

**Madan Mohan Malaviya University of Technology,
Gorakhpur-273010 (UP)**



**Project Report
on**

**Thermal analysis of Combined ORC & VCR Cycle
using EES**

Under the Guidance of

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APPROVAL SHEET

This thesis is entitled “Thermal analysis of Combined ORC & VCR Cycle using EES” by Vinay Kumar Yadav (2021051163), Atulya Vaibhav Pandey (2021051017), Prem Narayan Patel (2021051043), Divyansh Jaiswal (2021051023), and Abhishek Kumar (2021051102) meets the accepted standards for originality and is approved for the degree of Bachelor of Technology, Mechanical Engineering.

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ABSTRACT

This report presents a thermal analysis of a Combined Organic Rankine Cycle (ORC) and Vapor Compression Refrigeration (VCR) system, utilizing Engineering Equation Solver (EES) software to enhance energy efficiency and reduce environmental impact. A comparative study of various organic working fluids was conducted, evaluating their thermodynamic performance, safety, and environmental impact on the system's overall efficiency.

The EES software was used to systematically solve the governing equations and perform parametric studies over a wide range of evaporator temperatures, condenser temperatures, and boiler pressures. Graphical representations illustrate the trends of mass flow rate per kilowatt of cooling and overall Coefficient of Performance (COP) as functions of these parameters.

The analysis focused on environmentally friendly working fluids with low Global Warming Potential (GWP) and zero Ozone Depletion Potential (ODP). The findings provide insights into selecting the most suitable working fluid and offer thermodynamic data that can inform future studies and practical applications.

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CHAPTER-1: INTRODUCTION

The escalating global energy crisis, coupled with increasing environmental concerns, has driven significant research into waste heat recovery systems and energy efficiency improvements across industrial applications. Diesel engine operations, which waste approximately 60-70% of fuel energy as heat through exhaust gases and cooling systems, are of particular interest. This presents a compelling opportunity for energy recovery and utilization through innovative system integration.

The proposed Combined Organic Rankine Cycle (ORC) and Vapor Compression Refrigeration (VCR) configuration represents a novel approach to harnessing this waste thermal energy from diesel engine effluents. In conventional diesel engine operations, exhaust gases typically exit at temperatures ranging from 200°C to 400°C, while the engine cooling system releases heat at temperatures between 80°C to 95°C. These waste heat streams contain significant thermal energy that can be effectively utilized in a properly designed combined cycle system. [1]

The integration concept involves utilizing the high-temperature exhaust gases as the primary heat source for the ORC system. Additionally, the engine's cooling system waste heat can be used as a secondary heat source, further improving the overall system efficiency. This cascading energy utilization approach enables the creation of a self-sustaining cooling system that operates primarily on waste heat, significantly reducing the external power requirement for refrigeration needs.

This configuration offers multiple advantages over conventional systems. First, it addresses the significant energy wastage in diesel engine operations by recovering thermal energy that would otherwise be released to the environment. Second, it provides a means of generating useful refrigeration effect with minimal additional energy input, improving the overall energy efficiency of the combined system. Third, the integration helps reduce the environmental impact by lowering both direct emissions from the diesel engine and indirect emissions associated with conventional electricity-driven refrigeration systems.

The thermodynamic synergy between the diesel engine, ORC, and VCR components creates a sophisticated energy recovery and utilization system. The ORC subsystem, operating with carefully selected organic working fluids suitable for the available temperature ranges, converts the waste heat into mechanical power. This mechanical power is then utilized to drive the VCR

system compressor, creating a cooling effect that can be used for various applications, from space cooling to process refrigeration. [2]

The success of this integrated system depends heavily on optimal component sizing, working fluid selection, and control strategy implementation. The design must account for the varying nature of diesel engine operation, fluctuations in waste heat availability, and the dynamic cooling demands of the application. Furthermore, the system must be engineered to maintain stable operation across different load conditions while maximizing energy recovery and utilization efficiency.

This thesis provides a comprehensive analysis of the combined VCR-ORC system powered by diesel engine waste heat, examining the theoretical foundations, design considerations, performance characteristics, and practical implementation challenges. The investigation encompasses both steady-state and dynamic operation scenarios, considering various operating parameters and their impact on system performance. Through this analysis, we aim to demonstrate the viability and benefits of this innovative approach to waste heat recovery and utilization in industrial applications. Moreover, multiple working fluids were used for analysis and their comparative study was performed to choose more suitable for practical and feasible systems.

The subsequent sections will delve into detailed system analysis, component design considerations, performance optimization strategies, and economic viability assessment, providing a thorough understanding of this promising technology for sustainable energy utilization.

CHAPTER-2: LITERATURE REVIEW

2.1: NEED FOR COMBINED CYCLE

The VCR cycle based system faces several significant limitations. Since these systems need a lot of electrical power to operate the compressor, energy consumption is still a major concern. This high energy demand results in significant operating expenses, especially in areas with high electricity prices. These systems have the potential to seriously strain electrical infrastructure during times of high demand.

Another significant constraint is environmental concerns. It is impossible to overlook the possibility of refrigerant leaks and the ensuing harm to the environment. There is growing regulatory pressure on refrigerant handling and selection because many of the refrigerants used in VCR systems contribute to global warming.

Another major challenge is waste heat management. With few options for waste heat recovery, VCR systems release significant amounts of heat into the atmosphere. Operating in warm climates is made more difficult by the fact that system efficiency usually declines in high ambient temperature conditions.

Likewise, ORC based systems also face several significant limitations. The initial investment required for these systems presents a substantial barrier to adoption. High capital costs for system components, complex installation requirements, and significant space needs can make implementation challenging, particularly for smaller operations.

Performance constraints also pose significant challenges. ORC systems typically demonstrate lower thermal efficiency compared to traditional power cycles. Their sensitivity to heat source temperature fluctuations can impact reliability, and small-scale applications often face limitations in power output capability.

Technical challenges further complicate ORC implementation. The critical nature of working fluid selection, the need for component optimization, and increased maintenance complexity all require careful consideration and expertise. These factors can impact both initial implementation and ongoing operation.

The synergy between diesel engines, organic Rankine cycles (ORCs), and vapor compression

refrigeration (VCR) systems within a combined cycle system offers several compelling advantages. Firstly, the waste heat generated by the diesel engine, which would otherwise be dissipated into the environment, is harnessed by the ORC to generate additional power. This significantly improves the overall energy efficiency of the system, reducing fuel consumption and minimizing environmental impact.

Secondly, the integration of VCR and ORC systems enables the simultaneous provision of cooling and electricity. The waste heat from the ORC can be utilized to drive the refrigeration cycle, thereby reducing the energy input required for cooling. Additionally, the ORC's ability to operate efficiently at low-temperature heat sources can help maintain the VCR's performance, even in high ambient temperature conditions. This ensures consistent cooling capacity throughout the year. [2]

The combined system transforms the waste heat limitation of VCR system into an opportunity by utilizing the condenser heat as the primary energy source for the ORC system. This integration enables the recovery of 130-150 kW of thermal energy (for a 100 kW cooling capacity system) that would otherwise be wasted, channeling it into useful power generation through the ORC.

The energy efficiency enhancement achieved through this integration is remarkable. The ORC generates electricity using the recovered waste heat, which partially powers the VCR system compressor, resulting in a net power consumption reduction of 25-35%. This synergistic relationship improves the overall system coefficient of performance (COP) from a typical value of 3.3 for VCR system to approximately 5.0 for the combined system, representing a significant efficiency improvement.

Economic viability sees substantial improvement in the combined configuration. The shared infrastructure reduces total capital costs compared to implementing separate systems, while the generation of multiple utility outputs (both cooling and power) from a single installation enhances the value proposition. Operating costs decrease significantly through the self-generated power, resulting in typical payback periods of 4-7 years, compared to 8-10 years for ORC systems.

The combined system creates important technical synergies that improve overall performance.

The temperature matching between the VCR condenser and ORC evaporator achieves optimal efficiency, while the possibility of using compatible working fluids simplifies system design and maintenance. The integration also enables more stable operation through consistent heat source temperature provision to the ORC, improving part-load performance and simplifying control strategies.

Environmental benefits emerge as a significant advantage of the combined system. The lower net electricity consumption directly reduces indirect carbon emissions, while the cascaded energy use minimizes thermal pollution. By optimizing the utilization of energy and minimizing waste heat, these systems can contribute to a more sustainable energy landscape. The more efficient use of refrigerants and working fluids, combined with improved resource utilization, results in a substantially reduced environmental footprint compared to individual systems. Furthermore, combined cycle systems have the potential to utilize more environmentally friendly refrigerants, further reducing their environmental impact.

The combined cycle systems offer a promising solution to meet the growing demand for energy while minimizing environmental impact. By effectively integrating multiple power generation cycles, these systems can significantly improve energy efficiency, reduce emissions, and provide flexible and reliable energy solutions. As technology continues to advance and the need for sustainable energy intensifies, combined cycle systems are poised to play a vital role in shaping the future of energy generation.

The key integration points between the diesel engine, ORC, and VCR cycles are:

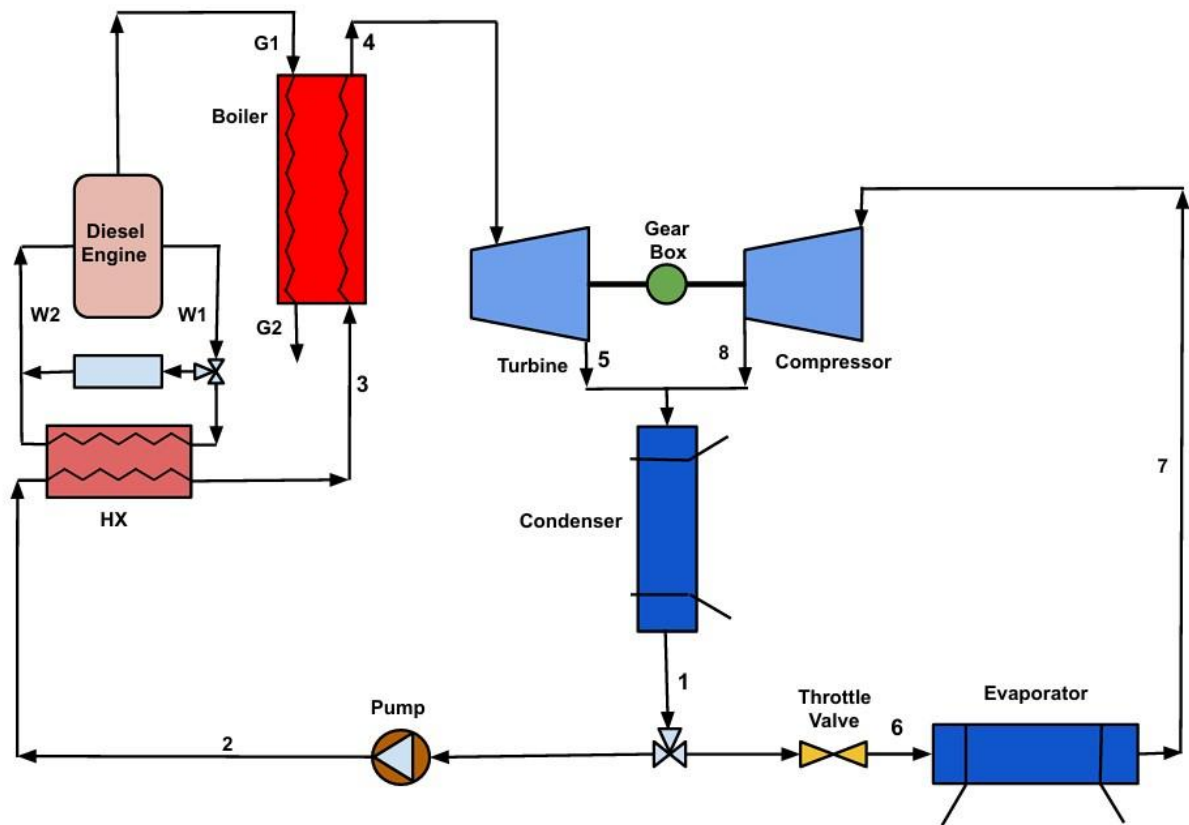
- The diesel engine's waste heat serves as a common heat source for both the ORC and VCR.
- The ORC and VCR share a common condenser, reducing the overall heat rejection to the environment.
- The ORC turbine is directly coupled to the VCR compressor, eliminating the need for an electrical motor and associated energy losses.

By integrating these cycles, the system achieves significant energy savings, reduced emissions, and enhanced overall efficiency. The efficient utilization of waste heat, the elimination of energy losses associated with electrical motors, and the synergistic operation of the cycles contribute to the system's superior performance. [1]

2.2: WORKING PRINCIPLE

The proposed system presents a novel approach to enhance energy efficiency and environmental sustainability by integrating a diesel engine, an organic Rankine cycle (ORC), and a vapor compression refrigeration (VCR) cycle. This integration enables the recovery of waste heat from the diesel engine to drive the ORC and VCR, resulting in significant energy savings and reduced emissions.

The diesel engine's cooling water is heated by a heat exchanger, preheating the condensed working fluid before entering the boiler. Hot exhaust gases vaporize the ORC working fluid, expanding it to produce shaft work. The vapor compression cycle uses a compressor directly coupled to the expander, eliminating conversion losses. The high pressure vapor undergoes condensing and evaporation processes before completing the loop as shown in Fig. 1.



Fluid Stream: 1-2-3-4-5-6-7-8 = Working fluid ; g1- g2= Exhaust Gas ; w2-w1= Cooling Water

Fig. 1: Schematic Diagram of Combined VCR and ORC System [3]

The diesel engine, a prime mover in the system, generates mechanical power through the combustion of fuel. A substantial portion of the energy released during combustion is converted into useful work, while the remaining energy is dissipated as waste heat in the form of exhaust

gases and cooling water.

The cooling water from the diesel engine, which is typically discharged at elevated temperatures, is passed through a heat exchanger. This heat exchanger preheats the condensed working fluid from the ORC condenser, reducing the energy input required to vaporize the fluid in the boiler. This preheating process further improves the overall thermal efficiency of the system.

The hot exhaust gases from the diesel engine are routed through a boiler. The thermal energy in these gases is transferred to the working fluid of the ORC, causing it to vaporize. The vaporized working fluid, now at high pressure and temperature, is expanded through a turbine to generate mechanical work.

The waste heat from the diesel engine is utilized to drive both the ORC and VCR. The ORC and VCR share a common condenser, reducing heat rejection to the environment. The mechanical power generated by the ORC turbine is used to drive the VCR compressor, eliminating the need for an additional electrical motor. [1] [2] [3]

2.3: COMPONENTS & THEIR FUNCTIONS

1. Diesel Engine: It generates mechanical power through combustion. Waste heat is released in the form of exhaust gases (g_2-g_1) and cooling water (w_2-w_1).
2. Heat Exchanger (HX): The cooling water from the diesel engine preheats the working fluid of the ORC before it enters the boiler.
3. Boiler: The exhaust gases heat the working fluid in the boiler, causing it to vaporize.
4. Turbine: The high-pressure vapor expands through the turbine, generating mechanical power.
5. Compressor: The compressor increases the pressure of the working fluids in the VCR cycle. The gearbox transfers the mechanical power from the turbine to the compressor via a coupled shaft.
6. Condenser: The high-pressure working fluids condenses in the condenser, releasing heat to the environment or to the ORC.

7. Evaporator: The low-pressure working fluids evaporates in the evaporator, absorbing heat from the cooling load.
8. Throttle Valve: The throttle valve reduces the pressure of the working fluids before it enters the evaporator.
9. Pump: The pump circulates the working fluid in condensed form back to the boiler.

2.4: THERMODYNAMIC PROCESSES

The following assumptions are made in order to make the analysis simpler:

- Every component takes into account steady-state conditions;
- The system's heat and friction losses are disregarded

The thermodynamic processes in the cycle are as follows:

- 1 → 2: Actual pumping work;
- 2 → 3: Heat addition in the HX (preheating);
- 3 → 4: Isobaric Heat addition in the boiler;
- 4 → 5: Actual expansion in the expander (turbine);
- 5 → 1: Heat rejection (in condenser) in the ORC;
- 1 → 6: Isenthalpic expansion in the throttle valve;
- 6 → 7: Heat absorption (in evaporator) in the VCR cycle;
- 7 → 8: Actual compression in the compressor;
- 8 → 1: Heat rejection (in condenser) in the VCR cycle; [3]

2.5: WORKING FLUIDS

For analysis and optimization, four (hydrocarbons or) working fluids R600 (Butane), R600a (Isobutane), R1234ze (Solstice ZE) and R1234yf (Solstice YF), were chosen due to their high molecular weight, high critical temperature, high critical pressure, ignition temperature, low global warming potential (GWP) and, low ozone depletion potential (ODP). A table with the thermodynamic properties is given below:

Table 1: Properties of Working Fluid [2]

Working Fluid	Mol. Weight	Critical pressure (kPa)	Critical Temperature (C)	ODP	GWP	Ignition Temperature (C)
Butane (R600)	58.1	3796	152.01	0	4	405
Isobutane (R600a)	58.1	3640	134.7	0	3	462
Solstice ZE (R1234ze)	114	3636	109.4	0	7	368
Solstice YF (R1234yf)	114	3381	94.7	0	4	405

2.5.1: Butane (R600)

Butane, also known as R600, is a hydrocarbon refrigerant with a simple molecular structure and chemical formula, C₄H₁₀. Butane (R600) is a natural refrigerant with a negligible GWP, making it an environmentally friendly option. It offers good energy efficiency, reducing energy consumption and operating costs. Its natural origin and excellent heat transfer properties further enhance its appeal.

However, its high flammability poses significant safety risks, limiting its use to smaller-scale applications. Additionally, its potential odor can be a drawback in certain scenarios. Despite these limitations, butane is primarily used in domestic refrigerators and small commercial refrigeration systems where strict safety measures can be implemented to mitigate risks. [1]

2.5.2: Isobutane (R600a)

R600a, another name for Isobutane, is a hydrocarbon refrigerant with the chemical formula C₄H₁₀ and an elementary molecular structure. It is an environmentally friendly choice as it is a natural refrigerant with a very low GWP. Excellent heat transfer qualities, the ability to function at moderate pressures, and good energy efficiency are further benefits that lower operating costs and energy consumption.

Its use is confined to smaller-scale applications due to the serious safety risks posed by its high flammability. Its possible odor may also be a disadvantage in some situations. Irrespective of these drawbacks, it is mostly utilized in small commercial refrigeration systems and residential refrigerators where stringent safety precautions can be taken to reduce hazards. [1]

2.5.3: Solstice ZE

Solstice ZE (R1234ze), also known as HFO-1234ze(E), is a hydrofluoro-olefin (HFO) refrigerant with a low Global Warming Potential (GWP) of less than 1. This makes it a significantly more environmentally friendly alternative to traditional refrigerants like R-134a. Solstice R1234ze offers several advantages, including its ultra-low GWP, which significantly reduces its impact on climate change, and its high energy efficiency, leading to lower operating costs. It is also mildly flammable, making it relatively safe to handle when proper precautions are taken.

However, it has some disadvantages, such as its higher cost compared to traditional refrigerants and its limited availability in certain regions. Additionally, its mild flammability requires specialized handling and storage. Despite these drawbacks, R1234ze is widely used in various applications, including automotive air conditioning, commercial refrigeration, and chillers, contributing to a more sustainable future for refrigeration and air conditioning technologies. [4]

2.5.4: Solstice YF

Solstice YF (R1234yf), also known as HFO-1234yf, is a hydrofluoro-olefin (HFO) refrigerant with a low Global Warming Potential (GWP) and is a promising refrigerant that offers significant environmental benefits and good performance characteristics. Along with low GWP, it offers several other advantages like high efficiency which reduces operating costs, operation at moderate pressure simplifying system design and reducing the need for heavy-duty equipment and compatibility with a range of lubricants, ensuring smooth operation of refrigeration systems which helps maintain system performance and longevity.

Its limitations include mild flammability which requires careful handling and storage to mitigate fire risks, higher cost and limited availability.

It finds its use in automotive air conditioning, commercial refrigeration and chillers contributing to reduced greenhouse gas emissions and offering energy-efficient and environmentally friendly cooling solutions. [4]

CHAPTER-3: OBJECTIVE

The goal is to design and analyze the combined Organic Rankine Cycle (ORC) and Vapor Compression Refrigeration (VCR) system that can effectively utilize this waste energy from a diesel engine and determine if this combined system can significantly improve energy utilization and provide both power generation and cooling capabilities, thereby enhancing the overall efficiency of diesel engine applications. Also, the effect of different organic working fluids, namely R600 (Butane), R600a (Isobutane), R1234ze (Solstice ZE) and R1234yf (Solstice YF), on thermodynamic properties and their environmental impact of the combined cycle should be investigated.

The problem involves thermal analysis using EES of system based on Combined VCR and ORC cycle which utilizes waste energy from Diesel Engine for:

1. Utilization of the waste energy streams from the Diesel Engine
2. Designing an integrated ORC-VCR system to recover and utilize this waste energy
3. Performing a comprehensive thermodynamic analysis of the combined system for four different hydrocarbon working fluids, namely R600 (Butane), R600a (Isobutane), R1234ze (Solstice ZE) and R1234yf (Solstice YF);

3.1: PROBLEM STATEMENT

A system/plant working on the Combined ORC & VCR cycle utilizes heat from exhaust emissions and cooling water of Diesel engine in which exhaust gases before and after passing through boiler are at 220 °C and 120 °C respectively; and cooling water (preheating the working fluid) enters and leaves radiator at 90 °C and 80 °C respectively.

Assuming every component takes into account steady-state conditions and neglecting system's heat and friction losses and taking efficiency of pump, turbine/expander and condenser be 0.75, 0.80 and 0.75 respectively, find and plot variation of mass flow rate of working fluid per kW of cooling capacity (MKW) and overall COP ($COP_{overall}$) with boiler pressure (between 2000kPa and below the critical working fluid pressure), evaporator temperature (between -15°C and 5°C) and condenser temperature (between 30°C and 55°C) for each abovementioned refrigerant.

Take specific heat capacities of cooling water and exhausts emissions be 4.18 and 1.005 kJ/kg-°C respectively and mass flow rate of cooling water and exhausts emissions be 2.946 and 4.669 kg/s respectively.

CHAPTER-4: MATHEMATICAL FORMULATIONS

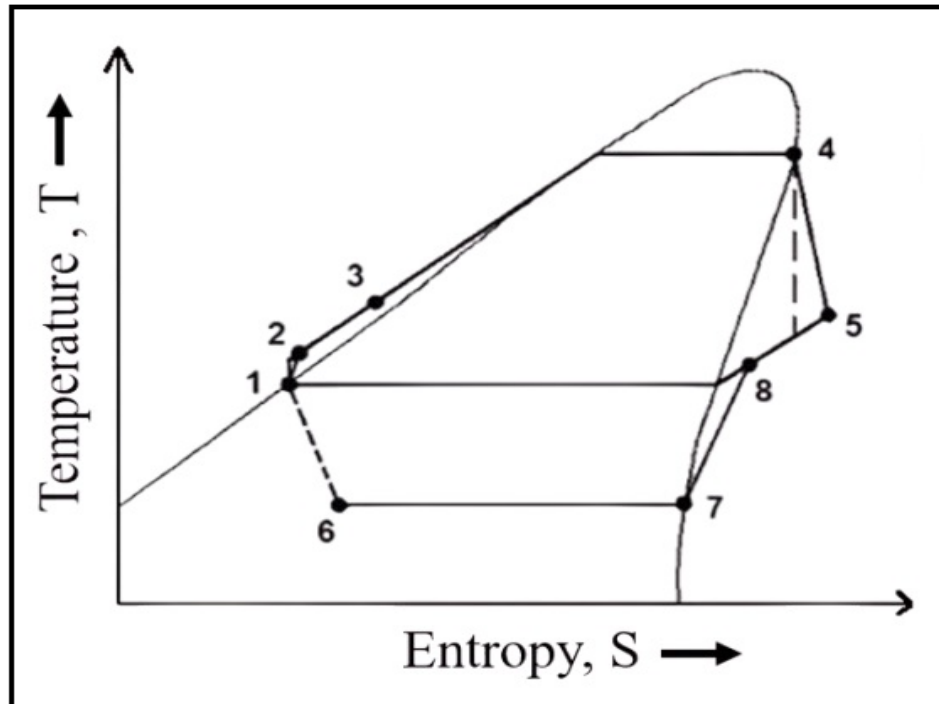


Fig 2: Plot of Temperature vs Entropy [2]

The mathematical formulations associated with the thermodynamic processes are as follows:

For Organic Rankine cycle (ORC) [1] [2]:

Process 1-2: Pump work input, (\dot{W}_p)

$$\Rightarrow \dot{W}_p = \frac{\dot{m}_{orc}(h_2 - h_1)}{\eta_p}$$

where, \dot{m}_{orc} = mass flow rate of working fluid in ORC cycle,

$$\eta_p = \text{efficiency of pump}$$

Process 2-3: Heat absorbed in heat exchanger from cooling water, (\dot{Q}_{cw}) (Process w1-w2)

$$\dot{Q}_{cw} = \dot{m}_{cw} c_{pw} (T_{cw_{in}} - T_{cw_{out}})$$

$$\dot{Q}_{cw} = \dot{m}_{orc}(h_3 - h_2)$$

where, \dot{m}_{cw} = mass flow rate of cooling water (cw),

$$T_{cw_{in}} = \text{temperature of cooling water at inlet,}$$
$$T_{cw_{out}} = \text{temperature of cooling water at outlet,}$$
$$c_{pw} = \text{specific heat capacity of water at constant pressure}$$

Process 3-4: **Heat absorbed in boiler from exhaust gases (Process g1-g2)**

$$\dot{Q}_{eg} = \dot{m}_{orc}(h_4 - h_3) = \dot{m}_g c_{pg}(T_{g1} - T_{g2})$$

where, \dot{m}_g = mass flow rate of exhaust gases (g),

T_{g1} = temperature of exhaust gases at inlet,

T_{g2} = temperature of exhaust gases at outlet,

c_{pg} = specific heat capacity of exhaust gases

Process 4-5: **Turbine work output, (\dot{W}_t)**

$$\dot{W}_t = \dot{m}_{orc}(h_4 - h_5)\eta_e$$

where, η_e = efficiency of turbine

Process 5-1: **Heat rejected to condenser, ($\dot{Q}_{c,orc}$)**

$$\dot{Q}_{c,orc} = \dot{m}_{orc}(h_5 - h_1)$$

The net work done of the ORC is defined as:

$$\dot{W}_{net\ orc} = \dot{W}_t - \dot{W}_p$$

The thermal efficiency of the ORC (η_{orc}) is defined as:

$$\eta_{orc} = \frac{\dot{W}_{net\ orc}}{\dot{Q}_{cw} + \dot{Q}_{eg}} [1] [2]$$

For Vapour Compression Refrigeration Cycle (VCR) [1] [2]:

Process 1-6: **Isenthalpic expansion in throttling valve**

$$h_1 = h_6$$

Process 6-7: **Refrigeration effect, (\dot{Q}_e)**

$$\dot{Q}_e = \dot{m}_{vcr}(h_7 - h_6)$$

where, \dot{m}_{vcr} = mass flow rate of working fluid in VCR cycle

Process 7-8: **Compressor work input, (\dot{W}_c)**

$$\dot{W}_c = \frac{\dot{m}_{vcr}(h_8 - h_7)}{\eta_c} = \dot{W}_{net\ vcr}$$

Process 8-1: **Heat rejected to condenser, ($\dot{Q}_{c,vcr}$)**

$$\dot{Q}_{c,vcr} = \dot{m}_{vcr}(h_8 - h_1)$$

The COP of the VCR (COP_{vcr}) is defined as:

$$COP_{vcr} = \frac{\dot{Q}_e}{\dot{W}_{net\ vcr}}$$

The overall COP ($COP_{overall}$) is defined as:

$$COP_{overall} = \eta_{orc} COP_{vcr}$$

The mass flow rate of working fluid per kW of cooling capacity, (MKW)

$$MKW = \frac{(\dot{m}_{orc} + \dot{m}_{vcr})}{\dot{Q}_e}$$

The expansion ratio across the expander (EPR) and the compression ratio across the compressor (CMR), which are proportional to the expander and compressor sizes, respectively:

$$EPR = \frac{v_5}{v_4}$$

$$CMR = \frac{p_8}{p_7} [1] [2]$$

CHAPTER-5: ABOUT ENGINEERING EQUATION SOLVER

5.1: INTRODUCTION

- EES (pronounced 'ease') is an acronym for Engineering Equation Solver.
- The basic function provided by EES is the numerical solution of linear and non-linear algebraic and differential equations.
- It is important to recognize that EES is an equation-solver. EES utilizes equations rather than the assignments that are used in a formal programming language.
- EES solves systems of equations (i.e., relationships between variables) automatically, which frees the user from having to develop their own iterative technique for solving a set of non-linear equations.[5]

5.2: CREATING PARAMETRIC TABLE

- a) A table is generated using the New Parametric Table command in the Table menu. The variables which are to appear in the table are selected from a list of variables currently appearing in the Equations window. Each new Parametric Table must be given a name by the user. The name appears in the tabs at the top of the Parametric Table Window.
- b) Variables may be added to or deleted from a selected Parametric Table using the Insert/Delete bars command in the Table menu. This menu can also be accessed by clicking the right mouse button in the column name header Cell (topmost row) and selecting Add Column from the pop-up menu that appears.
- c) The display format and units of the variable in each column of the table can be changed by clicking the right mouse button on the name at the top of the column. A pop-up menu will appear having Properties as one of the menu choices. If Properties is selected, the Format Table dialog window will appear in which the display format, units, column width, and column position can be entered.
- d) Values in a column of the Parametric Table can be entered, cleared, or changed by clicking the right mouse button within the column header cell. A pop-up menu will appear having Alter Values as one of the menu choices. Selecting Alter Values will bring up the Change Table Column Values dialog which provides a very easy way to enter or clear values in the column.
- e) Each row of the Parametric Table is a separate calculation that is initiated by the Solve Table command. The number of rows is selected when the table is generated. However rows can be inserted or deleted by clicking the right mouse button in the first column of the table

on the row for which the insertion or deletion is to occur.

f) A range of cells in the table can be copied to the Clipboard, after which they can be pasted into a different part of the table, into the Lookup Table, or into another application such as a word processor or a spreadsheet. One or more rows in the table can be selected by clicking the left mouse button in the row header column (the leftmost column). If the mouse is dragged to other rows while the left mouse button remains down, these rows will be added to the selection. A similar procedure can be use used to select one or more columns.[5]

5.3: CREATING DIAGRAM WINDOW

The Diagram window can be used in several ways:

- i. It provides a place to display graphic objects or text relating to the problem that is being solved.
- ii. The Diagram window can be used for both input and output of information thereby providing a graphic user interface.
- iii. In the Professional license, the Diagram Window is the focus of the animation capabilities.

Importantly, the Diagram Window is the central focus for Distributable programs. The graphics placed in the Diagram window can be simple shapes such as lines, circles or boxes that are drawn using the tools in EES or they can be complex graphics such as a picture or a scanned image copied from another application and pasted into the Diagram window.

In the Professional license, formatted text, including equations, can be copied from a word processor and objects can be animated using the values of EES variables. The Diagram window has development and application modes. Changes in the Diagram window are made in the development mode. Check boxes and radio button controls can be placed on the Diagram window which, when selected, execute specified EES commands and equations.

The Diagram window is commonly used to display a schematic diagram of a system identifying state point locations to help interpret the equations in the Equations window. A Calculate button can be placed on the Diagram window to initiate the calculations. Input data can be read from the Diagram window and used in the calculations. The selected outputs can then be displayed on the Diagram. [5]

5.4: INPUT

" ORC-VCR Combined System Analysis in EES "

$m_g = 4.669$ "Exhaust gas mass flow rate in kg/s"

$T_{g1} = 220$ "Boiler entry temperature of exhaust gases in °C"

$T_{g2} = 120$ "Boiler exit temperature of exhaust gases in °C"

$m_{cw} = 2.946$ "Cooling water mass flow rate in kg/s"

$T_{cw_in} = 90$ "Cooling water temperature from the diesel engine in °C"

$T_{cw_out} = 80$ "Cooling water temperature to the diesel engine in °C"

$\eta_p = 0.75$ "Pump isentropic efficiency"

$\eta_e = 0.80$ "Expander isentropic efficiency"

$\eta_c = 0.75$ "Compressor isentropic efficiency"

\$ifnot ParametricTable

$P_{boil} = 3441$ "Boiler pressure in kPa (Change in parametric table) , 2000 to 3796"

\$endif

$T_{cond} = 35$ "Condenser temperature in °C , 30 to 55"

$T_{evap} = -5$ "Evaporator temperature in °C , -15 to 5"

" Define Fluid Properties for Selected Refrigerant "

Fluid\$ = 'R600' "Working fluid: Butane"

$T_{crit} = t_{crit}(R600)$ "Critical temperature for R600 in °C"

$P_{crit} = p_{crit}(R600)$ "Critical pressure for R600 in kPa"

"Process 1-2 "

$h1 = \text{enthalpy}(\text{Fluid}\$, T=T_{cond}, x=0)$ "Enthalpy at state 1 (saturated liquid)"

$s1 = \text{entropy}(\text{Fluid}\$, T=T_{cond}, x=0)$ "Entropy at state 1"

$p1 = \text{pressure}(\text{Fluid}\$, T=T_{cond}, x=0)$ "Pressure at state 1"

$W_p = m_{orc} * (h2 - h1)/\eta_p$ "Pump work in kW"

"Process 2-3 "

$P2 = P_{boil}$ "Boiler pressure in kPa"

$h2s = \text{enthalpy}(\text{Fluid}\$, P=P2, s=s1)$ "Isentropic enthalpy at state 2"

$h2 = h1 + (h2s - h1)/\eta_p$ "Actual enthalpy at state 2"

$T2 = \text{temperature}(\text{Fluid}\$, P=P2, h=h2)$

$cp_w=4.18$ "Specific heat at constant pressure of water"

$Q_cw=m_cw*cp_w*(T_cw_in-T_cw_out)$ "Heat absorbed in heat exchanger from cooling water"

$T3 = \text{temperature}(\text{Fluid}\$, h=h3, P=P2)$

$Q_cw = m_orc * (h3 - h2)$ "Heat absorbed from cooling water in kW"

"Process 3-4"

$h4 = \text{enthalpy}(\text{Fluid}\$, x=1, P=P2)$ "Enthalpy at state 4"

$s4 = \text{entropy}(\text{Fluid}\$, x=1, P=P2)$ "Entropy at state 4"

$T4 = \text{temperature}(\text{Fluid}\$, x=1, P=P2)$

$cp_g=1.005$ "Specific heat at constant pressure of exhaust gas"

$Q_eg = m_g * cp_g * (T_g1 - T_g2)$ "Heat absorbed in boiler from exhaust gases in kW"

$m_orc = Q_eg / (h4 - h3)$ "ORC mass flow rate based on energy balance"

"Process 4-5"

$s4=s5s$

$h5s=\text{enthalpy}(\text{Fluid}\$, P=P1, s=s5s)$ "Enthalpy at state 5 (saturated vapor) "

$h5 = h4 - ((h4 - h5s) * \eta_{e})$

$T5= \text{temperature}(\text{Fluid}\$, P=P1, h=h5)$

$W_t = m_orc * (h4 - h5s)$ "Turbine work output in kW"

"Process 5-1 "

$Q_c_orc = m_orc * (h5 - h1)$ "Heat rejection in condenser (ORC side) in kW "

$W_net_orc = W_t - W_p$ "Net work of the ORC in kW"

$\eta_{orc} = W_net_orc / (Q_eg + Q_cw)$ "ORC thermal efficiency"

" VCR Cycle "

"Process 1 - 6"

$h6 = h1$ "Isenthalpic expansion"

"Process 6 - 7"

$h7 = \text{enthalpy}(\text{Fluid}\$, T=T_evap, x=1)$ "Enthalpy at state 7"

$s7= \text{entropy}(\text{Fluid}\$, T=T_evap, x=1)$ "Entropy at state 7"

$Q_e = m_vcr * (h7 - h6)$ "Refrigeration effect in kW, "

"Process 7-8"

$s_7 = s_{8s}$

$h_{8s} = \text{enthalpy}(\text{Fluid}\$, s=s_{8s}, P=P_1)$

$h_8 = h_7 + ((h_{8s} - h_7)/\eta_c)$ "Enthalpy at state 8, "

$T_8 = \text{temperature}(\text{Fluid}\$, P=P_1, h=h_8)$

$m_{vcr} = 1.315$ "Mass flow rate in VCR cycle"

$W_c = (m_{vcr} * (h_8 - h_7))$ "Compressor work in kW"

$W_{net_vcr} = W_c$ "Net compressor work in VCR cycle"

"Process 8-1"

$Q_{c_vcr} = m_{vcr} * (h_8 - h_1)$ "Heat rejected to condenser"

$COP_{vcr} = Q_e / W_{net_vcr}$ "Coefficient of Performance of VCR "

$COP_{overall} = \eta_{orc} * COP_{vcr}$ "Overall system COP"

$MKW = (m_{orc} + m_{vcr}) / Q_e$ "Mass flow rate per kW of cooling"

$W_{net} = W_{net_orc} - W_{net_vcr}$

$v_4 = \text{volume}(\text{Fluid}\$, P=P_{boil}, x=1)$

$v_5 = \text{volume}(\text{Fluid}\$, P=P_1, h=h_5)$

$EPR = v_5 / v_4$ "Expansion ratio across the expander"

$p_7 = \text{pressure}(\text{Fluid}\$, T=T_{evap}, x=1)$ "P_evap"

$p_8 = p_1$

$CMR = p_8 / p_7$ "Compression ratio across the compressor"

Similar code will be written for other working fluids by defining the fluid in the same manner as R600 and by defining the critical temperature and pressure for the working fluid in the "Define Fluid Properties for Selected Refrigerant" section of the code that was previously mentioned.

5.5: OUTPUT

The results obtained are as follows:

For R600 (Butane):

The data obtained on solution window of EES for R600 is given in Table 2 below:

Table 2: Solution Window obtained on EES for R600

CMR = 3.862	COP _{overall} = 0.753	COP _{vcr} = 4.242	cp _g = 1.005 [kJ/kg-°C]	cp _w = 4.18 [kJ/kg-°C]
EPR = 17.08	η _c = 0.75	η _e = 0.8	η _{orc} = 0.1775	η _p = 0.75
h ₁ = 284 [kJ/kg]	h ₂ = 291.4 [kJ/kg]	h _{2s} = 289.5 [kJ/kg]	h ₃ = 385.1 [kJ/kg]	h ₄ = 742 [kJ/kg]
h ₅ = 670.2 [kJ/kg]	h _{5s} = 652.2 [kJ/kg]	h ₆ = 284 [kJ/kg]	h ₇ = 577.4 [kJ/kg]	h ₈ = 646.6 [kJ/kg]
h _{8s} = 629.3 [kJ/kg]	MKW = 0.006815 [kg/kJ]	m _{cw} = 2.946 [kg/s]	m _g = 4.669 [kg/s]	m _{orc} = 1.314 [kg/s]
m _{vcr} = 1.315 [kg/s]	p ₁ = 329 [kPa]	p ₂ = 3441 [kPa]	p ₇ = 85.18 [kPa]	p ₈ = 329 [kPa]
P _{boil} = 3441 [kPa]	P _{crit} = 3796 [kPa]	Q _{cw} = 123.1 [kW]	Q _{c,orc} = 507.6 [kW]	Q _{c,vcr} = 476.8 [kW]
Q _e = 385.9 [kW]	Q _{eg} = 469.2 [kW]	s ₁ = 1.288 [kJ/kg-K]	s ₄ = 2.482 [kJ/kg-K]	s _{5s} = 2.482 [kJ/kg-K]
s ₇ = 2.408 [kJ/kg-K]	s _{8s} = 2.408 [kJ/kg-K]	T ₂ = 37.26 [°C]	T ₃ = 73.39 [°C]	T ₄ = 146 [°C]
T ₅ = 53.82 [°C]	T ₈ = 41.48 [°C]	T _{cond} = 35 [°C]	T _{crit} = 152 [°C]	T _{cw,in} = 90 [°C]
T _{cw,out} = 80 [°C]	T _{evap} = -5 [°C]	T _{g1} = 220 [°C]	T _{g2} = 120 [°C]	v ₄ = 0.007699 [m ³ /kg]
v ₅ = 0.1315 [m ³ /kg]	W _c = 90.96 [kJ/s]	W _{net,orc} = 105.2 [kJ/s]	W _{net,vcr} = 90.96 [kJ/s]	W _p = 12.92 [kJ/s]
W _t = 118.1 [kJ/s]				

Variation of overall COP and mass flow rate per kilowatt cooling with boiler pressure [P_{boil}], condenser temperature [T_{cond}] and evaporator temperature [T_{evap}] for R600 (Butane) is shown

in Table 3 below:

Table 3: Parametric Table for R600

P_{boil} (kPa)	COP overall	MKW (kg/kJ)	T_{cond} (°C)	COP overall	MKW (kg/kJ)	T_{evap} (°C)	COP overall	MKW (kg/kJ)
2000	0.6573	0.006835	30	0.9252	0.006451	-15	0.5545	0.007159
2167	0.6763	0.006809	32.78	0.8238	0.006648	-12.78	0.5913	0.00708
2333	0.6928	0.006787	35.56	0.7366	0.006858	-10.56	0.6318	0.007002
2500	0.7072	0.006772	38.33	0.6609	0.00708	-8.33	0.6765	0.006926
2667	0.7197	0.006761	41.11	0.5946	0.007317	-6.11	0.7261	0.006851
2833	0.7303	0.006757	43.89	0.5361	0.007571	-3.889	0.7815	0.006778
3000	0.7392	0.006759	46.67	0.4842	0.007842	-1.667	0.8437	0.006707
3167	0.7463	0.006769	49.44	0.4379	0.008132	0.5556	0.914	0.006637
3333	0.7513	0.006791	52.22	0.3964	0.008444	2.7778	0.9942	0.006568
3500	0.7533	0.006832	55	0.359	0.008781	5	1.086	0.0065

For R600a (Isobutane):

The data obtained on solution window of EES for R600a is given in Table 4 below:

Table 4: Solution Window obtained on EES for R600a

CMR = 3.555	COP _{overall} = 0.6488	COP _{ver} = 4.16	cp _g = 1.005 [kJ/kg-°C]	cp _w = 4.18 [kJ/kg-°C]
EPR = 11.66	η _c = 0.75	η _e = 0.8	η _{orc} = 0.156	η _p = 0.75
h ₁ = 284.3 [kJ/kg]	h ₂ = 291.4 [kJ/kg]	h _{2s} = 289.6 [kJ/kg]	h ₃ = 386.2 [kJ/kg]	h ₄ = 680.1 [kJ/kg]
h ₅ = 624.1 [kJ/kg]	h _{5s} = 610.1 [kJ/kg]	h ₆ = 284.3 [kJ/kg]	h ₇ = 548.3 [kJ/kg]	h ₈ = 611.8 [kJ/kg]
h _{8s} = 595.9 [kJ/kg]	MKW = 0.008283 [kg/kJ]	m _{cw} = 3.619 [kg/s]	m _g = 4.669 [kg/s]	m _{orc} = 1.596 [kg/s]
m _{ver} = 1.345 [kg/s]	p ₁ = 464.4 [kPa]	p ₂ = 3329 [kPa]	p ₇ = 130.7 [kPa]	p ₈ = 464.5 [kPa]
P _{boil} = 3329 [kPa]	P _{crit} = 3640 [kPa]	Q _{cw} = 151.3 [kW]	Q _{c,orc} = 542.3 [kW]	Q _{c,ver} = 440.5 [kW]

$Q_e = 355.1$ [kW]	$Q_{eg} = 469.2$ [kW]	$s_1 = 1.288$ [kJ/kg-K]	$s_4 = 2.345$ [kJ/kg-K]	$s_{5s} = 2.345$ [kJ/kg-K]
$s_7 = 2.3$ [kJ/kg-K]	$s_{8s} = 2.3$ [kJ/kg-K]	$T_2 = 37.32$ [°C]	$T_3 = 73.48$ [°C]	$T_4 = 129.4$ [°C]
$T_5 = 46.67$ [°C]	$T_8 = 40.24$ [°C]	$T_{cond} = 35$ [°C]	$T_{crit} = 134.7$ [°C]	$T_{cw,in} = 90$ [°C]
$T_{cw,out} = 80$ [°C]	$T_{evap} = -5$ [°C]	$T_{g1} = 220$ [°C]	$T_{g2} = 120$ [°C]	$v_4 = 0.007562$ [m ³ /kg]
$v_5 = 0.08819$ [m ³ /kg]	$W_c = 85.37$ [kJ/s]	$W_{net,orc} = 96.48$ [kJ/s]	$W_{net,vcr} = 85.37$ [kJ/s]	$W_p = 15.07$ [kJ/s]
$W_t = 111.9$ [kJ/s]				

Variation of overall COP and mass flow rate per kilowatt cooling with boiler pressure [P_{boil}], condenser temperature [T_{cond}] and evaporator temperature [T_{evap}] for R600a (Isobutane) is shown in Table 5 below:

Table 5: Parametric Table for R600a

P_{boil} (kPa)	COP overall	MKW (kg/kJ)	T_{cond} (°C)	COP overall	MKW (kg/kJ)	T_{evap} (°C)	COP overall	MKW (kg/kJ)
2000	0.008269	0.5547	30	0.8062	0.007775	-15	0.4758	0.008724
2167	0.008237	0.5749	32.78	0.7134	0.008049	-12.78	0.5079	0.008622
2333	0.008212	0.5923	35.56	0.6337	0.008344	-10.56	0.5431	0.008523
2500	0.008196	0.6073	38.33	0.5646	0.008659	-8.33	0.5821	0.008425
2667	0.008188	0.6201	41.11	0.5042	0.008999	-6.11	0.6253	0.00833
2833	0.00819	0.6308	43.89	0.451	0.009365	-3.889	0.6736	0.008237
3000	0.008203	0.6394	46.67	0.4038	0.009761	-1.667	0.7278	0.008146
3167	0.008232	0.6456	49.44	0.3618	0.01019	0.5556	0.7891	0.008056
3333	0.008285	0.6488	52.22	0.3242	0.01066	2.7778	0.859	0.007969
3500	0.008388	0.6464	55	0.2905	0.01116	5	0.9394	0.007883

For Solstice ZE (R1234ze):

The data obtained on solution window of EES for R1234ze is given in Table 6 below:

Table 6: Solution Window obtained on EES for R1234ze

CMR = 3.69	COP _{overall} = 0.4044	COP _{vcr} = 4.075	cp _g = 1.005 [kJ/kg-°C]	cp _w = 4.18 [kJ/kg-°C]
EPR = 7.763	η _c = 0.75	η _e = 0.8	η _{orc} = 0.09924	η _p = 0.75
h ₁ = 248.5 [kJ/kg]	h ₂ = 251.6 [kJ/kg]	h _{2s} = 250.8 [kJ/kg]	h ₃ = 286.7 [kJ/kg]	h ₄ = 420.6 [kJ/kg]
h ₅ = 400.7 [kJ/kg]	h _{5s} = 395.7 [kJ/kg]	h ₆ = 284.5 [kJ/kg]	h ₇ = 380.7 [kJ/kg]	h ₈ = 413.1 [kJ/kg]
h _{8s} = 4.5 [kJ/kg]	MKW = 0.02772 [kg/kJ]	m _{cw} = 2.946 [kg/s]	m _g = 4.669 [kg/s]	m _{orc} = 3.505 [kg/s]
m _{vcr} = 1.315 [kg/s]	p ₁ = 668.5 [kPa]	p ₂ = 3314 [kPa]	p ₇ = 181.2 [kPa]	p ₈ = 668.5 [kPa]
P _{boil} = 3314 [kPa]	P _{crit} = 3632 [kPa]	Q _{cw} = 123.1 [kW]	Q _{c,orc} = 533.6 [kW]	Q _{c,vcr} = 216.5 [kW]
Q _e = 173.9 [kW]	Q _{eg} = 469.2 [kW]	s ₁ = 1.166 [kJ/kg-K]	s ₄ = 1.643 [kJ/kg-K]	s _{5s} = 1.643 [kJ/kg-K]
s ₇ = 1.673 [kJ/kg-K]	s _{8s} = 1.673 [kJ/kg-K]	T ₂ = 37.11 [°C]	T ₃ = 61.17 [°C]	T ₄ = 104.6 [°C]
T ₅ = 35 [°C]	T ₈ = 40.47 [°C]	T _{cond} = 35 [°C]	T _{crit} = 109.4 [°C]	T _{cw,in} = 90 [°C]
T _{cw,out} = 80 [°C]	T _{evap} = -5 [°C]	T _{g1} = 220 [°C]	T _{g2} = 120 [°C]	v ₄ = 0.003499 [m ³ /kg]
v ₅ = 0.02678 [m ³ /kg]	W _c = 42.66 [kJ/s]	W _{net,orc} = 58.79 [kJ/s]	W _{net,vcr} = 42.66 [kJ/s]	W _p = 10.9 [kJ/s]
W _t = 69.69 [kJ/s]				

Variation of overall COP and mass flow rate per kilowatt cooling with boiler pressure [P_{boil}], condenser temperature [T_{cond}] and evaporator temperature [T_{evap}] for Solstice ZE (R1234ze) is shown in Table 7 below:

Table 7: Parametric Table for R1234ze

P_{boil} (kPa)	COP overall	MKW (kg/kJ)	T_{cond} (°C)	COP overall	MKW (kg/kJ)	T_{evap} (°C)	COP overall	MKW (kg/kJ)
2000	0.3282	0.02677	30	0.5129	0.02553	-15	0.2949	0.02928
2167	0.3452	0.02673	32.78	0.4489	0.02671	-12.78	0.3152	0.02892
2333	0.3599	0.02673	35.56	0.394	0.02799	-10.56	0.3375	0.02857
2500	0.3723	0.02676	38.33	0.3466	0.02938	-8.33	0.3621	0.02822
2667	0.3829	0.02681	41.11	0.3052	0.03089	-6.11	0.3895	0.02789
2833	0.3917	0.02690	43.89	0.2689	0.03256	-3.889	0.4201	0.02756
3000	0.3986	0.02705	46.67	0.2368	0.03438	-1.667	0.4544	0.02724
3167	0.4033	0.02730	49.44	0.2084	0.0364	0.5556	0.4932	0.02694
3333	0.4042	0.02781	52.22	0.1831	0.03863	2.7778	0.5374	0.02664
3500	0.3953	0.02910	55	0.1605	0.04112	5	0.5883	0.02634

For Solstice YF (R1234yf):

The data obtained on solution window of EES for R1234yf is given in Table 8 below:

Table 8: Solution Window obtained on EES for R1234yf

CMR = 3.37	COP _{overall} = 0.3192	COP _{ver} = 3.891	cp _g = 1.005 [kJ/kg-°C]	cp _w = 4.18 [kJ/kg-°C]
EPR = 5.015	η _c = 0.75	η _e = 0.8	η _{orc} = 0.08203	η _p = 0.75
h ₁ = 247.6 [kJ/kg]	h ₂ = 250.3 [kJ/kg]	h _{2s} = 249.7 [kJ/kg]	h ₃ = 280.3 [kJ/kg]	h ₄ = 394.2 [kJ/kg]
h ₅ = 379.7 [kJ/kg]	h _{5s} = 376.1 [kJ/kg]	h ₆ = 247.6 [kJ/kg]	h ₇ = 360 [kJ/kg]	h ₈ = 388.8 [kJ/kg]
h _{8s} = 381.6 [kJ/kg]	MKW = 0.03678 [kg/kJ]	m _{cw} = 2.946 [kg/s]	m _g = 4.669 [kg/s]	m _{orc} = 4.118 [kg/s]
m _{ver} = 1.315 [kg/s]	p ₁ = 895.2 [kPa]	p ₂ = 3048 [kPa]	p ₇ = 265.8 [kPa]	p ₈ = 895.2 [kPa]
P _{boil} = 3048 [kPa]	P _{crit} = 3382 [kPa]	Q _{cw} = 123.1 [kW]	Q _{c,orc} = 543.8 [kW]	Q _{c,ver} = 185.7 [kW]

$Q_e = 147.7$ [kW]	$Q_{eg} = 469.2$ [kW]	$s_1 = 1.162$ [kJ/kg-K]	$s_4 = 1.579$ [kJ/kg-K]	$s_{5s} = 1.579$ [kJ/kg-K]
$s_7 = 1.597$ [kJ/kg-K]	$s_{8s} = 1.597$ [kJ/kg-K]	$T_2 = 37.12$ [°C]	$T_3 = 57.46$ [°C]	$T_4 = 89.47$ [°C]
$T_5 = 35$ [°C]	$T_8 = 38.87$ [°C]	$T_{cond} = 35$ [°C]	$T_{crit} = 94.7$ [°C]	$T_{cw,in} = 90$ [°C]
$T_{cw,out} = 80$ [°C]	$T_{evap} = -5$ [°C]	$T_{g1} = 220$ [°C]	$T_{g2} = 120$ [°C]	$v_4 = 0.003836$ [m ³ /kg]
$v_5 = 0.01924$ [m ³ /kg]	$W_c = 37.96$ [kJ/s]	$W_{net,orc} = 48.59$ [kJ/s]	$W_{net,vcr} = 37.96$ [kJ/s]	$W_p = 11.15$ [kJ/s]
$W_t = 59.74$ [kJ/s]				

Variation of overall COP and mass flow rate per kilowatt cooling with boiler pressure [P_{boil}], condenser temperature [T_{cond}] and evaporator temperature [T_{evap}] for Solstice YF (R1234yf) is shown in Table 9 below:

Table 9: Parametric Table for R1234yf

P_{boil} (kPa)	COP overall	MKW (kg/kJ)	T_{cond} (°C)	COP overall	MKW (kg/kJ)	T_{evap} (°C)	COP overall	MKW (kg/kJ)
2500	0.2932	0.03578	30	0.4152	0.03335	-15	0.2311	0.03907
2578	0.2984	0.03584	32.78	0.3585	0.03518	-12.78	0.2474	0.03853
2656	0.3031	0.03592	35.56	0.3101	0.0372	-10.56	0.2654	0.03801
2733	0.3074	0.03602	38.33	0.2683	0.03943	-8.33	0.2852	0.03751
2811	0.3112	0.03615	41.11	0.232	0.04191	-6.11	0.3072	0.03702
2889	0.3144	0.03631	43.89	0.2003	0.04469	-3.889	0.3318	0.03654
2967	0.3171	0.03651	46.67	0.1725	0.04781	-1.667	0.3595	0.03608
3044	0.3191	0.03676	49.44	0.148	0.05134	0.5556	0.3907	0.03563
3122	0.3202	0.03711	52.22	0.1264	0.05536	2.7778	0.4264	0.03519
3200	0.3202	0.03759	55	0.1072	0.05998	5	0.4674	0.03477

The combined plot of overall COP vs Boiler Pressure for all working fluids is given below:

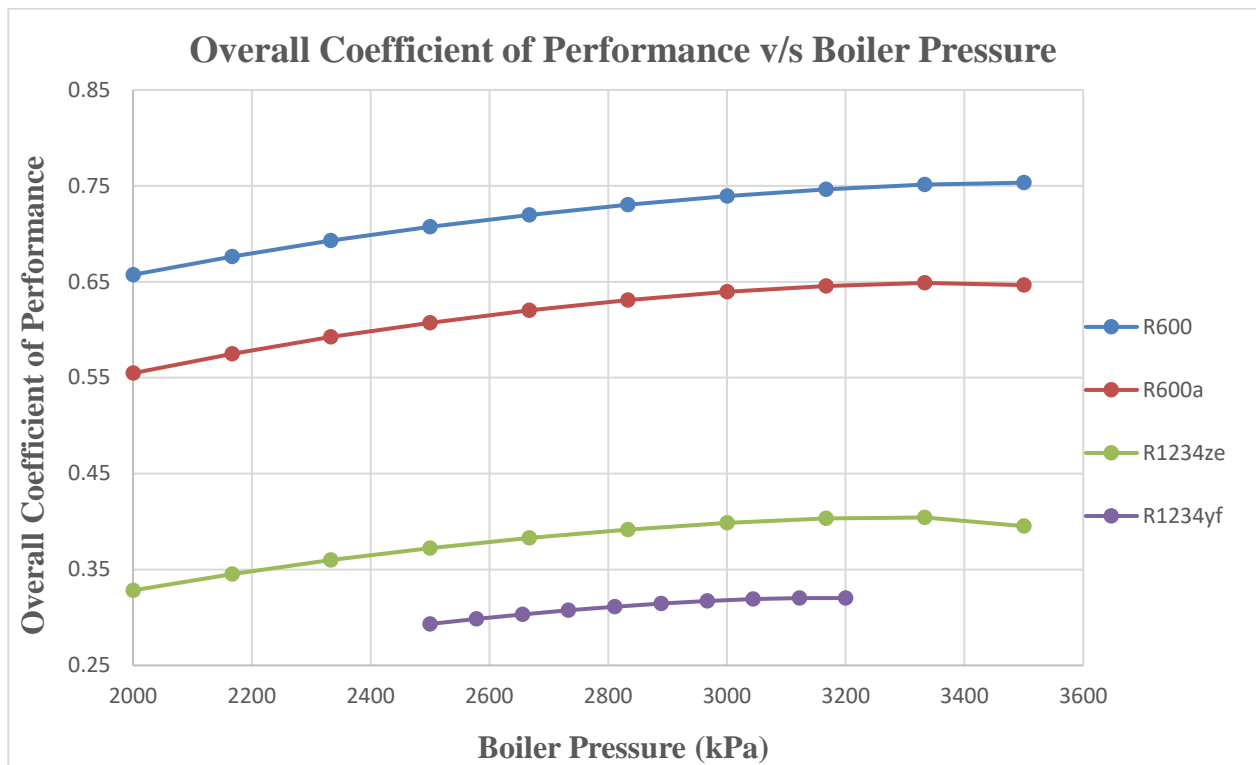


Fig 3: Combined Plot of overall COP vs Boiler Pressure

The combined plot of mass flow rate per kilowatt cooling (MKW) vs Boiler Pressure for all working fluids is given below:

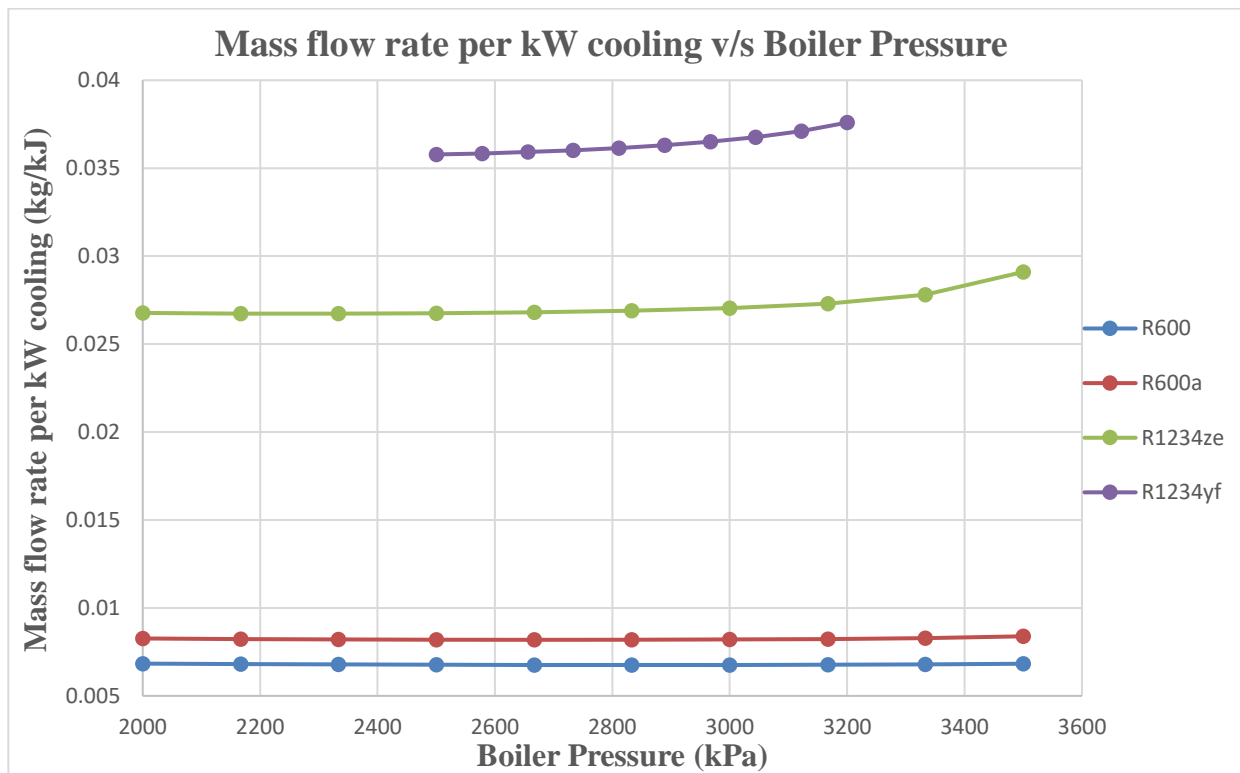


Fig 4: Combined Plot of MKW vs Boiler Pressure

The combined plot of overall COP vs Condenser Temperature for all working fluids is given below:

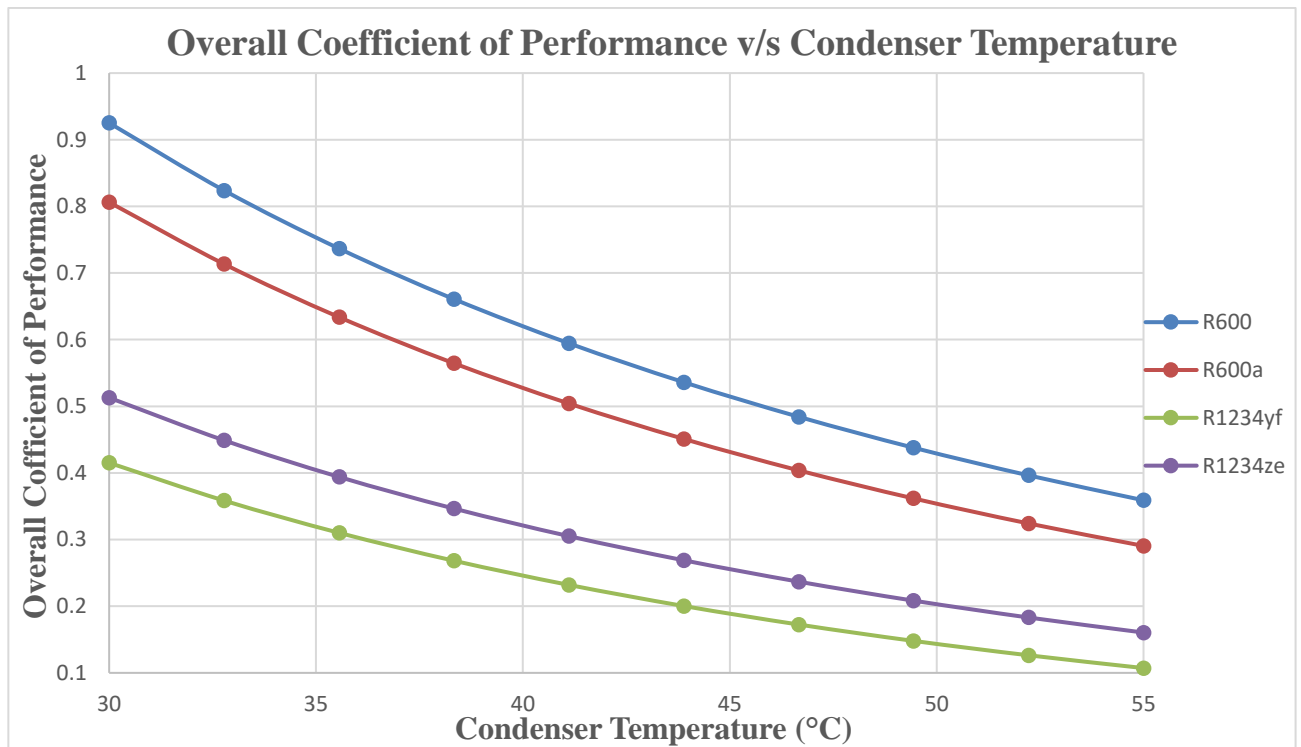


Fig 5: Combined Plot of overall COP vs Condenser Temperature

The combined plot of mass flow rate per kilowatt cooling (MKW) vs Condenser Temperature for all working fluids is given below:

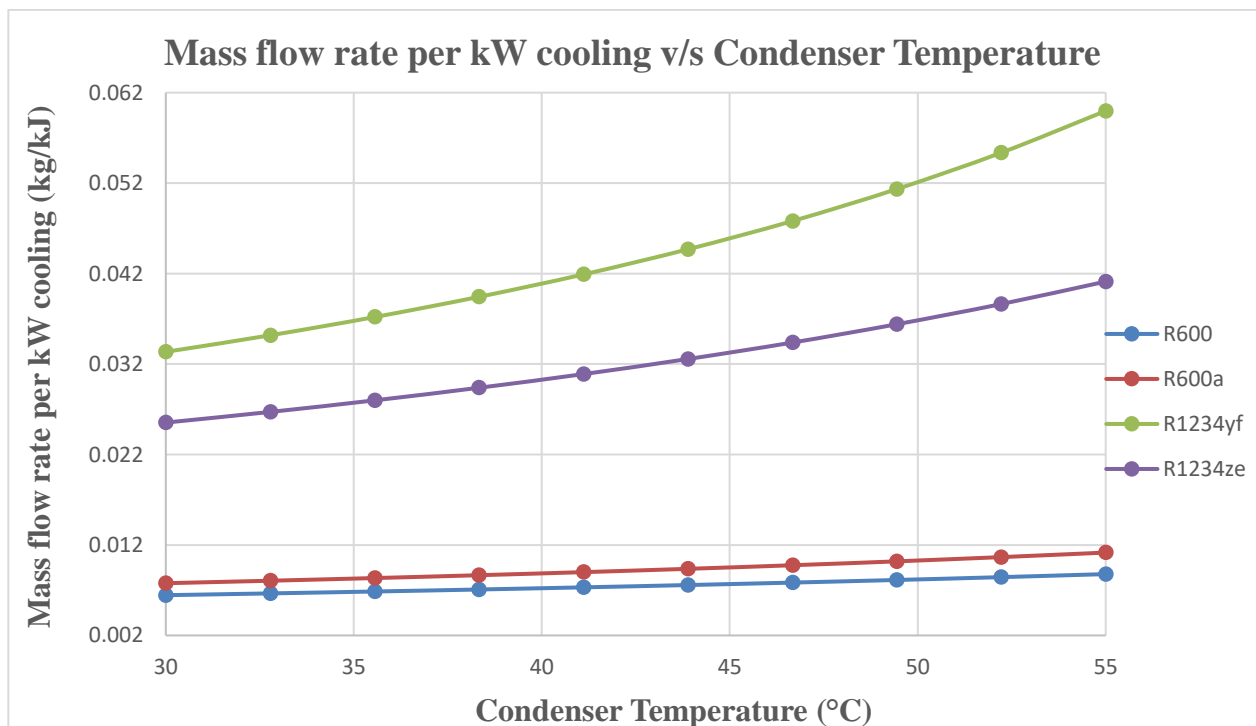


Fig 6: Combined Plot of MKW vs Condenser Temperature

The combined plot of overall COP vs Evaporator Temperature for all working fluids is given below:

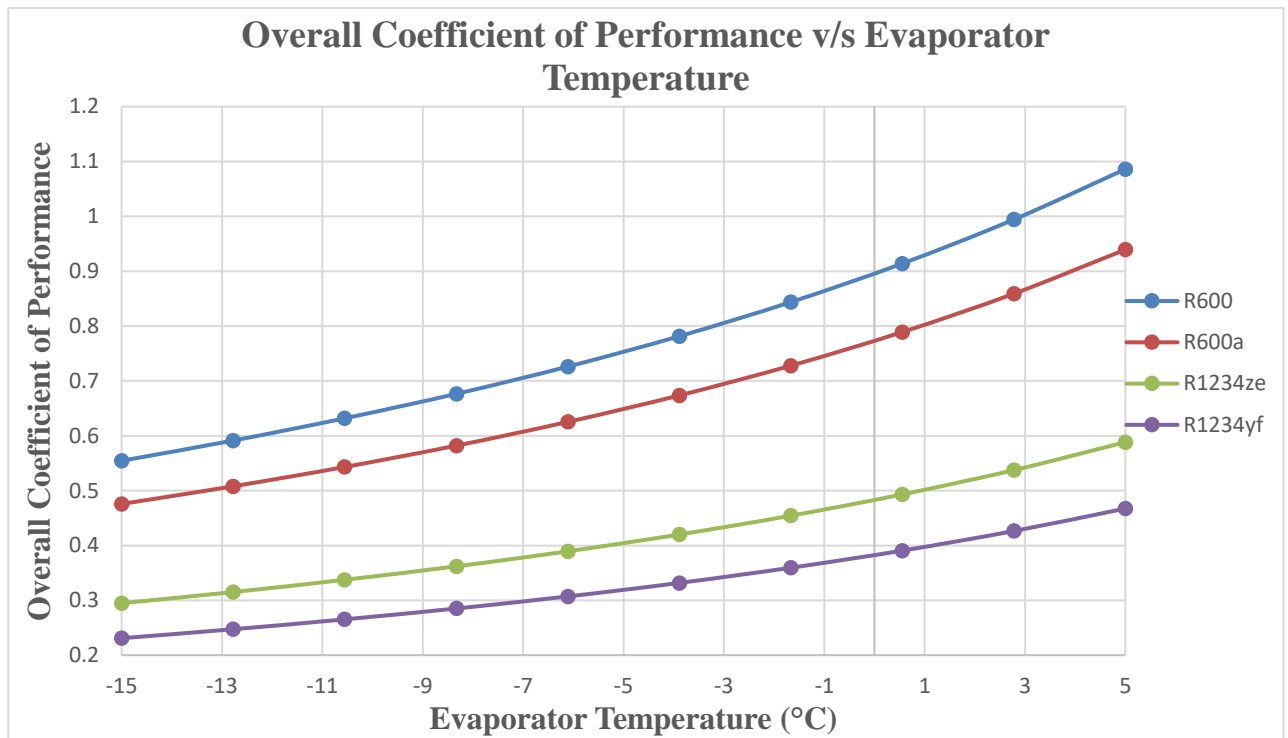


Fig 7: Combined Plot of overall COP vs Evaporator Temperature

The combined plot of mass flow rate per kilowatt cooling (MKW) vs Evaporator Temperature for all working fluids is given below:

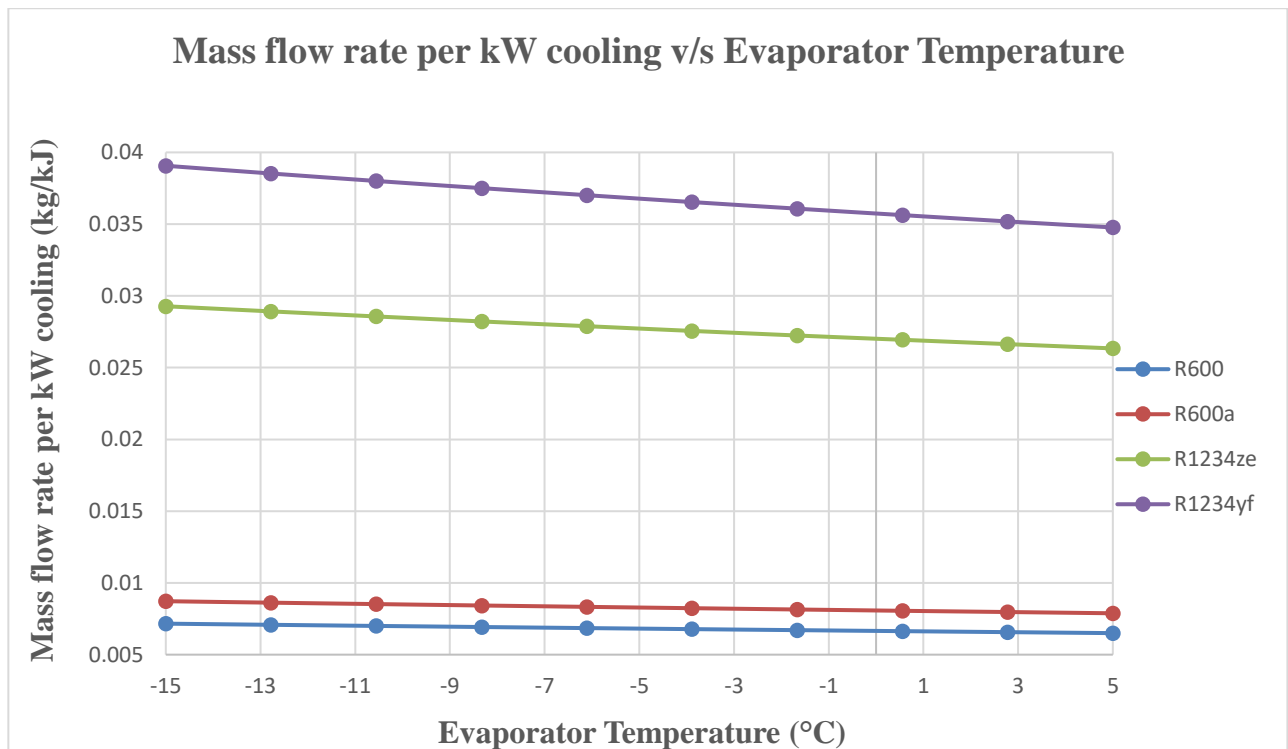


Fig 8: Combined Plot of MKW vs Evaporator Temperature

CHAPTER-6: CONCLUSION

This study explored the thermodynamic performance of a Combined Organic Rankine Cycle (ORC) and Vapor Compression Refrigeration (VCR) cycle, specifically designed to harness waste heat from diesel engines. The analysis focused on four hydrocarbon working fluids: R600 (Butane), R600a (Isobutane), R1234ze (Solstice ZE) and R1234yf (Solstice YF).

The results revealed that as the boiler pressure increased, the overall coefficient of performance (overall COP) also increased, reaching a peak before the critical pressure point. Conversely, the mass flow rate per kW of cooling capacity (MKW) showed a constant value for most of the values before slight rise and reaching a peak with rising boiler pressure before reaching critical pressure point. Additionally, the study found that the overall COP decreased with increasing condenser temperature and decreasing evaporator temperature, while MKW exhibited opposite behavior i.e. increase in MKW was observed with increase in condenser temperature but decrease in MKW was observed with increase evaporator temperature but for evaporator temperature, the variation of R600 and R600a with evaporator temperature is negligible compared to R1234ze and R1234yf.

Among the four working fluids, R600 (Butane) outperforms other refrigerants by demonstrating the highest thermodynamic performance, with the maximum overall COP indicating more cooling output for a given amount of input energy, making it highly efficient and lowest MKW showing efficient energy transfer and potentially lower operational costs, making it the most efficient choice for combined cycle systems. R600a, though offering slightly reduced values in both metrics, remains a suitable alternative. In contrast, R1234ze and R1234yf, with significantly lower overall COP and high mass flow rates, are less suitable for this application. Based on the graph, R600, by demonstrating superior thermodynamic performance, emerges as the optimal working fluid for combined cycle systems.

Though R600 (Butane) demonstrates the highest thermodynamic performance, practical considerations led to the selection of R600a (Isobutane) as the most suitable working fluid. Its thermodynamic performance is only second to R600 but is preferred due to its wider availability, lower cost and reduced flammability when compared to R600. Additionally, its compatibility with standard system materials and short atmospheric lifetime (that minimizes its long-term environmental impact) is also considered, making it more practical for real-world applications.

The applications of this technology are diverse. It can be used to recover waste heat from diesel engines for power generation and cooling and transportation (heavy-duty vehicles, marine vessels), industrial facilities with diesel generators, remote installations, and data centers, where the system can provide both cooling and supplementary power from waste heat.

Future research directions include integrating smart control systems, developing advanced heat exchanger designs, and exploring compact, modular units. Furthermore, combining this system with renewable energy sources could lead to hybrid solutions with enhanced energy efficiency and environmental benefits. It also shows that though thermodynamic performance is essential, practical considerations such as system pressure, safety factors, and component sizing must be carefully balanced when selecting working fluids. Further research, such as exergo-economic analysis, can provide valuable insights into the practical implementation of these systems.

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