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Exergy calculations and refrigerant selection using flue gas waste heat in organic rankine cycle

Organik rankine çevriminde baca gazı atık ısısının kullanılmasıyla ekserji hesaplamaları ve soğutucu seçimi

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Exergy Calculations and Refrigerant Selection Using Flue Gas Waste Heat in Organic Rankine Cycle

Highlights

- *According to the results of EES calculations, exergy efficiency, thermal efficiency and power generation of R32 are 0,284, 0,3049, 170,3 kJ/kg, respectively, and R32 is the most suitable refrigerant among the candidate refrigerants in terms of exergy efficiency, energy efficiency and power generation.*
- *By comparing the energy efficiency with exergy efficiency and examining the entropy production values for each of candidate refrigerants, it has been proven that the ORC design is correct and logical.*

Graphical Abstract

Calculations have been made in EES for each of the candidate refrigerants. The most suitable refrigerant selection criterion has been primarily exergy efficiency, while criterias such as energy efficiency and power generation have also been taken into consideration, during the comparison.

Table. Results Calculated in EES for Refrigerants Comparison

Criterions	**R32**	R125	R143A	R290	R600A
$\eta_{IIcycle}$	**0,284**	0,02013	0,1044	0,1656	0,01589
$\eta_{Thermal}$	**0,3049**	0,2461	0,2646	0,2772	0,2441
$w_{turbine}$	**170,3**	55,77	84,44	169,2	123,9
q_{in}	**549,4**	217,5	308,1	589,2	484,1
$x_{destructed,heatexchanger}$	**105,9**	55,47	70,01	111,8	58,25
$x_{destructed,condenser}$	**3,443**	4,888	7,891	27,16	77,36

Aim

The study aims the desinging basic subcritical ORC and determining the candidate refrigerant, Moreover, It is also aimed to prove that the designed ORC is logical and to select the most suitable refrigerant by comparing candidate refrigerants.

Design &Methodology

Basic subcritical ORC has been designed, candidate refrigerants have been selected according to the boiling temperature point data obtained from miniREFPROP, and each of selected candidate refrigerants have been calculated in the EES program, with the aim of comparing them with each other.

Originality

Generally, the temperature of the sink is selected as the ambient temperature. In order to perform ORC calculations, the temperature of sink was chosen as a temperature lower than the boiling temperature point of the refrigerant with the lowest boiling temperature point among the candidate refrigerants, instead of the ambient temperature.

Findings

R32, which has an exergy efficiency of 0,284, has been found to be the most suitable refrigerant among the candidate refrigerants because it has the highest exergy efficiency . R32 is also the most suitable among the candidate refrigerants in terms of work produced per mass and energy efficiency.

Conclusion

R32 is a usable and suitable refrigerant for ORC designed for the use of low temperature heat sources such as flue gas waste heat.

Declaration of Ethical Standarts

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Exergy Calculations and Refrigerant Selection Using Flue Gas Waste Heat in Organic Rankine Cycle

Research Article / Araştırma Makalesi

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ABSTRACT

In this study, the second law efficiency of the organic Rankine cycle (ORC) designed was calculated for each of refrigerants, and the exergies of the ORC components were examined separately. Candidate refrigerants were determined for comparison. When determining candidate refrigerants, the first thing to consider was the low boiling point temperature. Five candidate refrigerants were determined for comparison purposes, namely R290, R600a, R32, R125, R143a. When determining candidate refrigerants, importance was given to the criterion that their boiling temperature should be lower than the boiling temperature of water. The refrigerant with the highest exergy efficiency was selected as the most suitable refrigerant among the candidate refrigerants. The temperature of sink has been taken as a value that lower than the boiling point temperature o the refrigerant that has the lowest boiling point temperature among the others instead of ambient temperature.The power generation, first law efficiency and second law efficiency of R32were found to be 170,3 kJ/kg, 0,3049 and 0,284, respectively.The thermal efficiency and second law efficiency values of the ORC using R32 are higher than those using other refrigerants for conditions in the considered textile plant.
Keywords: Second law efficiency, ORC, exergy analysis, working refrigerant.

Organik Rankine Çevriminde Baca Gazı Atık Isısının Kullanılmasıyla Ekserji Hesaplamaları ve Soğutucu Seçimi

ÖZ

Bu çalışmada, tasarlanan organik Rankine çevriminin (ORÇ) ikinci yasa verimliliği her bir soğutucu için hesaplanmış ve ORC bileşenlerinin ekserjileri ayrı ayrı incelenmiştir. Karşılaştırma için aday soğutucular belirlenmiştir. Aday soğutucular belirlenirken, dikkate alınması gereken ilk şey düşük kaynama noktası sıcaklığıdır. Karşılaştırma amacıyla beş aday soğutucu belirlenmiştir: R290, R600a, R32, R125, R143a. Aday soğutucular belirlenirken, kaynama sıcaklıklarının suyun kaynama sıcaklığından düşük olması kriterine önem verilmiştir. Aday soğutucular arasında en yüksek ekserji verimliliğine sahip soğutucu en uygun soğutucu olarak seçilmiştir. Soğuk kaynak sıcaklığı, ortam sıcaklığı yerine diğerleri arasında en düşük kaynama noktası sıcaklığına sahip olan soğutucunun kaynama noktası sıcaklığından daha düşük bir değer olarak alınmıştır. R32'nin güç üretimi, birinci yasa verimi ve ikinci yasa verimi sırasıyla 170,3 kJ/kg, 0,3049 ve 0,284 olarak bulunmuştur. R32 kullanan ORC'nin termal verimi ve ikinci yasa verimi değerleri, ele alınan tekstil tesisindeki koşullar için diğer soğutucularınkinden daha yüksek bulunmuştur.
Anahtar Kelimeler: İkinci kanun verimi, ORÇ, ekserji analizi, çalışılan soğutucular.

1. INTRODUCTION

Due to the depletion of fossil resources day by day and their harmfulness to the environment, the aim is to use heat sources as little as possible, waste heat using have become attractive. Utilising the waste heat is one of the most important issues of energy efficiency. The ORC is a cycle similar to the Rankine Cycle, the difference from the Rankine Cycle is that the refrigerants of ORC is various types of organic refrigerants other than water. The ORC has resource, economic benefit as it is an efficiency-increasing cycle. Other indirect benefits include lower CO_2 emissions and less environmental damage due to the use of fewer heat sources. In addition to all these, many other benefits are provided indirectly such as environmental, climatic, economic.

Kaska (2014) conducted energy analysis and exergy analysis studies. The application area is on waste heat in ORC. For two different real cases, the energy and exergy efficiency varies as 10,2%; 48,5% and 8,8%; 42,2% respectively. He stated that the evaporation pressure is very significant effect on the efficiencies.

Obi (2015) This paper has carried out a comprehensive study of micro generation systems up to 100 kWe. It has been found that the most commonly used refrigerant in the ORC systems is R245fa.The study has revealed that the most suitable expander for applications of these plant size ranges is scroll expander for small plants and vane or screw expander for larger plants.

Petrollese et al. (2018) researched the optimal configuration of ORC plant using solar energy and biomass. The aim of the study is to maximize the exergy

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efficiency. Performance evaluation was conducted for various conditions. It was concluded that the best working fluids is the siloxanes (especially hexamethyldisiloxane) and linear alkanes with high molecular complexity (Pentane, Iso-Hexane, etc.).It was found that the exergy efficiency was about 56-58%, but the optimum working fluid varied according to the condenser temperature.

Meneses et al.(2019) aimed to analyze an ORC system. R245fa is used in the system. They investigated municipal waste in Terceira Island (Azores) from thermodynamic, environmental and economic perspectives. Visual Basic programming language was used for the thermodynamic model. Efficiency is 25%.The losses in the study exceed 500,000 Euros (per year) and as a result, this project was not found to be economically viable. Asim et al.(2021) evaluated the performance of a new integrated vapor compression cycle-ORC (i-VCC-ORC) system and made the working fluid selection. The cycle thermal efficiency and exergy efficiency were evaluated.R600a were found to be the best working fluid. ORC increased the coefficient of performance of the system by 12,5%.

Araya et al.(2021) emphasized that ORC is applicable for waste heat recovery to generate low-cost electricity. In the experimental results, they obtained thermal efficiencies. Isentropic efficiency of the expander was found to be 70%.

Al-Badri et al.(2022) stated that condensation temperature affects the thermal efficiency of the Organic Rankine cycle. At different condensing temperatures, thermal efficiency in ORC is calculated.R134a was used as the working fluid. Exergy efficiency is 18,26%. It has been shown that thermal efficiency and energy output in an ORC system increase with decreasing the condensation temperature.

Caglar and Bahadır (2022) have worked on the design of solar energy-assisted ORC. They used the EES program to define the ORC mathematical model. R123 and R600 were used and these two fluids were compared with each other. At the end of the study, the R600 fluid performed better than R123. It was found that the turbine efficiency parameter significantly affected the thermal efficiency.

Abdallah and El-shennawy (2023) conducted an application on a refinery located in Alexandria, Egypt (Alexandria National Refining and Petrochemicals Company). They conducted a study to evaluate the waste steam heat. The average flow rate of the recovered used steam is 15 Ton/hour, the average pressure is 4,5 bar and the average temperature is 180°C~220°C. The total cost and the financial benefits of the project were investigated. The expected electrical power to be produced is 1 750 kW. As a result of the financial analysis, it was found that the ORC system would cost approximately 4,8 million US dollars and the payback period was 4 years.

Ahmet and Imre (2023) emphasized the importance of efficiency in power cycles. They compared ORC and TFC. In their thermodynamic analysis, they used three dry working fluids with zero to partial recuperated heat ratio, subcritical base ORC and TFC. When heat recovery system was added for both systems, it was concluded that ORC could outperform TFC for Dodecane, Decane and Nonane, respectively.

Dominguez et al. (2023) analyzed the thermodynamic performance of a zero-emission solar trigeneration system using a numerical approach. Both the First and Second Laws of Thermodynamics were used for the analysis. The proposed ORC is based on a single-pressure regenerative, recovered, and superheated cycle.Theoptimal system achieves an energy efficiency of 152,4%, an exergy efficiency of 21,1%, and an electricity-exergy efficiency of 17,5%.The electricity, cooling, and heating productions are 82,1 kW, 200,4 kW, and 471,7 kW, respectively.SPTCs were identified as the main source of exergy destruction, responsible for the destruction of 73% of the input exergy.

Kılıç and Arcaklıoglu (2023) have performed energy and exergy analyses using real data on a combined cycle power plant using natural gas. They stated that fossil fuels were used efficiently in plants consisting of Brayton and Rankine cycles. At the end of the study, exergy efficiency was found to be 61.2%. In addition, the total power obtained was 205 MW, the power of the Brayton cycle was 134 MW, and the power of the Rankine cycle was 71 MW. The effects of increasing the parameters of the ambient temperature and the gas turbine inlet temperature on the system are also among the subjects of investigation.

Siddiqui et al.(2023) conducted a study considering factors such as thermodynamic performance of the cycle, plant size and compatibility with turbomachinery. Twelve different organic fluids were used, considering their suitability for exergy efficiency parameter. To ensure wide applicability, they considered source temperatures ranging from 150 to 300 C, including industrial waste heat, geothermal sources and solar energy sources.For each case, the aim was to select the appropriate organic fluid for the given source temperatures by calculating the specific net power output and UA value (heat exchanger conductivity). Low exergy efficiency was obtained from cyclohexane, benzene, isopropyl alcohol and hexafluorobenzene fluids, because of their high boiling points.

Hakim and etc. (2024) conducted an analysis to compare flash steam and binary cycle geothermal process technologies in Lahendong region in Indonesia. They studied the simulation of double flash steam and binary cycle with regeneration. As a result of the simulation studies, binary cycle technology and binary cycle with regeneration system were found suitable for geothermal power plant.

This study has different aspects from other studies. ORC applications where flue gas waste heat is used are frequently encountered. In order to select optimum refrigerant fairly and correctly among the candidate refrigerants, the same boiler pressure and the same condenser pressure were used. When determining the heat exchanger pressure, the candidate refrigerant with the lowest critical pressure among the candidate refrigerants was taken as basis. In order to provide the subcritical basic ORC condition for each of refrigerants, the heat exchanger pressure value which is lower than the pressure of the refrigerant with the lowest critical pressure, but very close to this value for providing high boiler presure condition which is necessary for increasing thermal efficieny methods was chosen. In the articles reviewed so far, the temperature of sink for the condenser has mostly been determined as the ambient temperature value, however, it is more suitable for the boiling temperature of the working refrigerant to be higher than the sink temperature that will receive the heat from the condenser to be able to perform heat transfer and condensation process, therefore it would be more logical to determine the temperature value of sink as lower than the boiling temperature point of the refrigerant. In order to meet the condition that the sink temperature can be lower than the boiling point temperature of each of the candidate refrigerants, the boiling point value of the candidate refrigerant with the lowest boiling point temperature among the candidate refrigerants was taken as basis, and a value lower than the boiling point of this refrigerantwas assigned as the temperature value of sink. Another subject content that has not been given much space in the articles reviewed so far, is ORC's design proof, therefore ORC design was evaluated holistically from every perspective and each calculated value was evaluated. In order to prove whether the designed ORC is correct or not, and the logical evaluation of the ORC design, the exergy destruction values of the ORC components (heat exchanger and condenser etc.) was made by paying attention to their being positive. The existing economizer system of a textile company was examined. In this company, an economizer is used to recover waste heat. For this study, instead of economizer, an ORC design suitable for flue gas waste heat of 130 °C temperature was made. Mass and mass flow rate were not specified in the calculations, all calculations are per mass, because the company must decide the mass amount according to the required work output and cost constraints. Determination of the optimumrefrigerant according to criterions such as turbine work output, first law efficiency and second law efficiency is also one of the subjects of the study.

2.MATERIALS AND METHODS

2.1. Theoretical Study of ORC Application

All refrigerants were calculated in the ORC under the same conditions. The ORC design was determined according to thermal efficiency increasing methods, heat exchanger pressure was assigned as high but subcritical, condenser pressure was low and atmospheric pressure was assigned.

In the articles reviewed so far, the temperature of sink for the condenser has mostly been determined as the ambient temperature value, however, it is more suitable for the boiling temperature of the working refrigerant to be higher than the sink temperature that will receive the heat from the condenser to be able to perform heat transfer and condensation process, therefore it would be more logical to determine the temperature value of sink as lower than the boiling temperature point of the refrigerant. In order to meet the condition that the sink temperature can be lower than the boiling point temperature of each of the candidate refrigerants, the boiling point value of the candidate refrigerant with the lowest boiling point temperature among the candidate refrigerants was taken as basis, and a value lower than the boiling point of this refrigerant was assigned as the temperature value of sink. Another subject content that has not been given much space in the articles reviewed so far, is ORC's design proof, therefore ORC design was evaluated holistically from every perspective and each calculated value was evaluated. In order to prove whether the designed ORC is correct or not, and the logical evaluation of the ORC design, the exergy destruction values of the ORC components (heat exchanger and condenser etc.) was made by paying attention to their being positive.

The ORC designed is a subcritical cycle, but according to efficiency improvement methods, the heat exchanger pressure needs to be high, so a pressure value lower than the critical pressure but still high pressure value was selected. In order for the designed ORC to be a subcritical cycle, the heat exchanger pressure must be lower than the critical pressure of refrigerant. Since five candidate refrigerants were determined, the critical pressure of each of candidate refrigerants was examined and the refrigerant with the lowest critical pressure was taken as the basis.In order that all refrigerants provide the subcritical ORC condition, the heat exchanger pressure was determined to be lower than the pressure of the refrigerant (R125) with the lowest critical pressure, but it was determined to be very close to the critical pressure of this refrigerant, because higher boiler pressure will increase thermal efficiency.

In the Table 1, the properties (critical pressure etc.) obtained from the Refprop program for each candidate refrigerant are shown and It is also possible to see the comparison of the critical pressure required that the heat exchanger pressure was determined. In order to determine the Heat Exchanger pressure, the refrigerant with the lowest critical pressure was first determined. The value lower than the critical pressure of the refrigerant with the lowest critical pressure, but close to this critical pressure, was assigned as the heat exchanger pressure, in this way, the condition of ORC being subcritical was met by all fluids.

In Figure 1, the T-s diagram of the ORC designed according to the values and assumptions determined in accordance with the waste heat temperature is shown.

Table 1. Refrigerants values table [miniREFPROPV9.5]

REFRİGERANTS	**Boiling Point Temp. (K)***,**	**Critical Temp. (K)** **	**Critical Pressure (kPa)** **
R290 (Propane,$C_3H_8(CH_3CH_2CH_3)$	231,06	369,83	4247,1
R600a (Isobutane, Methylpropane, $C_4H_{10}(CH(CH_3)_3)$	261,48	407,82	3640
R32 (Difluoromethane,CH_2F_2)	221,5	351,26	5782
R125 (Pentafluorethane, CF_3CHF_2)	225,06	339,17	**3617,7**
R143a (trifluoroethane, CF_3CH_3)	225,91	345,86	3761
* @1 atm, **miniREFPROP, Temp.: Temperature			

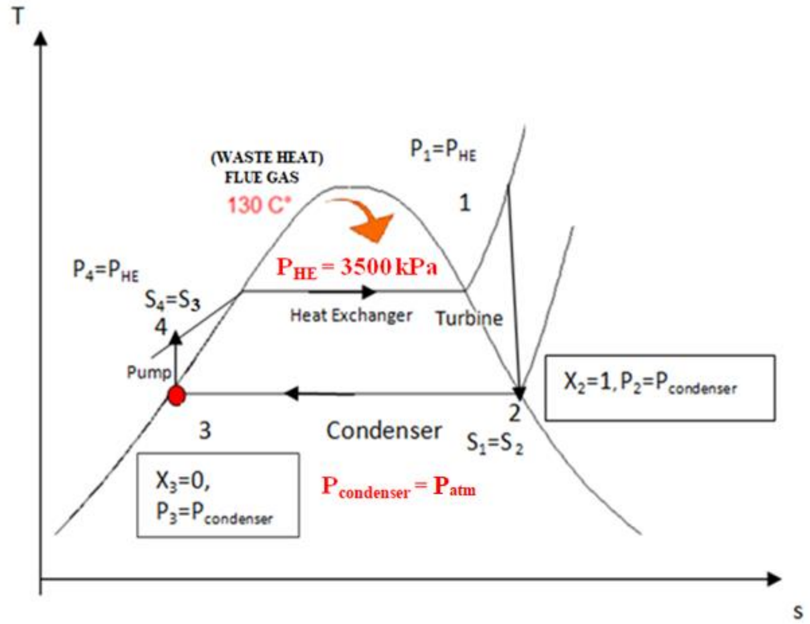

Figure 1. T-s diagram in ORC Design with assuptions and determined values

Heat exchanger pressure can be assumed as constant as follows:

$P_{HEATEXCHANGER} = P_4 = P_1 = 3500$ kPa (2.1)

According to the methods of increasing thermal efficiency, the condenser pressure should be selected low.When selecting the condenser pressure, it was selected the atmospheric pressure.The same condenser pressure value was taken for each of refrigerants.

$P_{CONDENSER} = P_2 = P_3 = P_{atm}$ (2.2)

The turbine outlet can be assumed as saturated vapor (due to the corrosion danger), and the condenser outlet can be assumed as saturated liquid as follows:

$x_2 = 1$ (Quality of Condenser Inlet= Quality of Steam Turbine Outlet)

$x_3 = 0$ (Quality of Pump Inlet = Quality of Condenser Outlet)

Since it was wanted to be fairer when comparing refrigerants, the same heat exchager pressure value and the same condenser pressure value were selected for each of refrigerants. Also, the same value of turbine outlet quality(x_2) and the same value of condenser outlet quality(x_3) were assigned for each of refrigerants. Heat transfer occurs due to temperature difference. Based on the principle that heat transfer occurs from hot to cold, the temperature of the sink where the condenser's heat is discharged is selected lower than the boiling point of the ORC refrigerant. In order to ensure the condition that the sink temperature is lower than the boiling point for all refrigerants, the refrigerant with the lowest boiling point is taken as basis, so the temperature value which is lower than the boiling point temperature of the refrigerant (R32) with the lowest boiling temperature among the all refrigerants was determined as the sink temperature. In the Table 2, the properties (Boiling Point Tempeature) obtained from the miniREFPROP V9.5 program, ODP and GWP values for each candidate refrigerant are shown and It is also possible to see the comparison of boiling point tempeatures the required that the sink temperaturewas determined.

While determining the sink temperature for the condenser, the boiling points of the candidate refrigerants were examined and a temperature value lower than the boiling point temperature of the refrigerant with the lowest boiling point among the candidate refrigerants was selected.

$T_{sink}=T_L=220$ K(Determined) (2.3)

Table 2. Refrigerants values table from miniREFPROP V9.5

REFRIGERANTS	**Boiling Point Temp. (K)*,****	**ODP**	**GWP(100 years)**
R290 (Propane,$C_3H_8(CH_3CH_2CH_3)$)	231,06	0***	3***
R600a (Isobutane, $C_4H_{10}(CH(CH3)_3)$)	261,48	0***	3***
R32 (Difluoromethane, CH_2F_2)	**221,5**	0***	675***
R125 (Pentafluorethane, CF_3CHF_2)	225,06	0****	3500****
R143a (trifluoroethane, CF_3CH_3)	225,91	0****	4470****
*@1 atm, **miniREFPROP V9.5, ***URL-1, ****URL-2			

2.2. Exergy Analysis

All exergy analyses and energy analyses were calculated per mass.

Exergy destruction results from entropy production. For this, first of all the entropy generation must be found, so the input and output entropies must be found.

$T_0=20$ ˚C (Assumption) (2.4)

Temperature of enviroment(ambient)

T_H=Temperature of heat source (2.5)

$x_{destructed} = x_d$ =Exergy that is destructed (2.6)

$s_{generation}$ = Entropy generated (2.7)

$s_{generation} > 0$(Real system) (2.8)

$s_{generation} = 0$ (Reversible system) (2.9)

The heat removed from the system (heat given off) is added to the exergy destruction(lost).The heat entering the system (heat received) is subtracted from the exergy destruction(lost).

$$x_{Heat} = \left(1 - \frac{T_0}{T_{Heatsource}}\right).q_{in} \quad (2.10)$$

Exergy transfer per mass with heat

$$\eta_{carnot} = \left(1 - \frac{T_0}{T_{Heatsource}}\right) \quad (2.11)$$

The Carnot efficiency

$$x_{mass}= m.\psi \quad (2.12)$$

Exergy transfer with mass transfer

$$\eta_{II} = \frac{w_{useful}}{w_{reversible}} \quad (2.13)$$

For work producing machine, for example turbine

$$\eta_{II}=\frac{w_{reversible}}{w_{useful}} \quad (2.14)$$

For work consuming machines, for example compressor, pump

Exergy AnalysisEquations in ORC

Since the pump and turbine are considered reversible, the destructed exergy of these components is zero.

$$x_{destructed} = T_0.s_{generation} \quad (2.15)$$

$$x_{destructed} = T_0.\left(s_{out} - s_{in} + \frac{q_{out}}{T_{sink}} - \frac{q_{in}}{T_{HeatSource}}\right) \quad (2.16)$$

$$x_{destructed,cycle}=x_{d,heatexchanger}+ x_{d,türbine}+ x_{d,condenser}+ x_{d,pump} \quad (2.17)$$

$$x_{supplied} = x_{thermal,in} + x_{pump} \quad (2.18)$$

$$x_{recovered} = x_{turbine} = w_{turbine} \quad (2.19)$$

$$\eta_{II,cycle} = \frac{x_{recovered}}{x_{supplied}} \quad (2.20)$$

$$\eta_{II,cycle}=1- \frac{x_{destructed,cycle}}{x_{supplied}} \quad (2.21)$$

Exergy Analysis of ORC's Components

Exergy Analysis of Turbine

$$\psi_1 - \psi_2 = \Delta\psi = (h_1 - h_2) - T_0.(s_1 - s_2) \quad (2.22)$$

$$w_{turbine}=h_1 - h_2 \quad (2.23)$$

$$q_{turbine} = 0 \quad (2.24)$$

$$x_{recovered} = x_{turbine}= w_{turbine} \quad (2.25)$$

$$\eta_{II,turbine} = \frac{w_{out}}{w_{reversible,out}} = \frac{h_1-h_2}{\psi_1-\psi_2} \quad (2.26)$$

(2.26)

or

$$\eta_{II,turbine} = 1 - \frac{T_0.s_{generation,turbine}}{\psi_1 - \psi_2} \quad (2.27)$$

$$\eta_{turbine,II} = \frac{\psi_1 - \psi_2 - (T_0.s_{generation,turbine})}{\psi_1 - \psi_2} \quad (2.28)$$

$$x_{in} - x_{out} - \cancel{x_{destructed}} = \cancel{\frac{dx_{cycle}}{dt}} \quad (2.29)$$

$x_{destructed}$ =0 reversible

$\frac{dx_{cycle}}{dt}$= 0 continuous flow

$x_{in} = x_{out}$

$$m.\psi_1 = m.\psi_2 + w_{reversible,out} \quad (2.30)$$

$x_{reversible,turbine} = \psi_1 - \psi_2$ per mass (2.31)

$\psi_2 = (h_2 - h_0) - T_0.(s_2 - s_0)$ (2.32)

$\psi_1 = (h_1 - h_0) - T_0.(s_1 - s_0)$ (2.33)

Exergy Analysis of Pump

$\eta_{II,pump} = \frac{w_{reversible,in}}{w_{in}} = \frac{\psi_4 - \psi_3}{h_4 - h_3}$ (2.34)

or

$\eta_{II,pump} = 1 - \frac{T_0.s_{generation,pump}}{h_4 - h_3}$ (2.35)

$s_{generation,pump} = s_4 - s_3$ (2.36)

$\psi_4 - \psi_3 = (h_4 - h_3) - T_0.(s_4 - s_3)$ (2.37)

$\psi_4 = (h_4 - h_0) - T_0.(s_4 - s_0)$ (2.38)

$\psi_3 = (h_3 - h_0) - T_0.(s_3 - s_0)$ (2.39)

Exergy Analysis of Heat Exchanger

$q_{in} = h_1 - h_4$ (2.40)

T_H=130˚C(Temperature of heat source)

Flue gas waste heat is used as the heat source for the shell tube heat exchanger.

$x_{destructed,heat\ exchanger} = T_0.(s_1 - s_4 - \frac{q_{in}}{T_{HeatSource}})$ (2.41)

$x_{thermal,in} = (1 - \frac{T_0}{T_h}).q_{in}$ (2.42)

$q_{in} = h_1 - h_4$ (2.43)

Exergy Analysis of Condenser

$x_{destructed,condenser} = T_0.(s_3 - s_2 + \frac{q_{out}}{T_{sink}})$ (2.44)

$q_{out} = h_2 - h_3$ (2.45)

$\psi_2 = (h_2 - h_0) - T_0.(s_2 - s_0)$ (2.46)

$\psi_3 = (h_3 - h_0) - T_0.(s_3 - s_0)$ (2.47)

2.3. Modelling Methodology and EES Programming

EES (Equation Engineering Solver) is a program used especially to solve thermodynamic problems, it is also possible to solve mathematical problems in this program. EES program has available functions and various fluid options for solving thermodynamic problems easily, and thermodynamic diagrams can also be drawn.

Exergy and energy equations were determined for each component of the organic rankine cycle and these equations and values assumed were entered into the EES equation window to calculate the second law efficiency in the EES program.

The functions are available in the EES program; knowing two values is sufficient to solve the functions. In the EES codes specified in the study, only the refrigerant name was changed and the same EES code was used.

EES was used to make the calculations required to measure the designed ORC performance, in Table 3the codes written in EES are showned. In order to correctly select the most suitable refrigerant, all refrigerants were evaluated under the same conditions, so while making the calculations for each of refrigerants, the same EES codes specified in Table 3 were used, only the name of the refrigerant was changed in the codes in the table below. In Table 3, the EES codes are applied on R32

3. RESULTS AND DISCUSSION

The heat source for the heat entering the heat exchanger is the flue gas waste heat, so the higher the q_{in} value, the more the flue gas is utilized.Since it is considered reversible adiabatic, the destructed exergy of the turbine($x_{destructed,turbine}$) and destructed exergy of pump are zero, but the heat exchanger's destructed exergy and condenser 's destructed exergy must be greater than zero and positive.

$x_{destructed,turbine} = 0$ (Because of reverversible, so $s_{generation,turbine}=0$)

$x_{destructed,pump} = 0$ (Because of reverversible, so $s_{generation,pump} = 0$)

At the end of this study, other values that should be considered are the second law efficiency of the cycle, the thermal efficiency of the cycle, and the work produced by the turbine. By comparing the second law efficiency of the cycle, the thermal efficiency of the cycle and the work produced by the turbine, the most suitable refrigerant was selected among the five candidate refrigerants.

In order to prove the accuracy of the studied program results and the ORC design, all of the following two values obtained as a result of the calculations in the ees program must be checked.

1-The second law efficiency must be lower than the first law efficiency

$\eta_{IIcycle} < \eta_{Thermal}$

2-In the ORC, the destructed exergy of condenser and the destructed exergy of heat exchanger must be greater than zero.

$x_{destructed,condenser} > 0$

$x_{destructed,heatexchanger} > 0$

According to the codes written in EES, all calculation results and are shown in Table 4a, Table 5a., Table 6a, Table 7a, and Table 8a, respectively, each Table shows a separate refrigerant calculation. Parametric tables were created for the important criteria found in the EES solutions table and the important values are shown in Table 4b, Table 5b, Table 6b, Table 7b, and Table 8b respectively

Table 3. EES codes display in equation window of R32 example

```
$UnitSystem SI;
P_3=P_2;    {Condenser inlet and conderser outlet are the same value, this value constant throughout the cycle}
P_1=P_4;    {Heat exchanger inlet and heat exchanger outlet are the same value, this value constant throughout the cycle}
P_4=P_heatexchanger
P_2=P_condenser
P_condenser=Po#                    {Determined Pressure value of Condenser}
P_heatexchanger=3500 [kPa]         {Determined Pressure value of Heat exchanger}

x_2=1     {Turbine Outlet quality}    "This value was taken because of the risk of corrosion"
x_3=0     {Condenser outlet quality }

T_WH=ConvertTEMP(C;K;130)   "Temperature of flue gas waste heat"
T_HeatSource=T_WH           {Flue gas waste heat was used as the heat source for the heat exchanger}
T_L=220[K]                  "Determined sink temperature for condenser"
T_0=ConvertTEMP(C;K;20)     {Environment Temperature}
P_0= Po#                    {Enviroment pressure, Atmospheric pressure was assigned as the ambient pressure value}

h_3=ENTHALPY(R32;P=P_3;x=x_3);     {Point of condenser outlet(inlet of pump)}
s_3=ENTROPY(R32;P=P_3; x=x_3);
T_3=TEMPERATURE(R32;P=P_3;x=x_3);
v_3=VOLUME(R32;P=P_3;x=x_3);

s_4=s_3;  "Since It is assumed that pump is reversible"
h_4=ENTHALPY(R32;P=P_4;s=s_4);       "Point of pump outlet(inlet of heat exchanger)"
T_4=TEMPERATURE(R32;P=P_4;s=s_4);
v_4=VOLUME(R32;P=P_4;s=s_4);

h_2=ENTHALPY(R32;P=P_2;x=x_2);       "Point of turbine outlet(inlet of condenser)"
s_2=ENTROPY(R32;P=P_2;x=x_2);
T_2=TEMPERATURE(R32;P=P_2;x=x_2);
v_2=VOLUME(R32;P=P_2;x=x_2);

s_1=s_2;      "Since It is assumed that turbine is reversible"
h_1=ENTHALPY(R32;P=P_1;s=s_1);       "Point of heat exchanger outlet(inlet of steam turbine)"
T_1=TEMPERATURE(R32;P=P_1;s=s_1);
v_1=VOLUME(R32;P=P_1;s=s_1);

s_0=Entropy(R32;T=T_0;P=P_0)
h_0=Enthalpy(R32;T=T_0;P=P_0)

h_1=h_4+q_in
h_1=w_turbine+h_2
q_out+h_3=h_2
w_pump+h_3=h_4
eta_thermal=(w_turbine-w_pump)/q_in

psi_1=(h_1-h_0)-T_0*(s_1-s_0)
psi_2=(h_2-h_0)-T_0*(s_2-s_0)
psi_3=(h_3-h_0)-T_0*(s_3-s_0)
psi_4=(h_4-h_0)-T_0*(s_4-s_0)

w_reversibleturbine=psi_1-psi_2
w_reversiblepump=psi_4-psi_3

x_destructedheatexchanger=T_0*(s_1-s_4-(q_in/T_HeatSource))
x_destructedturbine=T_0*(s_2-s_1)
x_destructedcondenser=T_0*(s_3-s_2+(q_out/T_L))
x_destructedpump=T_0*(s_4-s_3)

x_destructedcycle=x_destructedturbine+x_destructedpump+x_destructedcondenser+x_destructedheatexchanger

x_Thermalin=(1-(T_0/T_HeatSource))*q_in
x_supplied= x_Thermalin+w_pump
eta_IIcycle=1-(x_destructedcycle/x_supplied)
```

.

Table 4a. R32 solutions

Unit Settings: SI C kPa kJ mass deg

$\eta_{IIcycle}$ = 0,284	$\eta_{thermal}$ = 0,3049	h_0 = 839,5
h_1 = 666,8 [kJ/kg]	h_2 = 496,4	h_3 = 114,6
h_4 = 117,4	ψ_1 = 91,62	ψ_2 = -78,7
ψ_3 = 44,82	ψ_4 = 47,62	P_0 = 101,3 [kPa]
P_1 = 3500 [kPa]	P_2 = 101,3 [kPa]	P_3 = 101,3
P_4 = 3500 [kPa]	$P_{condenser}$ = 101,3 [kPa]	$P_{heatexchanger}$ = 3500 [kPa]
q_{in} = 549,4	q_{out} = 381,9	s_0 = 3,282
s_1 = 2,38 [kJ/(kg*K)]	s_2 = 2,38	s_3 = 0,6565
s_4 = 0,6565	T_0 = 293,2 [K]	T_1 = 167,2 [K]
T_2 = -51,65	T_3 = -51,65	T_4 = -50,77
$T_{HeatSource}$ = 403,2	T_L = 220 [K]	T_{WH} = 403,2 [K]
v_1 = 0,018 [m^3/kg]	v_2 = 0,3347	v_3 = 0,0008244
v_4 = 0,000822	w_{pump} = 2,798	$w_{reversiblepump}$ = 2,798
$w_{reversibleturbine}$ = 170,3	$w_{turbine}$ = 170,3	x_2 = 1
x_3 = 0	$x_{destructedcondenser}$ = 3,443	$x_{destructedcycle}$ = 109,3
$x_{destructedheatexchanger}$ = 105,9	$x_{destructedpump}$ = 0	$x_{destructedturbine}$ = 0
$x_{supplied}$ = 152,7	$x_{Thermalin}$ = 149,9	

Calculation time = 32 ms

Table 4b. R32 solutions in parametric tables

R32

1..1	1 $\eta_{IIcycle}$	2 $\eta_{thermal}$	3 q_{in}	4 $w_{reversibleturbine}$	5 $w_{turbine}$	6 $x_{destructedconde}$	7 $x_{destructedcycle}$	8 $x_{destructedheate}$	9 $x_{destructedpump}$	10 $x_{destructedturbin}$
Run 1	0,284	0,3049	549,4	170,3	170,3	3,443	109,3	105,9	0	0

Table 5a. R125 solutions

Unit Settings: SI C kPa kJ mass deg

$\eta_{IIcycle}$ = 0,02013	$\eta_{thermal}$ = 0,2461	h_0 = 621,6
h_1 = 363,3 [kJ/kg]	h_2 = 307,5	h_3 = 143,5
h_4 = 145,7	ψ_1 = -18,28	ψ_2 = -74,05
ψ_3 = -24,41	ψ_4 = -22,17	P_0 = 101,3 [kPa]
P_1 = 3500 [kPa]	P_2 = 101,3 [kPa]	P_3 = 101,3
P_4 = 3500 [kPa]	$P_{condenser}$ = 101,3 [kPa]	$P_{heatexchanger}$ = 3500 [kPa]
q_{in} = 217,5	q_{out} = 164	s_0 = 2,322
s_1 = 1,503 [kJ/(kg*K)]	s_2 = 1,503	s_3 = 0,7744
s_4 = 0,7744	T_0 = 293,2 [K]	T_1 = 74,51 [K]
T_2 = -48,09	T_3 = -48,09	T_4 = -47,04
$T_{HeatSource}$ = 403,2	T_L = 220 [K]	T_{WH} = 403,2 [K]
v_1 = 0,003947 [m^3/kg]	v_2 = 0,1472	v_3 = 0,0006607
v_4 = 0,0006575	w_{pump} = 2,241	$w_{reversiblepump}$ = 2,241
$w_{reversibleturbine}$ = 55,77	$w_{turbine}$ = 55,77	x_2 = 1
x_3 = 0	$x_{destructedcondenser}$ = 4,888	$x_{destructedcycle}$ = 60,35
$x_{destructedheatexchanger}$ = 55,47	$x_{destructedpump}$ = 0	$x_{destructedturbine}$ = 0
$x_{supplied}$ = 61,59	$x_{Thermalin}$ = 59,35	

Calculation time = 63 ms

Table 5b. R125 solutions in parametric tables

R125

1..1	1 $\eta_{IIcycle}$	2 $\eta_{thermal}$	3 q_{in}	4 $w_{reversibleturbine}$	5 $w_{turbine}$	6 $x_{destructedconde}$	7 $x_{destructedcycle}$	8 $x_{destructedheate}$	9 $x_{destructedpump}$	10 $x_{destructedturbine}$
Run 1	0,02013	0,2461	217,5	55,77	55,77	4,888	60,35	55,47	0	0

Table 6a. R143a solutions

Unit Settings: SI C kPa kJ mass deg

$\eta_{IIcycle}$ = 0,1044	$\eta_{thermal}$ = 0,2646	h_0 = 738,5
h_1 = 444,8 [kJ/kg]	h_2 = 360,3	h_3 = 133,7
h_4 = 136,6	ψ_1 = -5,674	ψ_2 = -90,11
ψ_3 = -22,65	ψ_4 = -19,74	P_0 = 101,3 [kPa]
P_1 = 3500 [kPa]	P_2 = 101,3 [kPa]	P_3 = 101,3
P_4 = 3500 [kPa]	$P_{condenser}$ = 101,3 [kPa]	$P_{heatexchanger}$ = 3500 [kPa]
q_{in} = 308,1	q_{out} = 226,6	s_0 = 2,722
s_1 = 1,739 [kJ/(kg*K)]	s_2 = 1,739	s_3 = 0,7359
s_4 = 0,7359	T_0 = 293,2 [K]	T_1 = 91,71 [K]
T_2 = -47,24	T_3 = -47,24	T_4 = -46,09
$T_{HeatSource}$ = 403,2	T_L = 220 [K]	T_{WH} = 403,2 [K]
v_1 = 0,006689 [m³/kg]	v_2 = 0,2097	v_3 = 0,0008574
v_4 = 0,0008531	w_{pump} = 2,907	$w_{reversiblepump}$ = 2,907
$w_{reversibleturbine}$ = 84,44	$w_{turbine}$ = 84,44	x_2 = 1
x_3 = 0	$x_{destructedcondenser}$ = 7,891	$x_{destructedcycle}$ = 77,9
$x_{destructedheatexchanger}$ = 70,01	$x_{destructedpump}$ = 0	$x_{destructedturbine}$ = 0
$x_{supplied}$ = 86,98	$x_{Thermalin}$ = 84,08	

Calculation time = 63 ms

Table 6b. R143a solutions in parametric tables

R143A

1..1	1 $\eta_{IIcycle}$	2 $\eta_{thermal}$	3 q_{in}	4 $w_{reversibleturbine}$	5 $w_{turbine}$	6 $x_{destructedconde}$	7 $x_{destructedcycle}$	8 $x_{destructedheate}$	9 $x_{destructedpump}$	10 $x_{destructedturbin}$
Run 1	0,1044	0,2646	308,1	84,44	84,44	7,891	77,9	70,01	0	0

Table 7a. R290 solutions

Unit Settings: SI C kPa kJ mass deg

(R290, Run 1)

$\eta_{IIcycle}$ = 0,1656	$\eta_{thermal}$ = 0,2772	h_0 = 1239
h_1 = 695 [kJ/kg]	h_2 = 525,8	h_3 = 100
h_4 = 105,8	ψ_1 = -12,18	ψ_2 = -181,4
ψ_3 = -66,93	ψ_4 = -61,1	P_0 = 101,3 [kPa]
P_1 = 3500 [kPa]	P_2 = 101,3 [kPa]	P_3 = 101,3
P_4 = 3500 [kPa]	$P_{condenser}$ = 101,3 [kPa]	$P_{heatexchanger}$ = 3500 [kPa]
q_{in} = 589,2	q_{out} = 425,8	s_0 = 4,262
s_1 = 2,449 [kJ/(kg*K)]	s_2 = 2,449	s_3 = 0,6057
s_4 = 0,6057	T_0 = 293,2 [K]	T_1 = 104,1 [K]
T_2 = -42,09	T_3 = -42,09	T_4 = -40,91
$T_{HeatSource}$ = 403,2	T_L = 220 [K]	T_{WH} = 403,2 [K]
v_1 = 0,01332 [m^3/kg]	v_2 = 0,4137	v_3 = 0,00172
v_4 = 0,001712	w_{pump} = 5,833	$w_{reversiblepump}$ = 5,833
$w_{reversibleturbine}$ = 169,2	$w_{turbine}$ = 169,2	x_2 = 1
x_3 = 0	$x_{destructedcondenser}$ = 27,16	$x_{destructedcycle}$ = 139
$x_{destructedheatexchanger}$ = 111,8	$x_{destructedpump}$ = 0	$x_{destructedturbine}$ = 0
$x_{supplied}$ = 166,6	$x_{Thermalin}$ = 160,8	

Calculation time = 47 ms

Table 7b. R290 solutions in parametric tables

R290

1..1	1 $\eta_{IIcycle}$	2 $\eta_{thermal}$	3 q_{in}	4 $w_{reversibleturbine}$	5 $w_{turbine}$	6 $x_{destructedconde}$	7 $x_{destructedcycle}$	8 $x_{destructedheate}$	9 $x_{destructedpump}$	10 $x_{destructedturbine}$
Run 1	0,1656	0,2772	589,2	169,2	169,2	27,16	139	111,8	0	0

Table 8a. R600A solutions

Unit Settings: SI C kPa kJ mass deg

$\eta_{IIcycle}$ = 0,01589	$\eta_{thermal}$ = 0,2441	h_0 = 1210
h_1 = 663,3 [kJ/kg]	h_2 = 539,4	h_3 = 173,5
h_4 = 179,2	ψ_1 = -67,26	ψ_2 = -191,1
ψ_3 = -146,8	ψ_4 = -141,1	P_0 = 101,3 [kPa]
P_1 = 3500 [kPa]	P_2 = 101,3 [kPa]	P_3 = 101,3
P_4 = 3500 [kPa]	$P_{condenser}$ = 101,3 [kPa]	$P_{heatexchanger}$ = 3500 [kPa]
q_{in} = 484,1	q_{out} = 365,9	s_0 = 3,937
s_1 = 2,301 [kJ/(kg*K)]	s_2 = 2,301	s_3 = 0,9012
s_4 = 0,9012	T_0 = 293,2 [K]	T_1 = 132,4 [K]
T_2 = -11,67	T_3 = -11,67	T_4 = -10,42
$T_{HeatSource}$ = 403,2	T_L = 220 [K]	T_{WH} = 403,2 [K]
v_1 = 0,006118 [m^3/kg]	v_2 = 0,3537	v_3 = 0,001684
v_4 = 0,001674	w_{pump} = 5,707	$w_{reversiblepump}$ = 5,707
$w_{reversibleturbine}$ = 123,9	$w_{turbine}$ = 123,9	x_2 = 1
x_3 = 0	$x_{destructedcondenser}$ = 77,36	$x_{destructedcycle}$ = 135,6
$x_{destructedheatexchanger}$ = 58,25	$x_{destructedpump}$ = 0	$x_{destructedturbine}$ = 0
$x_{supplied}$ = 137,8	$x_{Thermalin}$ = 132,1	

Calculation time = 32 ms

Table 8b. R600A solutions in parametric tables

R600a

1..1	1 $\eta_{IIcycle}$	2 $\eta_{thermal}$	3 q_{in}	4 $w_{reversibleturbine}$	5 $w_{turbine}$	6 $x_{destructedconde}$	7 $x_{destructedcycle}$	8 $x_{destructedheate}$	9 $x_{destructedpump}$	10 $x_{destructedturbine}$
Run 1	0,01589	0,2441	484,1	123,9	123,9	77,36	135,6	58,25	0	0

The first and second law efficiencies must logically be less than 1. Exergy destruction in irreversible systems must always be positive, so the exergy destruction of irreversible ORC components (heat exchanger, condenser) must be greater than zero. The second law efficiency can never be greater than the first law efficiency.
In Table 9, proof that the ORC design was designed correctly and the determined values and assumptions are logically correct can be seen for each of refrigerants.

Table 9. Proof of correctness conditions in ORC design

Conditions that must be met to prove the accuracy of the organic Rankine cycle	R32	R125	R143A	R290	R600A
$0<\eta_{IIcycle}<1$	✓	✓	✓	✓	✓
$0<\eta_{Thermal}<1$	✓	✓	✓	✓	✓
$\eta_{IIcycle}<\eta_{Thermal}$	✓	✓	✓	✓	✓
$x_{destructed,heatexchanger}>0$	✓	✓	✓	✓	✓
$x_{destructed,condenser}>0$	✓	✓	✓	✓	✓

The criteria determined for optimum refrigerant selection, these are second law efficiency, first law efficiency (thermal efficiency), turbine work, amount of utilized waste heat (heat used in the exchanger), condenser exergy destruction and heat exchanger exergy destruction, were tabled. Whenselecting the optimumrefrigerant, second law efficiency, first law efficiency and turbine work criteria were taken into consideration. The most important criterion taken into consideration is the second law efficiency value. The important values obtained as a result of the calculations can be seen in Table 10 for each of refrigerants.

Table 10. Results Calculated in EES for refrigerants comparison

Criterions	**R32**	R125	R143A	R290	R600A
$\eta_{IIcycle}$ (Second Law Efficiency)	**0,284**	0,02013	0,1044	0,1656	0,01589
$\eta_{Thermal}$	**0,3049**	0,2461	0,2646	0,2772	0,2441
$w_{turbine}$	**170,3**	55,77	84,44	169,2	123,9
q_{in}	**549,4**	217,5	308,1	589,2	484,1
$x_{destructed,heatexchanger}$	**105,9**	55,47	70,01	111,8	58,25
$x_{destructed,condenser}$	**3,443**	4,888	7,891	27,16	77,36

The refrigerant with the highest second law efficiency and thermal efficiency is R32 among the candidate refrigerants, the refrigerant that produces the most turbine work is also R32.After calculation, The results of the exergy and efficiency on the ORC design in EES are shown in Fig. 2 for R32, which was selected as the most suitable refrigerant.

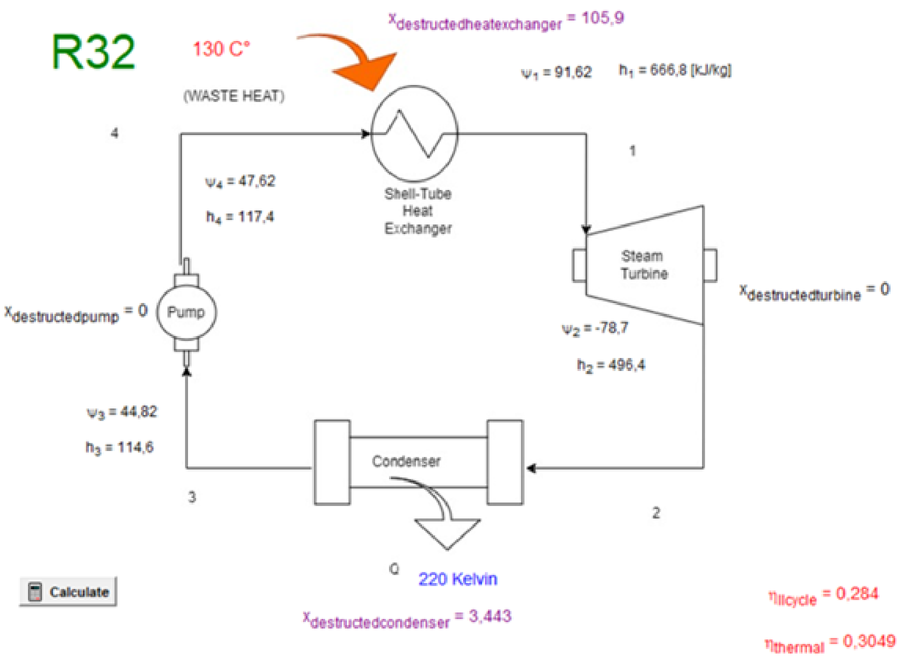

Figure 2. Calculation results of ORC design in EES for R32

4. CONCLUSIONS

In order to use the flue gas waste heat, an ORC suitable for the waste heat temperature has been designed. Five fluids were compared based on each refrigerant's thermal efficiency, second law efficiency, and turbine work produced.To prove that the ORC was designed correctly, some values from the exergy and energy calculations in EES were examined, these are thermal efficiency ($\eta_{Thermal}$), second law efficiency ($\eta_{IIcycle}$), destructed exergy value of each component in ORC (condenser, heat exchanger), work produced by the turbine ($w_{turbine}$), and the amount of heat taken into the heat exchanger (q_{in}). Care should be taken to ensure that the thermal efficiency and exergy efficiency (η_{II}) values are positive and less than oneand that the first law efficiency (thermal) is higher than the second law efficiency. The following conclusions were drawn from the article:

Both the thermal efficiency and second law efficiency values of R32 are higher than other refrigerants. Additionally, the maximum turbine work was obtained from R32.The ODP value of R32 is zero and it does not harm the ozone layer, but the GWP value of this refrigerant is 675. R32 is the most suitable refrigerant for the ORC due to its highest second law efficiency and the highest work produced by the steam turbine.

The power generation, first law efficiency and second law efficiency of R32 were found to be 170,3 kJ/kg, 0,3049 and 0,284, respectively.

In this study, the pool temperature was taken at a level lower than the boiling point temperature of the lowest boiling refrigerant among the candidate refrigerants, but it is possible to examine the ORC results at different pool temperatures. Another possible future study could be to change the heat exchanger pressure or to examine it at supercritical pressures.It is also possible to calculate the regression analysis and correlation betweenchangeable parameters and ORC efficiency and turbine work.

5. ACKNOWLEDGMENTS

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NOMENCLATURE

c	Specific heat capacity
COP	Coefficient of performance
EEF	Environmental effect factor
EES	Engineering Equation Solver
GWP	Global warming potential
IHE	Internal Heat Exchanger
h	Enthalpy
i-VCC-ORC	Integrated vapor compression cycle-ORC
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
P	Pressure
miniREFPROP	miniReference Fluid Propertiessoftware
TFC	Trilateral Flash Cycle
TSI	Thermo-sustainability indicator
P	Pressure
q	Heat per mass
s	Entropy
SPTC	solar parabolic trough collector

T	Temperature
Q	Heat Energy
q	Heat Energy per mass
v	Volume per mass
VVC	Vapor compression cycle
W	Work
WF	working fluids
w	Work per mass
x	Quality of refrigerant
x	Exergy per mass
X	Exergy

Greek Symbols

η	Efficiency
ψ	Exergy
Δ	Quantity of change(difference)
Δh	Quantity of enthalpy difference

Subscripts

atm	Atmosphere
cr	Critical Point
d	Destructed
f	Flue gas
HE	Heat Exchanger
H	Heat (Source)
hei	heat exchanger inlet
L	Low (Source)
WH	Waste Heat
0	Standard Temperature and Pressure

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