1187
$Re_{h,b}$ = 42006
$Re_{h,rec}$ = 48416
$\rho_{c,b,L}$ = 812.6
$\rho_{c,cond}$ = 1.27
$\rho_{h,a}$ = 0.8918
$\rho_{h,cond,L}$ = 1359
$\rho_{h,rec}$ = 5.776
$T_{c,b}$ = 145.1 [C]
$T_{c,in,cond}$ = 9.257 [C]
$T_{c,out,a}$ = 145.1 [C]
$T_{c,out,rec}$ = 20.37 [C]
$T_{f,c,a,k}$ = 292.2 [K]
$T_{f,c,rec}$ = 19.03
$T_{f,h,a}$ = 129.2
$T_{f,h,rec}$ = 44.55
$T_{h,cond}$ = 17.26 [C]
$T_{h,in,b}$ = 300 [C]
$T_{h,out,a}$ = 22.69 [C]
$T_{h,out,rec}$ = 42.69 [C]
T_{ref} = 273 [K]

$h_{c,cond}$ = 3665
$h_{h,a}$ = 112.4
$h_{h,cond}$ = 2068
$k_{c,a}$ = 0.06475
$k_{c,cond}$ = 0.02445
$k_{h,a}$ = 0.03299
$k_{h,cond}$ = 0.01332
k_{wall} = 15 [W/m-K]
$LMTD_b$ = 119.9
$LMTD_{rec}$ = 25.52
$\mu_{c,a}$ = 0.0001951
$\mu_{c,cond}$ = 0.00001781
$\mu_{h,a}$ = 0.00002301
$\mu_{h,cond}$ = 0.000009853
μ_{ref} = 0.0000174 [Pa-s]
$\mu_{w,rec}$ = 0.00002024
$\dot{m}_{c,a}$ = 0.1643 [kg/s]
$\dot{m}_{c,cond}$ = 10.63 [kg/s]
$\dot{m}_{h,a}$ = 0.15 [kg/s]
$\dot{m}_{h,cond}$ = 0.1643 [kg/s]
$Nusselt_{c,a}$ = 343
$Nusselt_{c,rec}$ = 114.1
$Nusselt_{h,b}$ = 11.42
$N_{p,a}$ = 91.92
$N_{p,cond}$ = 4.06
$N_{p,total}$ = 100.2
pco = 0.004 [m]
$pr_{c,a}$ = 4.406
$pr_{c,cond}$ = 0.7316
$pr_{h,a}$ = 0.707
$pr_{h,cond}$ = 0.6848
$P_{c,a}$ = 3090 [kPa]
$P_{c,cond}$ = 103.5 [kPa]
$P_{h,a}$ = 103 [kPa]
$P_{h,cond}$ = 110 [kPa]
Q_a = 33389 [W]
Q_{cond} = 32020 [W]
$Re_{c,a}$ = 6656
$Re_{c,cond}$ = 4.717E+06
$Re_{h,a}$ = 51524
$Re_{h,cond,eq}$ = 145158
$\rho_{c,a}$ = 1181
$\rho_{c,b,V}$ = 242.7
$\rho_{c,rec}$ = 1363
$\rho_{h,b}$ = 0.6633
$\rho_{h,cond,V}$ = 6.442
S = 80 [K]
$T_{c,in,a}$ = 17.64 [C]
$T_{c,in,rec}$ = 17.69 [C]
$T_{c,out,cond}$ = 12.26 [C]
$T_{f,c,a}$ = 81.37
$T_{f,c,cond}$ = 10.76
$T_{f,c,rec,k}$ = 338 [K]
$T_{f,h,b}$ = 267.9
$T_{f,h,rec,k}$ = 317.7 [K]
$T_{h,in,a}$ = 235.7 [C]
$T_{h,in,rec}$ = 46.41 [C]
$T_{h,out,b}$ = 235.7 [C]
t_p = 0.002 [m]
U_a = 108.6

U_b = 117.1
U_{rec} = 244.1
$x_{c,b}$ = 0.75
X_x = 1.571

U_{cond} = 1124
W_a = 0.253 [m]
$x_{h,cond}$ = 0.75