Analysis, economical and technical enhancement of an organic Rankine cycle recovering waste heat from an exhaust gas stream

Seyed Milad Shams Ghoreishi1 | Moslem Akbari Vakilabadi1 | Mokhtar Bidi2 | Alireza Khoeini Poorfar1 | Milad Sadeghzadeh3 | Mohammad Hossein Ahmadi4 | Tingzhen Ming5

1School of Mechanical Engineering, Iran University of Science and Technology (IUST), Tehran, Iran
2Mechanical and Energy Engineering Department, Shahid Beheshti University, Tehran, Iran
3Department of Renewable Energy and Environmental Engineering, University of Tehran, Tehran, Iran
4Faculty of Mechanical Engineering, Shahrood University of Technology, Shahrood, Iran
5School of Civil Engineering and Architecture, Wuhan University of Technology, Wuhan, China

Abstract

Any effort with the aim of increasing the total electrical power generation (EPG) due to the existed constraints in vessels is desired in power plants. Using Organic Rankine Cycle (ORC) for recovering waste heat of an exhausting gas is considered as an auxiliary way for improving EPG value. In this study, four ORC arrangements were first modeled for six refrigerants by EES software and the first and second laws of thermodynamics efficiency (ηI and ηII, respectively) and mass flow rate were studied for these cycles. Based on modeling results, the best arrangement was selected for each refrigerant such as; F-type (Reference cycle + Intermediate recuperator + Final recuperator) for R123, H-type (Reference cycle + Intermediate recuperator) for R114 and n-butane and R-type (Reference cycle + Final recuperator) for n-pentane, n-heptane, and toluene. The ηI value as a technical objective function was then optimized for the above cycles by Aspen HYSYS. n-heptane and toluene cycles were chosen due to higher first-law efficiency value and then were studied under different source temperature condition and n-heptane cycle showed better adaptability. Afterward, exergoeconomic was applied on n-heptane cycle and maximum cycle pressure was chosen as a design variable for economically optimizing net final income (NFI). Finally, NFI value is increased from 423.1$/kW·h to 609.9$/kW·h about 44.2%, while the second-law efficiency value is just decreased from 25.6% to 20.6% about 5%.

KEYWORDS

exergoeconomic, low temperature energy sources, net final income, organic fluids, organic Rankine cycle, waste heat recovery
Waste heat produced in a combustion process is considered as an amount of unused energy which is wasted to environment. Waste heat recovery results in efficiency improvement of power generation systems. In addition to waste heat, Solar, geothermal, and biomass are also regarded as low temperature heat sources to recover energy.

Organic Rankine Cycle (ORC) is employed for low-temperature heat sources to generate electrical power. ORC is a simple cycle with available refrigerants, flexible, and also reliable under different operation's condition. Nowadays, low-temperature heat sources are used widely around the globe. Organic fluids are considered as the promising working fluid in ORC power production systems. Organic fluids used in ORC have lower boiling point than water due to the higher molecular weight. Saturated steam curve in the T-S diagram is a criteria used for classification of organic cycles. Moreover, ORCs in which heat transfer of the working fluid occurs below the critical pressure of organic fluids, the working fluids go through a two-phase process, while this phase change will not occur in a supercritical state in transcritical cycles. In transcritical cycles the employed working fluid operates in both states of subcritical and supercritical.

There are a lot of papers focused on ORC for power generation from different low-temperature heat sources. Zhang et al evaluated the performance of a dual loop bottoming organic Rankine cycle (ORC) with a light-duty engine. An engine test bench was used to develop the performance map of the tested engine. Several significant parameters such as the exhaust, the intake air, and the coolant were discussed. Based on the reported results, higher output power would be achieved by using the low-temperature loop instead of the high-temperature loop. Moreover, a two percent improvement was resulted in the net total power and the thermal efficiency increased from 38% to 43% in the small load. The BSFC (brake specific fuel consumption) of the proposed cycle was also lowered considerably.

Freeman et al studied on the selection of used working fluid in the system composed of a solar system and the organic Rankine cycle and performed an optimization on the electrical performance of the proposed system in the United Kingdom. It was monitored that by utilizing HFC-245ca as the working fluid at an optimal evaporation saturation temperature of 126°C on the two-stage solar collector/evaporator layout, the highest total electrical work of 1070 kW would be produced in 1 year. The value of solar to electrical efficiency was reported here at 6.3%. The produced power in this condition covered the 32% of the electricity demand in the UK home. The proposed layout demonstrate better performance about 50% than the similar project conducted by Freeman et al.

Wang et al considered a hybrid structure consisting of a dual loop ORC with a conventional gasoline engine. The performance analysis of the presented structure showed that the low-temperature loop is more desired since it produced higher net output power. In addition, it was monitored that the total system efficiency increased from 3% to 6%.

Heberle et al considered zeotropic mixtures as the working fluid of the ORC in the geothermal applications. A thermodynamic analysis based on the second law was performed for the fluids including isobutene/isopentane and R227eq/R245fa. It was seen that using mixed fluid resulted in higher efficiency. This increase in the efficiency was about 15% in the low-temperature heat sources (eg, 120°C). Drescher and Brueggemann performed an investigation to obtain the working fluid of the ORC in biomass power and heat plants. Their study resulted the development of a software to evaluate fluids in order to be used in biomass power and heat plants. In final, the family of alkylbenzenes found to have the highest efficiency.

Heberle and Brueggemann carried out an investigation to select the suitable working fluid for being employed in a geothermal ORC. In series circuit, isopentane due to its high critical temperature obtained the highest efficiency. R227ea according to its low critical temperature was the suitable working fluid for power generation in the parallel circuit.

Quoilin et al presented a review study on techno-economic investigation of the ORC systems. Quoilin et al also performed a techno economic investigation on the waste heat recovery of the ORCs. A sizing assessment was done to predict the cycle performance at different operational conditions and by utilizing various working fluids. Based on the optimization results, n-butane was selected as the economical best selection by a total output power of 4.2 kW, corresponding to the overall efficiency of 4.5%. N-butane was also selected as the best fluid from thermodynamic optimization by total efficiency of 5.2%. Bao and Zhao performed a review on selection of working fluids and expanders in the ORC system. Oyewunmi and Markides studied a low-temperature organic Rankine cycle and performed a techno-economic analysis and heat transfer optimization of mixture working fluid over it. Using mixture working fluid, ie, 50% n-pentane+50% n-hexane, and 60% R245fa+ 40% R227ea resulted in a reduction in exergy losses since its non-isothermal phase-change behavior. In addition, the cycle net output power and total efficiency was increased in comparison to using pure fluid. Markides performed a comprehensive review on low-concentration solar systems based on ORCs for distributed-scale utilisations. Tchanche et al evaluated the economic feasibility of an ORC for waste heat recovery. Based on the experiments and evaluations, ORCs have high potential to be used in heat recovery applications. Mohammadi et al performed a thermo economic optimization on a power system including gas turbine, steam cycle and
an ORC. It was reported that the introduced combined system was able to achieve the exergy efficiency of 40.7% and annual product cost rate of 439 million $. Mirzaie et al. [22] considered several working fluids for using in the ORC which smelting furnace gases was defined as the heat source and carried out energy, exergy, and economic analyses. It was reported that m-xylene, P-xylene, and Ethylbenzene demonstrated higher efficiency and lower total cost, respectively. Ashouri et al. [23] evaluate a solar double pressure ORC and carried out exergy and exergoeconomic analyses. It was monitored that the system had this ability to provide the power in day and night, with a solar fraction of 100% diurnal and in night by the designed storage tank and the auxiliary heater. In overall, the system was able to achieve the efficiency of 22.7% and production cost rate of 2.6 million $/year. Zhang et al. [24] evaluate the ORC for waste heat recovery from the respect of emery. Based on the emery analysis, the emery yield ratio and the emery index of sustainability of the proposed system was obtained by 197.5 and 3.9, respectively. It was monitored that the sustainability of the ORC is higher than fossil fuel power plants but renewable power plants have higher sustainability. The used working fluid, ie, R134a, formed 13.3% of the total input emery flows in the construction stage. In geothermal field, Hettiarachchi et al. [25] presented a cost-effective optimum design for ORC by optimizing the ratio of the total heat exchanger area to the net output power as an objective function and using ammonia, HCFC123, n-Pentane and PF5050 as working fluids. Results showed that the objective function was minimum for ammonia cycle, but is not necessarily the maximum cycle efficiency. Saleh et al. [26] studied 31 organic fluids for using in the subcritical and supercritical conditions of geothermal power plants. Results showed that n-butane had the highest thermal efficiency with $\eta_{th} = 0.13$. Nami et al. [27] studied system components interactions in ORC using advanced exergy analysis. Results showed that the total system exergy destruction could be improved for 7% by decreasing the condenser exergy destruction for 15%.

In solar field, Tchanche et al. [28] studied and compared 20 organic fluids used as working fluids of an ORC system in low-temperature solar systems. Results revealed that R134a appeared as the most suitable working fluid for small scale solar applications. Taccani et al. [29] also studied Small Scale Solar Powered ORC using R-245fa. Experimental data showed that gross electrical efficiency was obtained up to 8%.

In waste heat recovery application, Hung et al. [30] studied thermal efficiency of 6 organic fluids including benzene, ammonia, R12, R11, R134fa, and R113 under M-type ORC arrangement. The authors concluded that isentropic fluids (ie, fluids that have infinite slope in the vapor curve diagram are called isentropic fluids) had better performance for low grade waste heat. Liu et al. [31] studied effects of 10 organic fluids on the thermal efficiency of an ORC system. Results showed that efficiency will be increased by the rise of the source waste heat temperature and selecting organic fluids with higher critical temperature as working fluids. Wang et al. [32] also studied 9 organic fluids in ORC for constant power generation of 10 kW. Results indicated that R11, R141b, R113, and R123 had better thermodynamic performances, while R245fa and R245ca are considered as more environmentally friendly working fluid than other investigated fluids. Ozdil et al. [33] also analyzed thermodynamically an ORC cycle based on the industrial data. Results indicated that cycle exergy efficiency increased as the pinch point temperature decreased and energy and exergy efficiency were computed 9.96% and 47.22%, respectively. Li et al. [34] experimentally studied ORC performance under different operating conditions. He concluded that system thermal efficiency could be optimized by the increase of the heat source temperature and working fluid pump speed. Seyedkavoosi et al. [35] presented an exergy-based optimization on ORC for heat recovery from an internal heat engine by using R-123, R-134a, and water as working fluids. The results revealed that R-123 is the most suitable working fluid by output power and exergy efficiency of 468 kW and 21%, respectively.

Besides technical point of view, a heat recovery system should be also designed from financial depreciation, development, investment return period, and return rate points of view.

In this study six working fluids including; R123, R114, n-butane, n-pentane, n-heptane, and toluene under 4 types of ORC arrangement are investigated in the waste heat recovery applications. The optimum cycle arrangement for each working fluid was first proposed by thermodynamic modeling of the cycles in EES with respect to high thermal efficiency, high exergy efficiency and low mass flow rate. The best arrangement for each working fluid was then technically optimized to improve its thermal efficiency by Aspen HYSYS. The technical section of this paper consists of two parts: modeling and optimization. In the modeling section, the conventional refrigerant used in the previously published papers, based on actual operating conditions, in which the minimum refrigerant pressure in the cycle should be greater than atmospheric pressure, is modeled in the EES software and a parametric analysis in these new conditions is then presented. In these models, maximum cycle efficiency is obtained in a range of temperature, until it is determined that the best performance of each refrigerant (in real operational condition) at which temperature and configuration is achievable. In the optimization section, the selected configurations of each refrigerant in the HYSYS software are refined with the assumption that the temperature of the thermal source is 350 degrees Celsius (in real operational condition). Cycles performances then were investigated under the effects of different operating conditions on $\eta_1$ and $\eta_{th}$ such as; evaporator pressure, ratio of extracted pressure in turbines and degree of
superheating. Also, all of the studied fluids and their optimal configurations have been compared with previous published papers with respect to the actual operating conditions. As a result, n-heptane under R-type arrangement was selected as the optimum cycle with the higher thermal efficiency than the others. The above cycle was analyzed by an exergoeconomic approach and maximum cycle pressure was then selected as a key variable of the techno-economic optimization. Using Aspen HYSYS, the optimum state was simultaneously computed for different values of the ratio of the net generated power to the input thermal power (as a technical objective function) and final net income (as an economic objective function). In this paper, the minimum cycle pressure is considered above the atmospheric pressure (110 kPa) to avoid air infiltration into the system; few studies have considered this factor.

2 | SYSTEM DESCRIPTION

In this article, ORC is investigated for low grade waste heat application in a vessel's exhaust (chimney).

2.1 | Cycle arrangements and working fluids

The main components in ORCs could be arranged in various methods in which the most common ORC arrangements shown in Figures 1-4, are as follows:

- **M-type arrangement**: Reference cycle which is consist of an evaporator, a turbine, a condenser and pumps.
- **R-type arrangement**: Reference cycle + Final recuperator
- **H-type arrangement**: Reference cycle + Intermediate recuperator
- **F-type arrangement**: Reference cycle + Intermediate recuperator + Final recuperator

2.1.1 | Working fluid selection

The choice of working fluid is carefully selected by considering the following factors:

1. Based on the slope of the vapor saturation of fluids

Monitoring the T-S diagram of fluids demonstrates that the fluids have different slopes in their saturated vapor curve and can be classified into three general categories:

- Dry fluids that have a positive gradient and usually have a higher molecular weight than other fluids, such as n-heptane.
- Isentropic fluids with extremely high (infinite) slope of vapor curves, such as R11 and R12.
- Wet fluids that have a negative slope in the saturated vapor curve. These fluids usually have a lower molecular weight than the two previous ones, such as carbon dioxide and water.
This positive slope of the saturated vapor curve in dry fluids eliminates the possibility of low-quality flow through a turbine, eliminating the complexity of the design of the cycle as well as removing the required de-aeration equipment. The degree of organic fluid dryness is generally dependent on the molecular mass of the fluid, as shown in the figure.

One of the issues that underlies this atmospheric pressure is the lack of filtering of non-condensing gases into the operating fluid, which will increase the requirement of de-aeration equipment in the system.

2. Based on the critical temperature of fluids

Examining different fluids in the organic cycle concludes that fluids with a higher critical temperature exhibit higher thermodynamic efficiency. Therefore, in the selection of working fluids, fluids with a higher boiling point are preferable.

3. Availability and non-toxicity of fluids

Another reason for choosing the fluids is to pay attention to the common fluids that are currently in use in the organic cycles. This is useful for comparing the obtained results of this study with running cycles.

In this study, six organic fluids such as R-123, R-114, n-butane, n-pentane, n-heptane, and toluene were selected as working fluids and investigated under above arrangements.

The M-type arrangement is considered as simple reference arrangement for ORC and its schematic is illustrated in Figure 5.

The R-type arrangement is obtained by adding a final recuperator to the reference arrangement (M-type) shown in Figure 6. In this cycle, the refrigerant temperature in the heat exchanger, LNG-100, increases, which plays the role of the final recuperator, and receives the rest of the heat load from the heat source, E-102, and then through the turbine, K-100, power is produced and then after passing through the recuperator, its temperature decreases in condenser, E-101, and reaches the saturation temperature of the working fluid.

H-type arrangement is derived same as R-type arrangement except using intermediate recuperator (a closed feed water heater) instead of final recuperator (Figure 7). In this type of cycle arrangement, the refrigerant pressure in the pump, P-100, increases and its temperature raises in the closed feed water heater, LNG-100. After that, it enters into the mixer, MIX-100 and after exchanging heat with the high temperature heat source, its temperature increases and in the turbine, K-100, generates electricity up to the pressure, where a fraction of the refrigerant is extracted. The extracted steam from the turbine enters into the closed feed water heater, LNG-100, to supply the required heat load and then its pressure increases in pump, P-101, up to the cycle maximum pressure and then enters into the mixer. The rest of the output flow from the turbine K-100, enters into the turbine K-101 and expands to the minimum pressure of the cycle. Finally, its temperature reaches the saturation temperature of the liquid through the condenser, E-101.

Figure 8 depicts the F-type arrangement of an ORC which included both intermediate (a closed feed water heater) and final recuperators in the reference arrangement. In this arrangement, the refrigerant pressure in the pump P-101 increases to the maximum pressure and then its temperature raises in the final recuperator, LNG-101, and then it goes into the closed feed water heater, LNG-100, the temperature of the stream will also increase. Afterwards, it enters into the final recuperator, LNG-101, and after exchanging heat with the high-temperature heat source, E-100, the temperature increases and will generate power in the turbine, K-100, up to the extracted stream pressure. The extracted stream of the turbine enters into the closed feed water heater, LNG-100, to supply the needed heat load and subsequently goes through pump, P-100. Its pressure reaches the maximum pressure of the cycle and then goes into the mixer. The rest of the output stream from the turbine, K-100, enters into the turbine, K-101, and expands till the minimum pressure of the cycle. Finally, its temperature reaches the saturated liquid temperature through the condenser E-102.

2.2 | Process simulation and assumption

Four common types of ORC arrangement (Figures 4-8) were simulated for all six working fluids using Aspen HYSYS. Peng-Robinson and LKP (Lee Kesler Plocker) equations...
SHAMS GHOREISHI ET AL.

were employed for equation of state for the working fluids in this simulation. The assumptions and operating conditions used in this simulation are also given in Table 1.

3 | METHODOLOGY

The common thermodynamics analysis is essentially based on the first law of thermodynamics that states the principle of energy conservation. Energy analysis of an energy conversion system is based on the study of the input and output amount of energy in a system. In above analysis, generally, efficiencies were defined as the ratio of energy values and were often used to evaluate and compare different systems. Energy balance are provided in Table 2. This table includes the characteristic points of the F-type cycle which cycle involves all the components that needs to be studied.

Exergy analysis is another thermodynamic analysis based on the second law of thermodynamics used for comparison among different cycles and processes. Exergy analysis demonstrates the proximity of the actual performance of a process to its ideal performance. Also, it implies the causes for the thermodynamic losses. Therefore, exergy analysis is a powerful tool to optimize system design. Energy efficiency describes system quantitatively not qualitatively. While, exergy analysis indicates the fundamental deficiencies of the system. So, exergy analysis can resolve many defects of the energy analysis by identifying the causes, locations and scale of the inefficiency of a process.

Exergy equation can be obtained by combining the first and second laws of thermodynamics. This concept is primarily derived from the works of William Rankine, Rudolf Clausius, and William Thomson that later became known as Lord Kelvin in the 1850s.

In a process, the difference between the total exergy that flows into and exits from the system, except the exergy stored, is known as exergy loss and expressed as follows:

$$ I = T_0 S_{gen} $$

when energy is transferred as work to the surroundings, exergy is also transmitted. It means that, by transferring energy to the environment through work, there is no exergy losses but if the energy being transferred through heat, due to the temperature difference between the system and the environment,
part of the exergy would be destructed during transferring into the environment.

The relationship between the work and the related exergy is simple, the maximum work can be achieved in the absence of any waste, equals to the amount of work done. Therefore, the exergy, $E_W$, related to the transfer of work done, $W$, is given as:

$$E_W = W$$ (2)

If the heat is transferred from one system to another, a certain amount of exergy will be transferred. The absolute value of exergy leaving the warmer system is always larger than the amount of exergy received by the cooler system. In the case of ideal reversible heat transfer, the two exergy values are equal.

The amount of the exergy involved in heat transfer can be obtained from the fundamental concept of exergy, it means the maximum work produced on a continuous basis, which is a work achieved through a reversible carnot cycle between the temperature of the heat source and the lowest temperature possible (the environment or heat sink).

If the amount of heat, $Q$, of a system at the temperature, $T$, transfers to another system at a lower temperature in an environment with a dead state temperature of $T_0$, in this case, the exergy, $E_Q$, related to heat transfer, $Q$, is equal to:

$$E_Q = \left[1 - \frac{T_0}{T}\right] Q$$ (3)

It is assumed that the system works in a continuous flow condition with multiple input and output streams. The system receives the heat energy flow from the hot reservoir at the temperature $T$ and transfers power to the final consumer in the environment which is characterized by the dead state temperature $T_0$.

Exergy calculation for an open system involves exergy of each flow and each component of energy transfer. Unlike the energy calculation that there should be always a balance, based on the first law, the real exergy calculation shows that less exergy value leaves the system than enters it. In other words an amount of exergy is being destructed. If all causes that lead to exergy losses to disappear and all processes are
TABLE 1 Assumptions and operating conditions used in ORC simulation

| Parameter                                      | Values            |
|------------------------------------------------|-------------------|
| Ambient pressure and temperature (kPa/°C)       | 101/25            |
| Minimum pressure of condenser (kPa)             | 110               |
| Evaporator pressure (kPa)                       | 2000              |
| Adiabatic efficiency of the Pump                | 80%               |
| Adiabatic efficiency of the Turbine             | 88%               |
| Temperature of the hot source (°C)              | 350               |
| Temperature of the cold source (°C)             | 25                |
| Recuperator performance coefficient             | 0.9               |
| Output temperature of the condenser (°C)        | 35                |
| Output fluid of the evaporator                  | Saturated vapor   |
| The inlet fluid to the pump                     | Saturated liquid  |
| Pressure drop in heat exchanger equipment (kPa) | 0                 |
| The flow rate of the exhaust (m³/h)             | 450 000           |
| The net power generated (kW)                    | 100               |
| Minimum temperature approach (°C)               | 2                 |

TABLE 2 Energy balance for main equipment in F-type arrangement

| Component     | Energy balance                                                                 |
|---------------|---------------------------------------------------------------------------------|
| Pump          | \( W_{\text{pump}} = \dot{m} \times (h_2 - h_1) \)                             |
| Evaporator    | \( Q_{\text{evaporator}} = \dot{m} \times (h_6 - h_3) \)                      |
| Turbine       | \( \eta_{\text{turbine}} = \frac{\dot{m} \times (h_6 - h_1)}{\dot{m} \times (h_6 - h_3)} \) |
| Condenser     | \( Q_{\text{condenser}} = (1-y) \times \dot{m} \times (h_{11} - h_1) \)         |
| Recuperator   | \( (1-y) \times h_2 + y \times h_{10} = (1-y) \times h_1 + y \times h_4 \)      |
| CFFH          | \( (1-y) \times h_7 + y \times h_3 = (1-y) \times h_4 + y \times h_4 \)         |

Carried out reversible, there would be no loss of exergy (Ideal condition).

\[
\dot{I} = \dot{m} \times \frac{dS_{\text{out}}}{dt} = \dot{m} \times T_0 \times \left[ \sum e S - \sum i S + \frac{dS_{\text{sys}}}{dt} + \sum q_i T \right] \tag{4}
\]

It is assumed that the fluid flow to be a steady flow therefore:

\[
\frac{dS_{\text{sys}}}{dt} = 0 \tag{5}
\]

If one input and one output for each of the components is taken into account:

\[
\dot{I} = \dot{m} \times T_0 \times \left[ s_e - s_i + \frac{q}{T} \right] \tag{6}
\]

The term, \( q \), is measured relative to the environment, it means that the heat transfer to the evaporator is negative and will be positive to the condenser. The term \( T \) in brackets is the temperature of the hot and cold source and should not be confused with the ambient temperature.

3.1 Economic modeling

In an economic modeling of a system, one should estimate the annual investment cost, operating, maintenance, and total cost of system. All mention costs are computed annually. In this method, the capital-recovery factor for equivalent amounts in an investment is defined as below:

\[
\text{CRF} = \frac{i(1+i)^n}{(1+i)^n - 1} \tag{7}
\]

where \( i \) is the interest rate and \( n \) is the number of the operation years which in this paper are assumed to be 10% and 30 years, respectively. The price of electricity is $0.09 (9 cents) per kilowatt hour.

In the analysis and optimization of energy systems, the carrying charge, fuel costs, operating, and maintenance costs of the systems must be calculated. In this section, the system’s total revenue requirement (TRR) is used to analyze the system economics, which is the method proposed by the Institute for Electric Power Research Institute (EPRI). This method calculates all the costs which are included such as the capital return. In this method, based on the economic assumptions and the calculation of the purchase price of equipment and fuel, the final required revenue is calculated year by year. Finally, all costs, including operation and maintenance costs, as well as fuel costs, are aligned on an annual basis.

3.1.1 Estimation cost of equipment

The best way to estimate costs is to use the manufacturer’s catalog. The following equations have been used in this research:

**Pump**

\[
\text{PEC}_{\text{pump}} = \frac{C_1 \times \dot{m} \times \eta_{is}}{C_2 - \eta_{is}} \times \text{Ln} \left( \frac{P_{\text{out}}}{P_{\text{in}}} \right) \times \frac{P_{\text{out}}}{P_{\text{in}}} \tag{8}
\]

where \( \text{PEC} \) indicates the purchase equipment cost, \( \dot{m} \) is the mass flow rate and \( \eta_{is} \) indicates the isentropic efficiency. \( C_1 \) and \( C_2 \) are constants, 573 s/kg and 0.8996, respectively. The isentropic efficiency is assumed to be 0.8.

**Heat exchangers**

\[
\text{PEC}_{\text{heatexchanger}} = A_f \times (C_a + C_b A_f) \tag{9}
\]
where \(A_f\) is the area of the heat exchanger. \(C_{at}, C_{bt},\) and \(A_f\) are constants, 0.322, 30,000, and 0.75, respectively.

**Turbine:**

\[
\frac{\text{PEC}_{\text{turbine}}}{C_3} = \frac{\dot{m}}{C_{32} - \eta_{is}} \times \ln \left( \frac{P_{in}}{P_{out}} \right) \times \left[ 1 + \exp \left( C_{33} T_{in} - C_{34} \right) \right] \tag{10}
\]

\[
\text{PEC}_{\text{turbine}} = 1000 \times 0.378 \times (\text{HP})^{0.81} \tag{11}
\]

\[
\text{HP} = \frac{W}{\eta_{is} \times 735.5} \tag{12}
\]

### 3.1.2 Calculation of the final cost of fixed investment

The annual final cost of the system is defined as the total cost of the system in 1 year includes the cost of product production and operating system costs, which consists of two parts:

1. **Carrying charge:** Expenditures include total capital recovery (TCR), return on investment (ROI), and Other Costs Including Taxes, and so on.

2. **Expense:** Expenses include fuel costs and repair and maintenance costs.

The total of carrying cost and expenses represents the total cost of the system in 1 year.

First, the total net investment which includes the purchase equipment cost and the cost of installation is calculated:

\[
\text{PEC}_k = \sum \text{PEC}_k \tag{13}
\]

PEC\(_k\) shows the purchasing cost of each equipment. Since the main reference for this study is in the Bejan\(^{39}\) in the Exergoeconomic section, the simplified relationships of this reference are used in accordance with Table 3.

\[
\text{TCI} = \text{FCI} + \text{SUC} + \text{WC} + \text{LRD} + \text{AFUDC} \tag{14}
\]

\[
\text{FCI} = \text{DC} + \text{IC} \tag{15}
\]

\[
\text{DC} = \text{ONSC} + \text{OFSC} \tag{16}
\]

\[
\text{ONSC} = 0.5 \times \text{OFSC} \tag{17}
\]

\[
\text{LRD} = \text{AFUDC} + 0.15\text{FCI} \tag{18}
\]

\[
\text{IC} = 0.25\text{DC} \tag{19}
\]

\[
\text{TCI} = 1.47\text{FCI} \tag{20}
\]

where TCI, FCI, SUC, WC, LRD, AFUDC, IC, DC, OFSC, and ONSC indicate the total capital investment, fixed capital investment, startup cost, working capital, research and development cost, allowance for using funds during construction (1-5 years), indirect cost, direct cost, off-site cost, and on-site cost, respectively.

Finally, after the placement of equations of the process:\(^{35}\):

\[
\text{PEC}_{\text{tot}} = \text{Initial cost of equipment investment (Total equipment prices).} \tag{14}
\]

Where the total capital investment (TCI) is defined as follows:

\[
\text{Total capital investment} = \text{Fixed capital investment} + \text{Other outlays} \tag{24}
\]

\[
\text{Total capital investment} = \text{direct costs} + \text{indirect costs} + \text{other outlays} \tag{25}
\]

### 3.1.3 Total capital recovery

It is assumed that the price of equipment is reached to zero at the end of its lifetime. With this assumption, total capital recovery (TCR\(_j\)) is obtained by the following equations:\(^{35}\):

\[
\text{TDI} = \text{TCI} \tag{26}
\]

\[
\text{TCR}_j = \text{BD}_j \quad j = 1, \ldots, \text{BL} \tag{27}
\]

\[
\text{BD}_j = \frac{\text{TDI}}{\text{BL}} \quad j = 1, \ldots, \text{BL} \tag{28}
\]

where BD\(_j\) is the annual system depreciation cost, BL denotes the number of working years in the system (in this research is 30 years), TCR\(_j\) indicates the annual capital recovery that includes other items such as taxes which is ignored in this analysis, and TDI is the total depreciable investment.

### 3.1.4 Minimum return on investment

Return on investment indicates the amount of investment which is not calculated. This cost, which is annually, is in fact the profit of the initial investment of the system, which is added to the cost of the initial investment. In order to calculate the return on investment, first the cost balance is
calculated per year. The cost balance at the beginning of the first year is equal to the initial investment cost.35

The cost balance is calculated at the beginning of each year from the reduction of the cost of the previous year from the amount of annual depreciation.

The annual return on investment is calculated as follows:

\[
BB_{Y1} = TD_1
\]  

(29)

The cost balance is calculated at the beginning of each year from the reduction of the cost of the previous year from the amount of annual depreciation.

\[
BB_{Yj} = BB_{Yj-1} - BD_j
\]  

(30)

The annual return on investment is calculated as follows:

\[
ROI_j = BB_{Yj} \times i \quad j = 1, ..., BL
\]  

(31)

where \(i\) is the annual profit rate of the system, which is considered 10% in this research.

### 3.1.5 Annual fuel cost

The cost of fuel in this system is the cost of steam, which is calculated in the first year as follows:

\[
FC_{\alpha,steam} = 3600 \times C_f \times \dot{W} \times \tau
\]  

(32)

where \(\tau\), \(\dot{W}\) and \(C_f\) are The operating hours of the system (365 × 24 hours), inlet steam mass flow rate into the system, and the steam cost constant (0.003 $/Mj), respectively.

The steam cost at the year of \(j\)th is calculated as follows:

\[
FC_{\alpha,steam} = FC_{O,steam} \times (1 + r_{FC,steam})^j
\]  

(33)

\(r_{FC,steam}\) denotes the inflation rate associated to the increase in steam prices is considered to be 5% in this research.

### 3.1.6 Annual maintenance cost

The maintenance cost is usually considered as a fraction of the annual direct cost. The maintenance costs for the first year of operation is considered as the 2% of the annual direct cost.35

The maintenance cost in the \(j\)th year of the system’s operation is calculated from the following equation:

\[
OMC_j = OMC_1(1 + r_{OM})
\]  

(35)

The inflation rate associated to the maintenance cost is considered 5% in this research.35

### 3.2 Total annual revenue (TRR)

The total required annual revenue in the \(j\)th year of operation is obtained from summing the reimbursement cost, return on investment, fuel costs, and maintenance cost in the \(j\)th year.35

\[
TRR_j = TCR_j + ROI_j + afc_{j,steam} + OMC_j
\]  

(36)

### 3.3 Cost levelization

A Comparison is done to obtain the effect of different costs in the system by comparing the annual cost of various factors such as fuel costs, maintenance costs and equipment. The total annual cost over the years is increasing with years of system operation. Therefore, there is a need for leveling these costs. The total levelized required annual revenue (cost) is calculated as follows35:

\[
TRR = CRF \sum_{i=1}^{n} \frac{TRR_i}{(1+i)^j}
\]  

(37)

The fuel cost is discarded since the temperature in the chimney system provides the needed energy to operate the system. The maintenance cost is increasing year by year and is obtained by using the annual cost of the first year as follows:

---

**TABLE 3 Breakdown of total capital investment (TCI)**39

| Fixed Capital Investment (FCI) |
|--------------------------------|
| **A. Direct Costs (DC)** |
| 1. Onsite Costs (ONSC) |
| Purchased equipment cost (PEC: 15%-40% of FCI) |
| Purchased equipment installation (20%-90% of PEC; 6%-14% of FCI) |
| Piping (10%-70% of PEC; 3%-20% of FCI) |
| Instrumentations and Controls (6%-40% of PEC; 2%-8% of FCI) |
| Electrical equipment and materials (10%-15% of PEC; 2%-10% of FCI) |
| 2. Offsite costs (OFSC) |
| Land (0%-10% of PEC; 0%-2% of FCI) |
| Civil, structural, and architectural work (15%-90% of PEC; 5%-23% of FCI) |
| Service facilities (30%-100% of PEC; 8%-20% of FCI) |
| **B. Indirect Costs (IC)** |
| 1. Engineering and supervision (25%-75% of PEC; 6%-15% of DC; 4%-21% of FCI) |
| 2. Construction costs including contractor’s profit (15% of DC; 6%-22% of FCI) |
| 3. Contingencies (8%-25% of the sum of above costs; 5%-20% of FCI) |
| **II. Other outlays** |
| A. Startup costs (5%-12% of FCI) |
| B. Working capital (10%-20% of TCI) |
| C. Costs of licensing, research, and development |
| D. Allowance for funds used during construction (AFUDC) |

---

The maintenance cost is usually considered as a fraction of the annual direct cost. The maintenance costs for the first year of operation is considered as the 2% of the annual direct cost.35

\[
OMC_1 = 0.02 \text{PEC}_{tot}
\]  

(34)

The maintenance cost in the \(j\)th year of the system’s operation is calculated from the following equation:

\[
OMC_j = OMC_1(1 + r_{OM})^j
\]  

(35)

The inflation rate associated to the maintenance cost is considered 5% in this research.35

The total required annual revenue in the \(j\)th year of operation is obtained from summing the reimbursement cost, return on investment, fuel costs, and maintenance cost in the \(j\)th year.35

\[
TRR_j = TCR_j + ROI_j + afc_{j,steam} + OMC_j
\]  

(36)
where \( K_{\text{OMC}} \) indicates the rate of increase in annual maintenance costs.

Finally, the Levelized annual carrying charge is calculated from the following equation:

\[
\text{CCL} = \text{TRRL} - FCL_{\text{steam}} - FCL_{\text{CW}} - \text{OMCL}
\]  

### 3.3.1 Annualized maintenance cost, total cost

Annual maintenance and repairing cost were predicted by considering coefficient of 1.06. Annual maintenance and repairing cost were predicted by considering coefficient of 1.06. By summation of annualized investment and maintenance costs, Total cost is obtained.

### 3.4 Exergoeconomic analysis

The objectives of the exergoeconomic analysis are as follows:

1. Estimation of the cost of each product generated by the system with more than one product separately.
2. The cost formation process and the flow costs are determined in the system.
3. The specific parameters are independently optimized in a single component.
4. The cost related to the overall system to be minimized.

Due to the costs changes from year to year, when the design of a thermal system is assessed from the economic point of view, cost levelization approach could be used. Therefore, the amounts of costs in an exergoeconomic analysis are levelized.

To estimate the costs in an exergoeconomic analysis, the cost balance is required which is used in this study. The cost balance and flow cost rate \( \dot{Z} \) for the entire system operating at steady state condition is expressed as follows:

\[
\dot{Z} = \dot{Z}_{\text{CI}} + \dot{Z}_{\text{OM}}
\]

By solving the linear cost rate equations simultaneously, the cost flow rates and unit cost of exergy for all streams will be calculated.

Examine the costs of system components.

The main difference between ordinary economic analysis and the present analysis is that this analysis takes place at the level of the system components and also shows how the cost distribution in system components is. CI superscript indicates the capital investment:

\[
\dot{Z}_{\text{CI}} = \frac{\text{CC}_{\text{L}} \ast \text{PEC}_{k}}{\tau} \frac{\text{PEC}_{k}}{\text{PEC}_{\text{tot}}}
\]

\[
\dot{Z}_{\text{OM}} = \frac{\text{OMC}_{L} \tau}{\tau} \frac{\text{PEC}_{k}}{\text{PEC}_{\text{tot}}}
\]

\[
\dot{Z}_{\text{tot}} = \frac{\text{CC}_{\text{L}}}{\tau}
\]

\[
\dot{Z}_{\text{OM}} = \frac{\text{OMC}_{L} \tau}{\tau}
\]

where, \( \dot{Z}_{\text{CI}}, \dot{Z}_{\text{OM}}, \dot{Z}_{\text{CI}}^\text{tot}, \dot{Z}_{\text{OM}}^\text{tot} \) and \( \text{PEC}_{k} \) denote the capital investment of the component \( k \) ($/h), operating and maintenance cost of the component \( k \) ($/h), the total capital investment ($/h), the total operating and maintenance cost ($/h), and the purchasing equipment cost of the component \( k \) ($), respectively.

The total cost of the K component is calculated from the following equation:

\[
\dot{Z}_{k} = \dot{Z}_{\text{CI}} + \dot{Z}_{\text{OM}}
\]

The fuel cost ($/h) is calculated by dividing the annualized levelized cost to the annual operating time:

\[
\dot{C}_{\text{F,ste}} = \frac{\text{FC}_{\text{L,steam}}}{\tau}
\]

\( C_{F} \) and \( Z_{k} \) are the input data of the exergoeconomic analysis.

Finally, the objective function is defined as:

\[
\dot{C}_{F,K} = \dot{C}_{F,K} + \dot{Z}_{k}
\]
3.4.1 | Exergy cost flow

In the process of costing exergy flow a cost is attributed to each exergy flow. The cost rate for the jth material stream, $\dot{C}_j$, is expressed as the product of the exergy rate of flow, $\dot{E}_j$, at the average cost per exergy unit, $c_j$:

$$\dot{C}_j = c_j \dot{E}_j$$  \hspace{1cm} (51)

Exergy transfers associated with heat transfer and work are also a cost is attributed:

$$\dot{C}_W = c_W \dot{W}$$  \hspace{1cm} (52)

$$\dot{C}_q = c_q \dot{E}_q$$  \hspace{1cm} (53)

where $c_j$, $c_W$, and $c_q$ are the average cost per exergy unit and is dollar per kilojoules of exergy. The cost of each of the material or energy flows in the system is calculated by the cost balance equations and the auxiliary cost equations.$^{35}$

Cost balance

The exergy costing process involves cost-balancing equations, which are typically formulated separately for each component of the system. The cost balance used for the Kth system component states that the total cost of the output stream is equal to the total cost of the inputs plus the costs associated with the investment and the costs of operation and maintenance.

$$\sum_{j=1}^{n} \dot{C}_{j,k,\text{in}} + \dot{Z}_{k}^{\text{CI}} + \dot{Z}_{k}^{\text{OM}} = \sum_{j=1}^{m} \dot{C}_{j,k,\text{out}}$$  \hspace{1cm} (54)

Cost balance equations are usually written in such a way that all sentences are positive. As in the exergy analysis for each component of the system, a fuel and a product are determined, the cost flow rate of the fuel $\dot{C}_F$ and product of a component $\dot{C}_P$ are in the same fashion and obtained similar to the exergy flow rate calculation of $\dot{E}_F$ and $\dot{E}_P$. Therefore, the cost balance will be as follows:

$$\dot{C}_P = \dot{C}_F + \dot{Z} - \dot{C}_L$$  \hspace{1cm} (55)

### TABLE 4 | The cost rate balance and auxiliary equation for main equipment in F-type arrangement

| Component | Cost flow rate balance equation | Auxiliary equation |
|-----------|---------------------------------|--------------------|
| Evaporator | $C_1 + C_{1w} + Z_{1w} = C_2 + C_{2w}$ | $C_{1w} = C_{2w}$  |
| Pump      | $C_1 + C_{WP} + Z_{2W} = C_2$ | $C_{WP} = C_{W}$  |
| Turbine   | $C_4 + Z_4 = C_5 = C_{WT}$ | $C_4 = C_5$  |
| IHX       | $C_2 + Z_{2m} = C_3 + C_5$ | $C_2 = C_3$  |
| Condenser | $C_1 + C_{1v} + Z_{cond} = C_4 + C_{2v}$ | $C_1 = C_4$  |

$$\dot{C}_{P,tot} = \dot{C}_{F,tot} + Z_{tot}$$  \hspace{1cm} (50)

### TABLE 5 | Exergy flow rates, cost flow rates and unit exergy costs for n-heptane fluid with the R-type arrangement

| Material stream | $E_i$ (kW) | $C$ ($/h$) | $c$ ($$/Gj$$) |
|-----------------|------------|------------|---------------|
| 101             | 5.1        | 0.000011   | 2.156         |
| 102             | 96.5       | 0.000739   | 7.658         |
| 103             | 221.1      | 0.001557   | 7.042         |
| 104             | 4792.5     | 0.010335   | 2.156         |
| 105             | 531.2      | 0.001145   | 2.156         |
| 106             | 304.8      | 0.000657   | 2.156         |

### FIGURE 9 | The efficiency changes of the first law of thermodynamics in different arrangements for refrigerants of R123, R114, n-butane, n-pentane, n-heptane, and toluene using EES modeling

### FIGURE 10 | The efficiency changes of the second law of thermodynamics in different arrangements for refrigerants of R123, R114, n-butane, n-pentane, n-heptane, and toluene using EES modeling
In above equation, $\hat{C}_L$ is the cost of waste due to the exergy which is wasted to the environment.

The cost balances and auxiliary equations related to each component of the cycle are embedded in Table 4.

By writing the cost balance equations for each component of a cycle and auxiliary equations, it leads to a linear system of equations. By solving this, the costs of unknown streams are obtained. The exergoeconomic assessments of cycles could be done through the exergoeconomic parameters. These parameters consist of exergy flow rates ($\dot{E}$), cost flow rates ($\dot{C}$) and unit exergy costs ($c$), the cost flow rate related to the exergy destruction ($\dot{C}_D$) and the exergoeconomic factor ($f_k$)\(^{42}\) presented in Table 5. The exergoeconomic factor ($f_k$) is a parameter that indicates the relative importance of the cost of a component to the cost of exergy destruction and loss related to that component.\(^{43}\)

3.5 Technical optimization, objective function, and designing variables

The generated power with regards to the obtained thermal power is an effective criterion for evaluating the ORC systems which is selected as an objective function in processes. Therefore, the objective function for the optimization of processes can be expressed as follows:

$$ f(x) = \max \left( \frac{\dot{W}_{\text{net}}}{\dot{Q}_h} \right) $$

where $\dot{W}_{\text{net}}$ is the net generated power of the refrigerant compression and expansion and $\dot{Q}_h$ is the received thermal power from the heat source. In that, $\dot{W}_{\text{net}}$ described as below:

$$ \dot{W}_{\text{net}} = \sum \dot{W}_{\text{compressor}} - \sum \dot{W}_{\text{expander}} $$

Several factors affect the performance of processes with different arrangements. The most important are high and low pressure in the cycle $p_{102}$ and $p_{101}$, the refrigerant temperature before and after the heat source $t_{103}, t_{104}$, and the outlet temperature of the condenser $t_{101}$. Also, the minimum allowable temperature difference between the hot and cold fluid in heat exchangers cannot be $<2^\circ C$ and the optimization problem is to find the optimal values of the parameters to maximize power production under the aforementioned restrictive conditions. Therefore, the designing parameters include:

$$ X_{\text{ORC}} = [p_{101}, p_{102}, t_{103}, t_{104}]^T $$

![FIGURE 11](image1.png) The mass flow rate changes of the working fluid in different arrangements for refrigerants of R123, R114, n-butane, n-pentane, n-heptane, and toluene using EES modeling.

![FIGURE 12](image2.png) The changes of the first-law efficiency with respect to the evaporator pressure changes for fluid of R123, R114 and Toluene before optimization (for the case that heat source temperature is a function of maximum temperature of working fluid in the cycle) using EES modeling.
The following constraints are effective in the optimization process:

\[
\text{main approach (LNG} - 100) \geq 2 \quad (59)
\]

\[
\text{main approach (LNG} - 101) \geq 2 \quad (60)
\]

3.6 | Economic optimization, objective function, and designing variables

The process that has the best performance from technical optimization viewpoint, will be optimized economically through the final cost method with respect to the exergo-economic concept.

In this optimization, designing variables, and constraints are adjusted and considered for a 30 years period and the aim is to maximize the difference between the revenue of the produced electricity and the final costs of the cycle in the period of 30 years. Finally, the objective function is defined as follows:

\[
f(x) = \min \left( \sum_{i=1}^{30} \text{income}(i) - \text{Fuel Cost}(i) - \text{O&M Cost}(i) - \text{Capital Cost} \right) \quad (61)
\]

In the above formula, income is the revenue of the produced electricity by the turbine, O&M is the equipment maintenance costs and Capital Cost refers to the cost of investment.

4 | RESULTS AND DISCUSSION

The results of the thermodynamic modeling, exergetic modeling of ORC and energy and exergy analysis in the EES software are obtained, considering the first and second laws efficiencies in M, R, H, and F-type arrangements. The flow rate of each working fluid (organic fluid) in the mentioned arrangements is calculated. According to the first-law and second-law efficiency and the required flow rate for each arrangement, the optimal arrangement for each organic fluid is presented. The variations of the first-law and second-law
efficiency and the amount of irreversibility in the components of the cycle are studied considering the changes of evaporator and condenser's temperature and pressure. Six organic fluids namely, R123, R114, n-butane, n-pentane, n-heptane, and toluene are employed. For each of the considered fluids, the first-law efficiency in different cycle arrangements is studied. In Figure 9, all values are obtained for the thermal efficiency at the maximum pressure of 2000 kPa and the condenser temperature of about 35°C.

Results show that using recuperator for fluids R114, n-heptane, and toluene and intermediate closed heater for n-butane and R123 and intermediate closed heater and recuperator for n-pentane, will be more useful and will result into higher first-law efficiency.

Selection of the optimal arrangement for an ORC requires more information that is obtained through exergy analysis of the aforementioned arrangements. Therefore, the second-law efficiency parameter is added to the study shown in Figure 10.

Using intermediate recuperator for R114, n-pentane, n-heptane, and toluene and intermediate and final recuperators simultaneously for the other two fluids will lead to higher second-law efficiency.

The mass flow rate of the working fluid in each of the arrangements can be one of the key parameters in decision-making for a proper arrangement. Due to the high cost of the organic fluids, results draw the attention toward an arrangement that requires less mass flow rate. The mass flow rate of each of fluids in different arrangements is obtained, in Figure 11.

All processes are designed in accordance to the mentioned and assumed conditions. One of the most important and most influential parameters in the analysis of the power generation cycles, is the maximum cycle pressure or the evaporator pressure. This parameter directly affects the first-law and second-law efficiency. In Figures 12-15 this effect can be observed.

According to Figures 9-11, the optimum arrangement for each refrigerant is selected and presented in Table 6.

As shown in Figures 12 and 13, the efficiency of the first thermodynamic law for various working fluids is illustrated to be 7%-25%, which is consistent with the outcome of Dresher and Bruggemann.13

As shown in Figures 12 and 13, the efficiency of the first thermodynamic law for various working fluids is illustrated to be 7%-25%, which is consistent with the outcome of Dresher and Bruggemann.13

It is also clear from the comparison of Figures 16 and 17 that the toluene and n-heptane working fluids exhibit the highest efficiency of the second law of thermodynamics (the least amount of exergy destruction) among other working fluids, which is consistent with the results of Gang Li’s paper.

The stream data of each node in the optimized cycle is extracted from Aspen HYSYS Software and is presented in Table 7, containing the information about entropy, enthalpy, exergy, and mass flow rate.

Among the six studied fluids, toluene has the maximum thermal efficiency. The cause of increase in the thermal efficiency is that, with the rise of the evaporator pressure, the saturation steam temperature will be raised at the turbine inlet which that leads to the rise of the fluid enthalpy. Toluene which has the highest critical point temperature, has the maximum thermal efficiency and n-butane and R114, that have the lowest critical point temperature, have the minimum thermal efficiencies. For fluids that use the F-type arrangement, such as; R123, the increase in the evaporator pressure will always result into a rise in

| Organic fluid | R123 | R114 | n-butane | n-pentane | n-heptane | Toluene |
|---------------|------|------|----------|-----------|-----------|---------|
| The optimum arrangement type | F    | H    | H        | R         | R         | R       |

**TABLE 6** The optimum arrangement type of the organic fluids for low grade heat recovery application
245

SHAMS GHOREISHI ET AL.

the second-law efficiency but fluids that using the R &H-
type arrangements, the second-law efficiency at a certain
pressure will reach its maximum value. The cycle second
law of thermodynamics will reach its maximum value for
fluids n-butane and R114 in the H-type arrangement at a
pressure of about 2 MPa. While the pressure for fluids that
using R-type arrangement is at about 1 MPa. In order to
study the minimum irreversibility or maximum second-law
efficiency in terms of evaporator’s pressure, Figures 14 and
15 are plotted under the same conditions of superheating
and minimum temperature approach of 15°C in evapora-
tor. But in Figures 16 and 17, depicted for constant heat
source temperatures (250°C and 350°C), cycles are consid-
ered having the same superheating condition happening in
real cycle condition. It is clear that, the first-law efficiency
and mass flow rate in Figures 14-17 are the same as these
parameters are independent of heat source temperature and
calculated from the refrigerant saturated temperature at the
maximum cycle pressure (20 bar).

Figure 18 illustrates the effect of the superheated vapor of
the turbine inlet on the thermal efficiency. In this figure, the
evaporator and the condenser’s pressure are considered con-
stant values of 600 and 110 kPa, respectively. The optimal
arrangement for fluids such as; R114, n-butane and R123 is
kind of arrangement that in which intermediate recuperator is
used. The working fluid pressure extracted from the turbine
which then is sent to the closed feed water heater is called ex-
traction pressure. The extraction pressure is a fraction of the
turbine inlet pressure calculated by

\[ P = p_n^{inlet} \]

where \( n \) denotes the power.

In Figure 19, the effect of the power variation of the ex-
ttracted pressure on the thermal efficiency of the cycle is
shown.

4.1 | Results of the technical optimization

In the section of technical optimization, cycles are op-
timized for producing of a constant net power with the
assist of Aspen HYSYS software and the obtained results are given in Table 8. After optimization of the cycles, the cycle that has the higher efficiency, will be selected for economic optimization. From the study of the optimized results in Aspen HYSYS software, it is defined that, the ORC with recuperator (R-type arrangement) has the highest efficiency. ORC cycles have the first law cycle efficiency between 15.47% and 25.57%. The lowest amount occurs when the H-type arrangement exists and the refrigerant is n-butane and the higher amount happens for n-heptane refrigerant with the R-type arrangement. Among the cycles with high efficiency, the R-type arrangement with refrigerants such as: toluene and n-heptane; these have similar efficiencies but the flow rate of toluene is lower, compared to the refrigerant, n-heptane. The results of the technical optimization of the cycles are shown in Table 8. By comparing the results of this table with the results presented in the Gang Li43 for high-temperature ORC cycles, it is evident that, although the boundary conditions in this study (minimum pressure and cold air temperature) are close to actual conditions, there was no noticeable drop in the efficiency based on the first law of thermodynamics and it is almost the same as the results presented by the Gang Li et al43 In the following, these two cycles will be further studied.

In Figures 20 and 21, these two cycles are shown according to the inlet smoke temperature enters the stack:

Also, based on the results of the exergoeconomic analysis of the n-heptane cycle which is presented in Table 9, it can be seen that the obtained values, and parameters in this study, despite the differences in the working fluid and boundary conditions and the pump efficiency, are in agreement with the results presented by Shokati et al44

According to the Figures 20 and 21, it is clear that the effect of the rise of the heat source temperature on the cycle efficiency has a linear trend, with the difference that in the toluene cycle, the minimum possible temperature for heat source is about 280°C and in the n-heptane cycle is about 255°C. In other words, at the current pressure, reducing the heat source temperature results into the refrigerant condensation which is problematic before entering into the turbine. Therefore, the efficiency of the Toluene cycle is higher than in comparison to the n-heptane cycle and covers a broader temperature range. Hence, the n-heptane cycle was selected for economic optimization. Then, through the exergoeconomic and economic strategies, the cycle will be enhanced from the economic viewpoint.

### 4.2 The exergoeconomic evaluation results

After technical optimization, the economic optimization of the cycle is discussed by exergoeconomic evaluation. In these calculations, the annual interest rate of 10% and the BL of 30 years are considered.
One way to prioritize components from exergoeconomic viewpoint is, sorting them based on the amount of $\dot{z}_k$ ($$/h) + C_D$$ ($$/h$). This value is calculated for all of the cycle components. If this value is higher for a component compared to the other ones in a cycle, that component has higher priority to be studied and analyzed. As it is obvious, turbine can be considered as the key component from exergoeconomic point of view.

In the next stage, components will be arranged based on the $f$ value. In components with high values of $f$, due to the high investment cost, replacing it with a less expensive component may be effective economically. Moreover, in components with low $f$ value, because of the high maintenance cost compared to the investment costs, components are not economically effective and their replacement should be examined.

Therefore, the investment cost of turbine should be decreased due to its high value of $f$. In this case, maximum pressure cycle could be considered as a key variable for decreasing the turbine investment cost.

### 4.3 Techno-economic optimization of the system

Aspen HYSYS software is used to optimize the system economically. There are generally 3 steps for adjusting the HYSYS optimizer. The first is to select the primary variables, which are actually the same variables that must be manipulated to optimize the objective function. In this study, the pressure ratio is chosen as the primary variable.

In the optimization process, the total cost of products and the revenue from the generated power are selected as the objective function. The effect of the maximum pressure change is evaluated on the economic objective function in the range of 5-20 bars. Figure 22 shows the effect of changing of the cycle maximum pressure on the economic objective function.

Based on Figure 22, the economic objective function is presented for the pressure range (5-20 bar) and will be improved with the tendency toward less pressure. In other
words, the final net revenue at lower pressure ratios will be increased and has no extreme point. Afterward, the effect of the cycle maximum pressure change on the technical objective function will be studied.

As can be seen in Figure 23, by reducing the maximum pressure, the technical efficiency of the cycle will be decreased but this trend will be stopped at the maximum pressure of 8.4 bar and at this pressure, there will be an increase in the cycle efficiency. The failure on the curve is that, from 20 bar to 8.4 bar, condensation (phase change) of the fluid happens in evaporator. But at pressures lower than 8.4 bar, condensation (phase change) happens in recuperator. Therefore, at the pressures lower than 8.4 bar, evaporator needs lower heat to superheat the fluid. Hence, with regards to the Energy efficiency formula \( \eta = \frac{W}{Q} \) by reducing the required heat, the efficiency will increase.

In Table 10, the results of the initial state (maximum cycle pressure of 2000 kPa) are compared with the optimum state
(maximum cycle pressure of 840 kPa). Economic objective function is improved about 44% in the optimum state compared to the initial state. Whereas, the technical objective function decreases just 5%.

4.4 Validation and comparison of the results of the simulation and the results of thermodynamic modeling

To validate the obtained results were compared with the results of Gang Li. To validate the results, the boundary conditions taken by Gang Li were employed as shown in Table 11.

| Equipment                  | \( C_D \) ($/h) | \( \varepsilon \) (%) | \( C_D + Z_k \) ($/h) | \( f_k \) (%) |
|----------------------------|----------------|-------------------|-------------------|-------------|
| Pump P-100                 | 0.27           | 63.63             | 1.05              | 74.3        |
| Cooler E-101               | 0.17           | 69.27             | 9.26              | 98.2        |
| Turbine K-100              | 4.42           | 64.68             | 41.18             | 89.3        |
| Heat Exchanger LNG-100     | 0.87           | 41.11             | 1.98              | 56.1        |
| Evaporator                 | 4.54           | 67.79             | 10.16             | 55.3        |

In Gang Li’s work, the maximum cycle temperature is 250°C. Therefore, by obtaining the results for the maximum temperature of 250°C, the results were compared with. As shown below, results of the current modeling are in a fairly good agreement with the Gang Li’s findings (Figure 24).

Also, Table 12 shows a sample of stream data of the current modeling for \( P_{cond} = 1000 \) kPa, \( T_{evp} = 20^\circ\)C and Intermediate exchanger effectiveness = 0.6.

As seen, it is clear that the results have a good consistency with each other. The difference of the results of the current modeling with Gang Li’s is less than 3%. The reason may be due to the different equation of states models used. In other words, using different equation of state models with different thermodynamics properties give different results which in the current modeling, the difference between the results is fairly low.

The results of the simulation and thermodynamic modeling of the net output power and thermal efficiency for refrigerant R245fa are also validated to determine the accuracy of the results obtained from the simulation of the ORC system.

As shown in Table 13, the refrigerant R245fa is modeled as the working fluid in the ORC system in accordance with the assumptions made that, the isentropic efficiencies of the pump
and turbine are equal to 0.65 and 0.85 and $T_{\text{cond}} = 30^\circ$C and $T_{it} = 100^\circ$C. Then, the thermal efficiency of the ORC system with the refrigerant, R245fa, as the working fluid will be compared with the previously published results\(^4^5\) at $T_{\text{cond}} = 26^\circ$C and without considering that the working fluid is superheated at the turbine inlet Figure 25. In this comparison, it should be noted that the assumptions of the isentropic efficiencies of the pump and turbine, which are in accordance with the previous studies assumptions,\(^4^5\) are considered to be equal to 0.75 and 0.8, respectively. Furthermore, the simulation results are validated with the experimental results of Declay et al,\(^4^6\) for an ORC system with the working fluid, R245fa, at the condition of $T_{it} = 100^\circ$C, is shown in Figure 26. The turbine inlet pressure is kept constant at the amount of $P_{it} = 1.2$ MPa and the turbine outlet pressure will be changed with the change of the exhaust condenser temperature $t$ and thus one of its results is the change of the system pressure ratio in the range of 3.1-5.8. From the obtained results, it can be seen that at low pressure ratio, a small deviation would be created which is justified with pointing out the assumption of constant efficiencies of the pump and turbine despite the pressure ratio change and using of the Peng-Robinson equation of state in this simulation. Overall, by studying Table 12 and Figures 25 and 26, it is clear that the results of the simulation obtained in this study are compatible with the results reported in Saleh et al,\(^2^6\) Shengjun et al,\(^4^5\) and Declay et al.\(^4^6\)

### Table 10: Comparison of the optimized results with the results of initial for n-heptane fluid with the R-type arrangement

| Parameter                             | Initial state | Optimum state |
|---------------------------------------|---------------|---------------|
| Pressure ratio                        | 18.18         | 7.64          |
| $P_{\text{min}}$ (kPa)                 | 110           | 110           |
| $P_{\text{max}}$ (kPa)                 | 2000          | 840           |
| Power consumption of Compressor (kW)  | 3.175         | 1.631         |
| Power generation of Turbine (kW)      | 103.2         | 101.6         |
| Received thermal power (kW)           | 391.1         | 484.8         |
| First law of thermodynamics (%)       | 25.57         | 20.62         |
| Net final income ($/kW·h)             | 423.1         | 609.9         |

5 Conclusion

In this study, 6 organic working fluid for waste heat recovery application were evaluated under 4 common ORC arrangements. According to the results of the first and second laws of thermodynamic and refrigerant mass flow rate, the optimum arrangement for each working fluid was selected. The optimum arrangement for each refrigerant was technically optimized in Aspen HYSYS to maximize the first law of thermodynamic and results were compared. Among the 6 studied fluids, toluene showed the maximum increasing in thermal efficiency as evaporator pressure rises. Moreover, results revealed that fluids with higher critical points bring higher thermal efficiency of ORC. In the case of the second-law efficiency, the conditions will be different when heat source temperature is a function of maximum temperature of working fluid in the cycle. It was concluded that increasing of the evaporator pressure will not always lead to an increase in the second-law efficiency. For fluids that use the F-type arrangement, the increase in the evaporator pressure will always result a rise in the second-law efficiency but fluids that using the R or H-type arrangements, the second-law efficiency at a certain pressure will reach its maximum value. Without superheating condition and assuming constant temperature for heat source, the second-law efficiency will be reduced for all 6 organic fluids as heating source temperature rises but with the rise of evaporator pressure, the second-law efficiency will always increase. As evaporator
pressure increases, the evaporator outlet temperature will be increased and obviously, difference of evaporator temperature with heat source temperature decreased; so, the value of irreversibility will decline and thus the efficiency of the second-law will be raised. In high-temperature waste heat recovery application, only toluene, n-heptane, and R123 can be used, because only these fluids showed the efficiency of above 30% at the maximum pressure of 3 Mpa. As intermediate pressure in turbine increases, the thermal efficiency will be increased to the extent that this amount reaches its maximum and then decreases with further increasing of intermediate pressure of turbine.

N-heptane cycle, under R-type arrangement was chosen as the optimum cycle among the other candidate cycles. Afterward, both technical and economic objective functions of the above cycle were simultaneously optimized with respect to exergoeconomic results by assuming maximum cycle pressure as a variable. By decreasing maximum cycle pressure, thermal efficiency and net final income were decreased and increased respectively and the optimum point was observed in maximum pressure of 840 kpa. The summery of the obtained results of this study are given as follows:

- The first-law efficiency will be increased by using intermediate and final recuperators for all fluids in any cycle arrangements with respect to the M-type arrangement. The

---

**TABLE 13** The comparison of the achieved results of the current study with the presented results under conditions of $T_{eva} = 100°C$, $T_{cond} = 30°C$, $T_{it} = 100°C$, and $m_r = 33.42$ kg/s

| Parameter          | $P_{ot}$ (MPa) | $P_{it}$ (MPa) | $W_{net}$ (kW) | $\eta_{th}$ (%) |
|--------------------|----------------|----------------|----------------|-----------------|
| R245fa             | 0.1790         | 1.275          | 915.9          | 13.54           |
| R245fa$^{10}$      | 0.1785         | 1.265          | 997.1          | 12.49           |
| Error (%)          | 0.28           | 0.79           | 8.14           | 8.41            |

**FIGURE 25** The comparison of the thermal efficiency results of the current research with the corresponding work for R245fa as refrigerant in the ORC system.
rise of efficiency ranges between 2% and 8% without superheating conditions. This difference in the behavior of organic fluids is due to the difference between the saturation vapor pressure at ambient temperature, the turbine inlet temperature at constant pressure of 2000 kPa and the molecular weight.

- In all cases, employing of recuperators lead to the rise of the second-law efficiency. Utilization of recuperator will enhance the second-law efficiency about 7-15 percent. It should be noted that this results are investigated regardless of the superheating condition and the maximum cycle temperature is variable for each refrigerant and it is considered to be equal to the saturated temperature in the maximum pressure of cycle. Increasing the cycle evaporator pressure, the first-law efficiency will be always increased for each of six fluids (Figures 8 and 9). This increase in efficiency may reach 15%, this occurs when the pressure rises from 500 to 3 MPa.

- Superheating of the working fluid at about 30°C, will only increase the thermal efficiency about 1%-2% in high pressure (20 bar). Due to the fact that basically superheating of the organic fluid may occurs at low pressures, superheating of the working fluid in the evaporator outlet should always be avoided in high pressure.

- Thermodynamic optimized cycle with the refrigerant n-heptane under the R-type arrangement depicts the maximum performance for generating the net power of 100 kW. Finally, changing the maximum cycle pressure of the considered cycle, according to economic and technical objective functions, the optimum maximum pressure is 840 kPa. At this condition, the thermal efficiency of 20.62% is estimated.

- According to the technical and economic objective functions, the maximum pressure of 840 kPa and the minimum pressure of 110 kPa are considered as the optimal values for n-heptane cycle under R-type arrangement. In this state, net final income of the cycle increased for 44.2% and thermal efficiency just decreased for lower than 5% in comparison with its initial state.

**NOMENCLATURE**

| Symbol | Description |
|--------|-------------|
| Q      | Heat transfer (kW) |
| $T_H$  | Exhaust hot gas temperature |
| $e_r$  | Recuperator efficiency coefficient |
| $h_{PR}$ | Heat of reaction (kJ/mole) |
| $Y$    | Mass flowrate of extraction working fluid of turbine stage |
| $T_0$  | Reference temperature (°C) |
| $T_0$  | Ambient temperature (°C) |
| P      | Pressure (kPa) |
| $\dot{m}$ | Mass flow rate of working fluid (kg/s) |
| W      | Power (kW) |
| $S$    | Entropy (kJ/(kgK)) |
| $h$    | Enthalpy (kJ/kg) |
| R      | Gas constant |
| ORC    | Organic rankine cycle |
| $V$    | Specific volume |
| con    | Condenser |
| sys    | System |
| CFFH   | Closed feed fluid heater |
| EPG    | Electrical power generation |

(Continues)
NOMENCLATURE (Continued)

| Q  | Heat transfer (kW) | $T_H$ | Exhaust hot gas temperature |
|----|--------------------|-------|-----------------------------|
| FLTE | First law of thermodynamic efficiency | $p$ | Product |
| SLTE | Second law of thermodynamic efficiency | $t$ | Turbine |
| NFI | Net final income | $n$ | Power of extraction pressure |
| IT | Ignition temperature ($^\circ$C) | $I$ | Exergy loss (kJ/s) |

OM | Operating and maintenance | cl | Capital investment |

Subscripts
- C: Condenser
- x: Mole fraction
- k: Binary iteration coefficient
- $\eta$: Efficiency
- 1,2,s,3,4,5,6,7,s,8,9,10,10s,11: State points in Figure 8
- $\epsilon_r$: Recuperator efficiency coefficient
- gen: Generated
- $w$: Related to transfer of workdone
- tot: Total
- i: Input
- e: Output

ACKNOWLEDGMENTS

This research was supported by the National Natural Science Foundation of China (Grant No. 51778511), Hubei Provincial Natural Science Foundation of China (Grant No. 2018CFA029), Key Project of ESI Discipline Development of Wuhan University of Technology (Grant No. 2017001), the Scientific Research Foundation of Wuhan University of Technology (Grant No. 40120237).

ORCID

Milad Sadeghzadeh https://orcid.org/0000-0001-8574-5463
Mohammad Hossein Ahmadi https://orcid.org/0000-0002-0097-2534

REFERENCES

1. Bianchi M, De Pascale A. Bottoming cycles for electric energy generation: parametric investigation of available and innovative solutions for the exploitation of low and medium temperature heat sources. Appl Energy. 2011;88(5):1500-1509.
2. Chen H, Goswami DY, Stefanakos EK. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. Renew Sustain Energy Rev. 2010;14(9):3059-3067.
3. Caf A, Urbanc D, Trop P, Goricanec D. Exploitation of low-temperature energy sources from cogeneration gas engines. Energy. 2016;108:86-92.
4. Lecompte S, Huisseune H, van den Broek M, Vanslambrouck B, De Paepe M. Review of organic Rankine cycle (ORC) architectures for waste heat recovery. Renew Sustain Energy Rev. 2015;47:448-461.
5. Lizarte R, Palacios-Lorenzo ME, Marcos JD. Parametric study of a novel organic Rankine cycle combined with a cascade refrigeration cycle (ORC-CRS) using natural refrigerants. Appl Therm Eng. 2017;127:378-389.
6. Yamamoto T, Furuhata T, Arai N, Mori K. Design and testing of the organic Rankine cycle. Energy. 2001;26(3):239-251.
7. Saloux E, Sorin M, Nesreddine H, Teyssedou A. Reconstruction procedure of the thermodynamic cycle of organic Rankine cycles (ORC) and selection of the most appropriate working fluid. Appl Therm Eng. 2018;129:628-635.
8. Zhang HG, Wang EH, Fan BY. A performance analysis of a novel system of a dual loop bottoming organic Rankine cycle (ORC) with a light-duty diesel engine. Appl Energy. 2013;102:1504-1513.
9. Freeman J, Hellgardt K, Markides CN. Working fluid selection and electrical performance optimisation of a domestic solar-ORC combined heat and power system for year-round operation in the UK. Appl Energy. 2017;186:291-303.
10. Freeman J, Hellgardt K, Markides CN. An assessment of solar-powered organic Rankine cycle systems for combined heating and power in UK domestic applications. Appl Energy. 2015;138:605-620.
11. Wang EH, Zhang HG, Zhao Y, Fan BY, Wu YT, Mu QH. Performance analysis of a novel system combining a dual loop organic Rankine cycle (ORC) with a gasoline engine. Energy. 2012;43(1):385-395.
12. Heberle F, Preißinger M, Brüggemann D. Zeotropic mixtures as working fluids in organic Rankine cycles for low-enthalpy geothermal resources. Renew Energy. 2012;37(1):364-370.
13. Drescher U, Brüggemann D. Fluid selection for the organic Rankine cycle (ORC) in biomass power and heat plants. Appl Therm Eng. 2007;27(1):223-228.
14. Heberle F, Brüggemann D. Exergy based fluid selection for a geothermal organic Rankine cycle for combined heat and power generation. Appl Therm Eng. 2010;30(11-12):1326-1332.
15. Quoilin S, Van Den Broek M, Declaye S, Dew Allauff P, Lemort V. Techno-economic survey of organic rankine cycle (ORC) systems. Renew Sustain Energy Rev. 2013;22:168-186.
16. Lemort V, Quoilin S, Cuevas C, Lebrun J. Testing and modeling a scroll expander integrated into an organic Rankine cycle. Appl Therm Eng. 2009;29(14-15):3094-3102.
17. Bao J, Zhao L. A review of working fluid and expander selections for organic Rankine cycle. Renew Sustain Energy Rev. 2013;24:325-342.
18. Oyewunmi O, Markides C. Thermo-economic and heat transfer optimization of working-fluid mixtures in a low-temperature organic Rankine cycle system. Energies. 2016;9(6):448.

19. Markides CN. Low-Concentration solar-power systems based on organic rankine cycles for distributed-scale applications: overview and further developments. Front Energy Res. 2015;3:1-16.

20. Tchanche BF, Lambrinos G, Frangoudakis A, Papadakis G. Exergy analysis of micro-organic Rankine power cycles for a small scale solar driven reverse osmosis desalination system. Appl Energy. 2010;87(4):1295-1306.

21. Mohammadi A, Ashouri M, Ahmadi MH, Bidi M, Sadeghzadeh M, Ming T. Thermoeconomic analysis and multiobjective optimization of a combined gas turbine, steam, and organic Rankine cycle. Energy Sci Eng. 2018;6:506-522.

22. Mirzaei M, Ahmadi MH, Mobin M, Nazari MA, Alayi R. Energy, exergy and economics analysis of an ORC working with several fluids and utilizes smelting furnace gases as heat source. Therm Sci Eng Prog. 2018;5:230-237.

23. Ashouri M, Razi Astaraei F, Ghasempour R, Ahmadi MH, Feidt M. Thermodynamic and economic evaluation of a small-scale organic Rankine cycle integrated with a concentrating solar collector. Int J Low Carbon Technol. 2017;12(1):54-65.

24. Zhang H, Guan X, Ding Y, Liu C. Energy analysis of organic Rankine cycle (ORC) for waste heat power generation. J Clean Prod. 2018;183:1207-1215.

25. Madhawa Hettiarachchi HD, Golubovic M, Worek WM, Ikekagi Y. Optimum design criteria for an organic Rankine cycle using low-temperature geothermal heat sources. Energy. 2007;32(9):1698-1706.

26. Saleh B, Koglbauer G, Wendland M, Fischer J. Working fluids for low-temperature organic Rankine cycles. Energy. 2007;32(7):1210-1221.

27. Nami H, Nemati A, Jabbari Fard F. Conventional and advanced exergy analyses of a geothermal driven dual fluid organic Rankine cycle (ORC). Appl Therm Eng. 2017;122:59-70.

28. Tchanche BF, Papadakis G, Lambrinos G, Frangoudakis A. Fluid selection for a low-temperature solar organic Rankine cycle. Appl Therm Eng. 2009;29(11-12):2468-2476.

29. Taccani R, Obi JB, De Lucia M, Micheli D, Toniato G. Development and experimental characterization of a small scale solar powered organic rankine cycle (ORC). Energy Procedia. 2016;101:504-511.

30. Hung TC, Shai TY, Wang SK. A review of organic rankine cycles (ORCs) for the recovery of low-grade waste heat. Energy. 1997;22(7):661-667.

31. Liu B-T, Chien K-H, Wang C-C. Effect of working fluids on organic Rankine cycle for waste heat recovery. Energy. 2004;29(8):1207-1217.

32. Wang EH, Zhang HG, Fan BY, Ouyang MG, Zhao Y, Mu QH. Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. Energy. 2011;36(5):3406-3418.

33. Tumen Ozdil NF, Segmen MR, Tantekin A. Thermodynamic analysis of an organic Rankine cycle (ORC) based on industrial data. Appl Therm Eng. 2015;91:43-52.

34. Li L, Ge YT, Luo X, Tassou SA. Experimental investigations into power generation with low grade waste heat and R245fa organic Rankine cycles (ORCs). Appl Therm Eng. 2017;115:815-824.

35. Seyedkavoosi S, Javan S, Kota K. Exergy-based optimization of an organic Rankine cycle (ORC) for waste heat recovery from an internal combustion engine (ICE). Appl Therm Eng. 2017;126:447-457.

36. Peng D-Y, Robinson DB. A new two-constant equation of state. Ind Eng Chem Fundam. 1976;15(1):59-64.

37. One C. The estimation of physical properties 1-1 Introduction.

38. Twu CH, Coon JE, Cunningham JR. A new generalized alpha function for a cubic equation of state Part I. Peng-Robinson equation. Fluid Phase Equilib. 1995;105(1):49-59.

39. Bejan A, Tsatsaronis G, Moran M. Thermal Design and Optimization, 1st edn. Canada: John Wiley & Sons Inc; 1996.

40. Sanaye S, Mohammadi Nasab A. Modeling and optimizing a CHP system for natural gas pressure reduction plant. Energy. 2012;40(1):358-369.

41. Zare V. A comparative exergoeconomic analysis of different ORC configurations for binary geothermal power plants. Energy Convers Manag. 2015;105:127-138.

42. Shokati N, Mohammadhkani F, Yari M, Mahmoudi SMS, Rosen MA. A comparative exergoeconomic analysis of waste heat recovery from a gas turbine-modular helium reactor via organic rankine cycles. Sustain. 2014;6(5):2474-2489.

43. Li G. Organic Rankine cycle performance evaluation and thermoeconomic assessment with various applications part I: Energy and exergy performance evaluation. Renew Sustain Energy Rev. 2016;53:477-499.

44. Shokati N, Ranjbar F, Yari M. Exergoeconomic analysis and optimization of basic, dual-pressure and dual-fluid ORCs and Kalina geothermal power plants: A comparative study. Renew Energy. 2015;83:527-542.

45. Shengjun Z, Huaxin W, Tao G. Performance comparison and parametric optimization of subcritical organic Rankine cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation. Appl Energy. 2011;88(8):2740-2754.

46. Declaye S, Quoilin S, Guillaume L, Lemort V. Experimental study on an open-drive scroll expander integrated into an ORC (organic Rankine cycle) system with R245fa as working fluid. Energy. 2013;55:173-183.

How to cite this article: Shams Ghoreishi SM, Akbari Vakilabadi M, Khoeini Poorfar A, et al. Analysis, economical and technical enhancement of an organic Rankine cycle recovering waste heat from an exhaust gas stream. Energy Sci Eng., 2019;7:230-254. https://doi.org/10.1002/ese3.274